Robust Micromachined Gyroscopes for Automotive Applications

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ABSTRACT

This paper reports a micromachined gyroscope with a 2-DoF sense-mode oscillator that provides a flat region in the sense-mode frequency response curve, where the amplitude and phase of the response are insensitive to parameter fluctuations. The sensitivity is also improved by utilizing dynamical amplification of oscillations in the 2-DoF sense-mode oscillator. Thus, improved long-term stability and robustness to fabrication variations, structural and thermal parameter fluctuations and vacuum degradations are achieved, solely by the mechanical system design. Bulk micromachined prototype gyroscopes exhibited a measured noise-floor of $0.64^0/\mathrm{s}/\sqrt{\mathrm{Hz}}$ over a 50Hz bandwidth at atmospheric pressure. The sense-mode response in the flat operating region was also experimentally demonstrated to be inherently insensitive to pressure, temperature and DC bias variations.

1 INTRODUCTION

There is currently a drive within the automotive industry to offer and standardize improved safety and comfort capabilities, including electronic stability control, rollover detection and prevention, and intelligent brake systems. Many of these systems have or are in the process of being realized thanks to low-cost and low-power budget based micro inertial sensors. One of the current automotive efforts is in the integration of gyroscopes in rollover detection systems. These systems have strict requirements for cost, robustness and operation under high g acceleration. To meet these requirements, the gyroscope structure must be robust to fabrication variations and fluctuations in temperature, pressure, and external accelerations. This is a tremendous challenge in single degree of freedom gyroscope systems where frequency and amplitude shifts due to environmental changes result in a large variation in output sensitivity. Towards a robust sensor design for automotive systems, we introduce a novel multi-degree of freedom gyroscope dynamical system.

The two degree-of-freedom (2-DoF) sense-mode dynamical system allows the device to operate in a frequency region that is robust to structural, thermal and pressure fluctuations, where amplitude and phase remain largely unchanged compared to operating close to the frequency peaks. Thus, the disturbance-rejection capability is achieved by the mechanical system instead of active control and compensation strategies. In this paper, we present fundamentals of the device design, electronics design for drive, sensing, and control, and optimization of structural parameters in order to maximize sensitivity while retaining a large bandwidth.

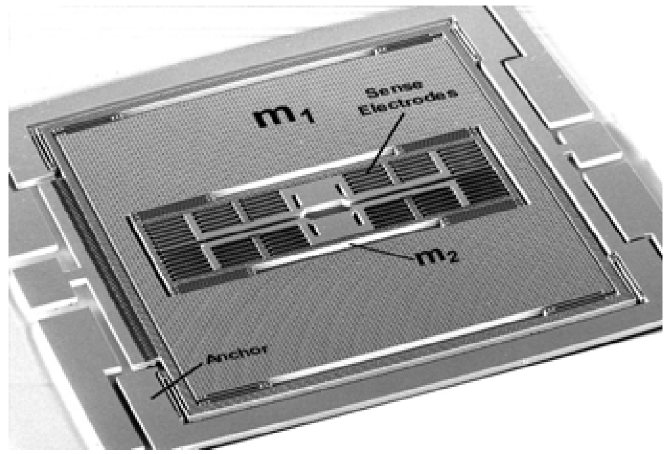

Figure 1: Scanning electron micrograph of the prototype bulk-micromachined 3-DOF gyroscope with 2-DOF sense-mode.

2 THE 3-DoF MICROMACHINED GYROSCOPE STRUCTURE

The presented design concept addresses the following major MEMS gyroscope design challenges: 1) The requirement of precisely controlling the relative location of the drive and sense resonance modes from die to die, from wafer to wafer, and within the required temperature range; 2) Variation in the Coriolis signal phase due to the shift in natural frequencies; 3) Long-term variation in sensitivity due to packaging pressure degradation over time; 4) Minimizing the quadrature error due to mechanical coupling between the drive and sense modes.

