Pis'ma v ZhETF, vol. 109, iss. 4, pp. 254 – 255 © 2019 February 25

On-chip piezoelectric actuation of nanomechanical resonators containing a two-dimensional electron gas

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Submitted 5 December 2018
Resubmitted 7 December 2018
Accepted 7 December 2018

DOI: 10.1134/S0370274X19040106

Micro- and nanomechanical resonators provide a versatile platform for creation of various sensors [1] and for fundamental studies [2]. Their downsizing can be desirable, but it often complicates the actuation. Among the piezoelectric materials used for fabrication of nanomechanical resonators, especially interesting are the AlGaN/GaN and AlGaAs/GaAs [3–5] heterostructures containing a two-dimensional electron gas (2DEG). These systems are integrable with high-electron-mobility transistors within a single chip. Moreover, at low temperatures, the coupling between mechanical motion and electron transport makes it possible to study the transport phenomena under unusual conditions and to trace electron-transport processes by precise measuring the change in the resonant frequency [4], thus widening the range of existing experimental techniques. Previously, the piezoelectric actuation has been successfully used [4, 5] for driving the flexural vibrations of micromechanical AlGaAs/GaAs-based resonators with a 2DEG. In the present paper, we experimentally show that this technique is suitable for driving flexural and torsional vibrations of smaller (0.166 nm-thick) resonators in the MHz frequency range at room temperature. We also show that, due to the buckling and static bending of axially-loaded thin resonators, their piezoelectric actuation has important features which should be taken into account for the driving to remain effective.

We studied two types of resonators, namely, a cantilever and a bridge shown in Fig. 1a, b. Their total lengths L are 8 and 10 μm, respectively. Thicknesses h of both resonators are 166 nm. The width of the resonators equals $W_0 = 2\,\mu$m near the clampings. During the experiment, the samples were placed in a vacuum chamber with optical access. Amplitude and phase of the vibrations were measured using a Doppler vibrometer combined with a vector signal analyzer. The vibrations were piezoelectrically driven by application of an ac voltage between the top gate and the 2DEG.

Figure 1c, d shows the measured frequency dependences of the amplitude of flexural vibrations. The experimentally measured responsivities $(u_0/V_0)_{\rm exp}$, i.e., the ratios of the resonant amplitude u_0 to the driving voltage V_0 are 790 nm/V for the cantilever and 20 nm/V for the bridge. When V_0 increases, the resonant curves characterizing the bridge are tilted to the left (see the inset in Fig. 1d), indicating softening nonlinearity. This behavior is typical for buckled and bent bridges [3], while initially straight bridges usually demonstrate hardening due to the beam elongation at high amplitudes. Figure 1e, f shows the amplitude and phase corresponding to the second vibrational mode of the cantilever. The presented data confirm that the observed mode is torsional.

The maximal expected amplitude of flexural vibrations u_0 and its ratio to the voltage amplitude V_0 can be estimated as

$$\left(\frac{u_0}{V_0}\right)_{\rm theor} = \alpha\frac{e_{14}Q}{\rho L^2\Omega_0^2}, \qquad (1)$$

where the non-dimensional coefficient α is

$$\alpha = \frac{1}{8}\frac{{\rm d}\Psi}{{\rm d}(x/L)}\bigg|_{x=L_{\rm g}}\bigg/\int_0^1 \frac{W(x)}{W_0}\Psi^2\left(\frac{x}{L}\right){\rm d}\left(\frac{x}{L}\right). \qquad (2)$$