The proposed gyroscope dynamical system consists of a 2-DoF sense-mode oscillator and a 1-DoF drive-mode oscillator, formed by two interconnected proof masses (Figure 1). The first mass, m_1, is free to oscillate both in the drive and sense directions, and is

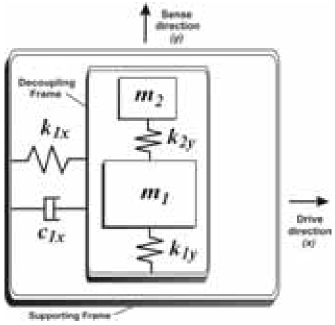

Figure 2: Lumped mass-spring-damper model of the overall 3-DoF gyroscope with 2-DoF sense-mode.

excited in the drive direction. The second mass, m_2, is constrained in the drive direction with respect to the first mass. In the drive-direction, m_1 and m_2 oscillate together, and form a resonant 1-DoF oscillator. The smaller proof mass m_2 forms the passive mass of the 2-DoF sense-mode oscillator (Figure 2), and acts as the vibration absorber to dynamically amplify the sense mode oscillations of m_1.

Conventional Gyroscopes 3-DoF Design Concept

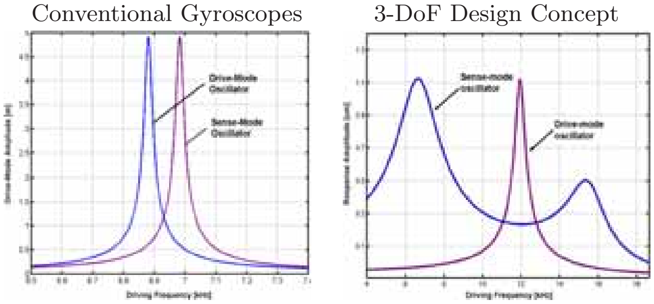

Figure 3: Frequency response comparison of the single-mass conventional gyroscopes and the disclosed Robust Micromachined Gyroscope with 2-DoF Sense-Mode Oscillator.

The 2-DoF sense-mode oscillator provides a frequency response with two resonant peaks and a flat region between the peaks, instead of a single resonance peak as in conventional gyroscopes (Figure 3). The device is nominally operated in the flat region of the sense-mode response curve, where the gain is less sensitive to variations in the natural frequencies and damping. Thus, reduced sensitivity to structural and thermal parameter fluctuations and damping changes are achieved, leading to improved long-term stability and robustness against temperature variations and fabrication variations from device to device.

Furthermore, the sensitivity is improved by utilizing dynamical amplification of oscillations in the 2-DoF sense-mode oscillator. The smaller mass m_2 is employed as the sensing mass, and the larger mass m_1 generates the Coriolis force that excites the 2-DoF sense-mode oscillator. For a given Coriolis force, the amplitude of the sensing mass m_2 increases with decreasing ratio of the proof masses m_2/m_1, which dictates minimizing the mass of m_2 and maximizing the mass of m_1. Also, it is desired that a large Coriolis force is induced on m_1, which also dictates maximizing the mass of m_1. With these two aligning major design constraints, the concept could potentially yield better sensitivity than a conventional gyroscope with mismatched modes; while providing enhanced robustness.

Since the gyroscope structure oscillates as a 1-DoF resonator in the drive direction, the frequency response of the device has a single resonance peak in the drive-mode. The device is operated at resonance in the drive-mode, while the wide-bandwidth frequency region is obtained in the sense-mode frequency response. Thus, the flat region of the sense-mode oscillator is designed to coincide with the drive-mode resonant frequency. This allows utilization of well-proven drive-mode control techniques, while providing robust gain and phase in the sense-mode.

3 PARAMETRIC MODELING

The 3-DoF gyroscope dynamical system is analyzed in the non-inertial coordinate frame associated with the gyroscope. Each of the interconnected proof masses are assumed to be rigid bodies. The following constraints and assumptions further simplify the dynamics of the 3-DoF system: The structure is stiff in the out-of-plane direction; the position vector of the decoupling frame is forced to lie along the drive-direction; m_1 oscillates purely in the sense-direction relative to the decoupling frame; m_1 and m_2 move together in the drive direction; and m_2 oscillates purely in the sense-direction relative to m_1. Thus, the equations of motion of m_1 and m_2 are decomposed into the drive and sense directions to yield

$$\begin{aligned}
(m_1+m_2+m_f)\ddot{x_1}+c_{1x}\dot{x_1}+k_{1x}x_1 &= \\
(m_1+m_2+m_f)\Omega_z^2 x_1+F_d(t) & \\
m_1\ddot{y_1}+c_{1y}\dot{y_1}+k_{1y}y_1 &= \\
k_{2y}(y_2-y_1)+m_1\Omega_z^2 y_1-2m_1\Omega_z\dot{x_1}-m_1\dot{\Omega}_z x_1 & \\
m_2\ddot{y_2}+c_{2y}\dot{y_2}+k_{2y}y_2 &= \\
k_{2y}y_1+m_2\Omega_z^2 y_2-2m_2\Omega_z\dot{x_2}-m_2\dot{\Omega}_z x_2. &
\end{aligned}$$

where Ω_z is the z-axis angular rate, m_f is the mass of the decoupling frame, $F_d(t)$ is the driving electrostatic force applied to the active mass at the driving frequency ω_d. The Coriolis force that excites m_1 and m_2 in the sense direction is $2m_1\Omega_z\dot{x_2}$, and the Coriolis response of m_2 in the sense-direction (y_2) is detected for angular rate measurement.

In order to better understand the effect of increasing frequency on gyroscope performance, a lumped system model was created in MATLAB where the gyroscope dynamics were represented in state space form. Due to the mechanical decoupling of the drive and sense modes, the two dynamic systems can be modeled independently of each other, noting that the decoupling frame parameters can be designed in such a way that the 1-DoF drive mode resonance peak will be in the flat gain region of the sense mode. Thus, the major focus is on the sense mode and how the dynamic system is affected by increasing the operational frequency of the device.

Qualitatively, it can be seen that an ideal sense mode system is one where the motions of the smaller mass are maximized relative to the larger mass. Therefore, by viewing the sense mode in this manner, its response can be maximized and the dynamic system properties can be designed according to the following equation,

$$F_{c1,2}^2 = \frac{F_{n1}^2}{2}\left[1 + f^2(1+v) \pm \sqrt{(1+f^2(1+v))^2 - 4f^2}\right]$$

where $f = \frac{F_{n2}^2}{F_{n1}^2}$ is the uncoupled frequency ratio, $v = \frac{m_2}{m_1}$ is the mass ratio, F_{n1} is the uncoupled natural frequency of the larger mass, and $F_{c1,2}$ are the coupled system natural frequencies [2]. Thus, the system has been reduced to three independent quantities that completely define the coupled system. Using the above equation, a dynamic system can be designed for any mass ratio and desired coupled natural frequencies. Keeping the current mass ratio constant, the effect of increasing the operational frequency of the device was simulated and its effect on static deflection due to g-loads is presented in Figure 4. As expected, the deflection due to acceleration decreases with increasing frequency, and it can be seen that frequencies greater than 5kHz are needed to achieve deflections of less than 1μm for a 100g input, while over 20kHz are needed for 1000g case.

However, by increasing the frequency of the device and keeping the mass ratio constant, the coupled system equations reveal that the minimum achievable peak spacing must also increase. This effect can be seen in Figure 5. As the lowest frequency of the coupled system (F_{c2}) is increased, the minimum peak spacing increases into kHz range for frequencies of 5kHz and above.

The sense mode gain for these frequencies and peak spacings were also computed and presented in Figure 5. The values have been normalized to the current prototype design and it can be seen that increases in frequency cause orders of magnitude drop in gain.