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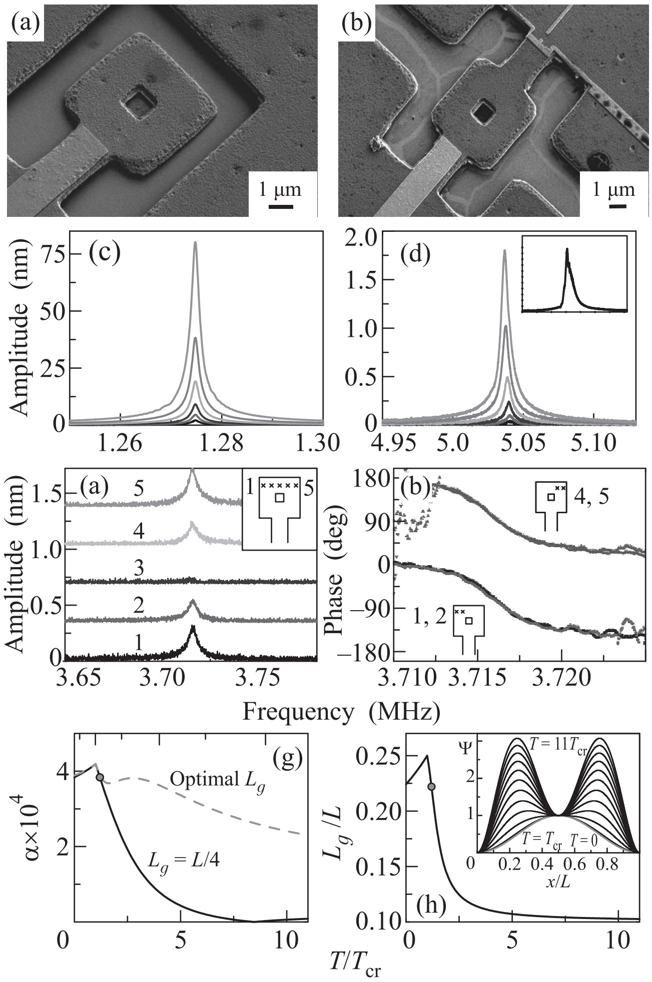

Fig. 1. (Color online) Images of the studied cantilever (a) and bridge (b). Below (c), (d) are the corresponding amplitudes of the piezoelectrically-driven flexural vibrations. At the driving voltage of 400 mV, the doubly-clamped beam demonstrates softening nonlinearity indicating buckling (see the inset in (d)). Amplitude (e) and phase (f) of the cantilever torsional vibrations. (g) – The dependence of coefficient α determining the vibrations amplitude on compression. The solid black curve corresponds to gate length $L_g = L/4$, and the dashed red curve is calculated for optimal $L_g(T)$ values, shown in (h)

Here e_{14} is the piezoelectric constant, Q is the quality factor, L is the resonator length, Ω_0 is the resonant frequency, $\Psi(x)$ is the normalized mode shape and W is the resonator width. These equations give estimates $u_0/V_0 = 1800\,\mathrm{nm\,V^{-1}}$ and $160\,\mathrm{nm\,V^{-1}}$ for cantilever and bridge, respectively. Comparing these values with the experimentally measured responsivities, we obtain the actuation efficiencies 13 % for the bridge and 44 % for the cantilever. Their ratio is close to the resonant frequencies inverse ratio. This speaks in favour of the fact that the efficiency is largely determined by the driving voltage attenuation which should be also proportional to the frequency, rather than by mechanical effects. This attenuation is caused by finite resistance of the contacts and 2DEG, as well as by the distributed parasitic capacitance between the gate and the 2DEG, which tends to equalize their electrical potentials. The obtained efficiencies can be considered as high in comparison to those derived from the existing papers [5].

The buckling of the bridge affects the vibrations. As shown in inset in Fig. 1h, the static bending changes the modal shape $\Psi(\frac{x}{L})$ included in Eq. (2). At a moderate axial load T, the shape of the fundamental mode has one peak at the beam center. According to Eq. (2), the gate ending at the center, where $\frac{d\Psi}{dx} = 0$, cannot drive the vibrations, since the bending moments piezoelectrically induced in the oppositely curved regions cancel each other. At a large T, the modal shape becomes a function with two peaks. If the gate ends at these two additional points, the vibrations are also suppressed. The estimated longitudinal compression in our case is $T \approx 1.2T_{\mathrm{cr}}$. The solid black line Fig. 1g shows coefficient α calculated as a function of T. It can be seen that, at this compression, α is reduced by only 6 % in comparison to the case of a non-compressed beam. Thus, in our case, the buckling influence is small and can be neglected when estimating the actuation efficiency. However, it would be of great importance for long and thin beams which buckle at small critical load T_{cr}. Figure 1h shows the optimal L_{g} maximizing the vibrations. The dashed red line in Fig. 1g shows the corresponding α values. It can be seen that the gate shortening makes it possible to largely eliminate the buckling-induced suppression.

The work is partially supported by Russian Foundation for Basic Research grant # 16-32-60130, State Programme (Grant # 0306-2016-0015) and MEXT Grant-in-Aid for Scientific Research on Innovative Areas "Science of hybrid quantum systems" (Grant # JP15H05869 and JP15K21727). The theoretical part of work is supported by Russian Science Foundation grant # 18-72-10058.

Full text of the paper is published in JETP Letters journal. DOI: 10.1134/S0021364019040052

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