4 SINGLE MASS GYROSCOPE COMPARISON

In order to meet automotive specification for acceleration inputs, it is necessary to increase the lower eigenfrequency of the dual mass system, which in turn mandates that the frequency split between peaks must also increase. As the frequency split increases, the gain decreases while the bandwidth increases. For automotive applications, a bandwidth of 50 Hz is normally sufficient

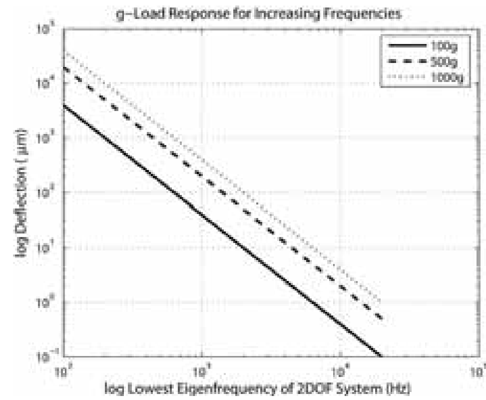

Figure 4: Deflection of sense mass due to acceleration inputs for increases in lowest eigenfrequency and constant mass ratio ($v = 0.0624$).

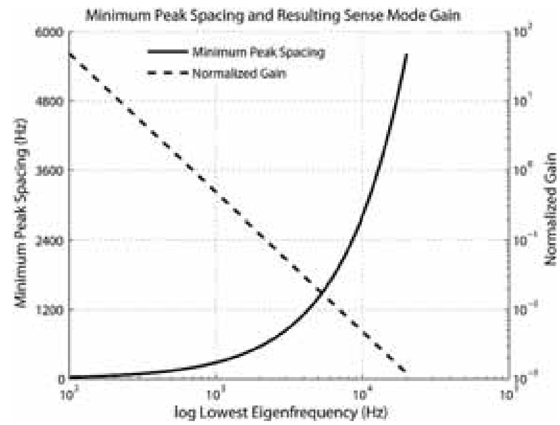

Figure 5: Minimum peak spacing for increases in lowest eigenfrequency and the corresponding gain normalized to current device for constant mass ratio ($v = 0.0624$).

and such a bandwidth is also achievable by a single mass gyroscope by sacrificing gain for increased bandwidth. It is therefore a logical question to ask, at what point is the advantage of a dual mass system lost over the use of a single mass system operating under reduced gain?

For a single mass gyroscope operating at matched drive and sense modes at frequency ω_n, the bandwidth is equal to BW $= \omega_n/2Q$ where Q is the quality factor. If the device is driven at constant amplitude X_D, then the motion induced along the sense direction is

$$\frac{Y}{\Omega} = \frac{2QX_D}{\omega_n} \tag{1}$$

If we substitute in the value for bandwidth, then we see that the output sensitivity is a function of only bandwidth and deflection amplitude

$$\frac{Y}{X_D\Omega} = \frac{1}{\mathrm{BW}} \tag{2}$$

For typical micromachined gyroscopes, Q values range from 50 to 10,000 and natural frequencies range from 1-20 kHz. Thus, bandwidths can range from .05 to 200 Hz.

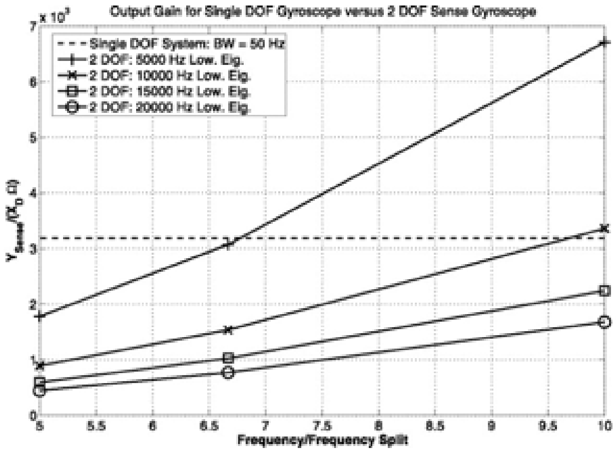

Figure 6: Output gain for the dual mass sense mode structure. Going right represents decreasing frequency splits with 10 representing the minimum reachable dynamic set. For a lower eigenfrequency of 10 kHz or above, a single mass gyroscope has better gain characteristics.

The gain of the sensed mass of the dual mass gyroscope structure has the form

$$\frac{Y_2}{X_D \Omega} \quad = \quad 2\omega_D \left(|G_1|\, m_1 + |G_2|\, m_2 \right), \tag{3}$$

where ω_D is the drive frequency located exactly halfway between the two frequency peaks and G_1 and G_2 are transfer functions for the first and second mass, respectively. From the plot of Equations (2) and (3) for varying lower eigenfrequencies and frequency splits (Figure 6), we see that for a lower eigenfrequency greater then 10 kHz, the dual mass structure is not as advantageous from a gain standpoint versus a single mass device.

5 AGC CONTROL

The advantage of using a single drive mode with a dual sense mode implementation is that an automatic gain control (AGC) can be used to lock the drive mode into resonance. An output signal proportional to the drive velocity is demodulated to obtain a DC signal proportional to the amplitude of the velocity. This signal is compared to a reference voltage and the error signal is fed back to the gyroscope through a PI controller until the drive amplitude locks to the valued specified in the reference (Figure 7). The feedback loop with exclusively the P controller is shown to be asymptotically stable according to the operating point (V_{ref} - **mag** $\dot{X}$) plus an offset proportional to damping and natural frequency. Thus, with only the proportional controller, there is steady state error along the drive direction (Figure 7) which varies with damping. The addition of an integral controller eliminates this steady state error and successfully locked the amplitude to the desired value.

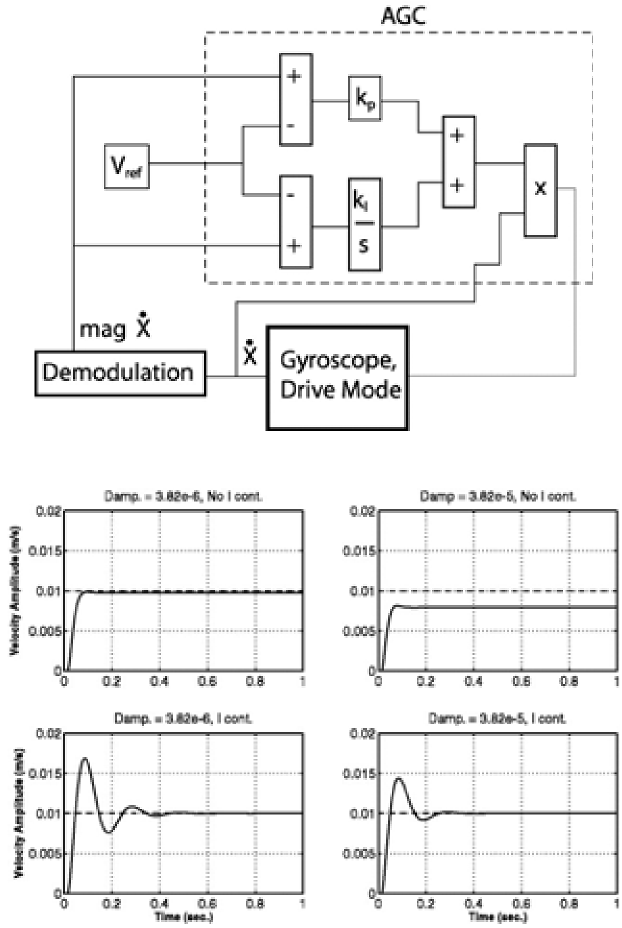

Figure 7: The AGC control takes an output velocity signal $\dot{x}$, demodulates it to obtain amplitude **mag** $\dot{X}$, and drives this to a defined value V_{ref} using a PI control. With only the P control, the amplitude **mag** $\dot{X}$ will have a steady state error proportional to damping. With both controls, the amplitude locks to the desired value.

6 CONCLUSION

In this paper, we have introduced a gyroscope concept that is inherently robust to fabrication imperfections and environmental variations due to its decoupled dual mass sense structure. By designing lower eigenfrequencies on the 5 kHz range, the gyroscope can be made resilient to accelerations on the order 100g while still preserving the gain advantage over conventional single mass gyroscope designs. Finally, the use of a single degree of freedom drive mode allows the use of AGC control to preserve the oscillation amplitude.

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