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Measurement of the electrical and mechanical responses of a force transducer against impact forces

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A method for measuring the electrical and mechanical responses of force transducers to impact loads is proposed. The levitation mass method (LMM) is used to generate and measure the reference impact force used. In the LMM, a mass that is levitated using an aerostatic linear bearing (and hence encounters negligible friction) is made to collide with the force transducer under test, and the force acting on the mass is measured using an optical interferometer. The electrical response is evaluated by comparing the output signal of the force transducer with the inertial force of the mass as measured using the optical interferometer. Simultaneously, the mechanical response is evaluated by comparing the displacement of the sensing point of the transducer, which is measured using another optical interferometer, with the inertial force of the mass. To demonstrate the efficiency of the proposed method, the impact responses of a force transducer are accurately determined. © *2006 American Institute of Physics.* [DOI: 10.1063/1.2239035]

I. INTRODUCTION

Force is one of the most basic mechanical quantities and is defined as the product of mass and acceleration. Force transducers are widely used in various industrial and research applications. However, force transducers are typically calibrated by standard static methods using static weights under static conditions. At present, there are no standard methods of evaluating the dynamic characteristics of force transducers. This leads to two major problems in measuring varying force. One is that it is difficult to evaluate the uncertainty of the measured varying force. The other is that it is difficult to evaluate the uncertainty of the moment at which the varying force is measured.

Although methods for dynamic calibration of force transducers are not well established, several attempts have been developed to calibrate transducers against an impact force,[1–3] a step force,[3] and an oscillation force.[4–7] However, it is not clear how such calibrations can be applied to the actual wave profile of a varying force. Validation is therefore needed when applying the frequency response of one type of varying force to other types.

Some of the above methods are based on the "levitation mass method" (LMM) proposed by the author.[1,3,7] In this method, the inertial force of a mass levitated using a pneumatic linear bearing[8] is used as the reference force applied to the object being tested, such as a force transducer[1,3,7] or material.[9–11] Recently, a pendulum mechanism has been developed as a substitute for the aerostatic linear bearing.[12] The inertial force of the levitated mass is measured using an optical interferometer. As for the methods proposed by the author for the dynamic force calibration of force transducers against some typical dynamic forces, such as impact force,[1] step force,[3] and oscillation force,[7] the electric responses of the force transducer against the typical varying forces are evaluated. However, the mechanical behavior of the transducer at the time when the impact load is applied to it, which would yield valuable information for understanding its electrical response, remains unknown.

On the other hand, some methods of analyzing the electric response of force transducers against impact loads have been proposed.[13,14] In these methods, the inertial mass of the part of the transducer itself is considered to be the cause of the difference between the static response and the dynamic response of the transducer. However, no impact force traceable to the International System of units (SI units) is used as the reference force in these methods. Therefore these methods cannot be considered to be dynamic calibration methods for force transducers.

In this article, a novel method is proposed, in which the electrical and mechanical responses of a force transducer against an impact load are simultaneously evaluated. The method is based on the levitation mass method, and the present analysis shows its efficiency.

II. EXPERIMENTAL SETUP

Figure 1 shows a schematic diagram of the experimental setup for measuring the electrical and mechanical responses of the force transducer against the impact load. A conventional S-shaped strain-gauge-type force transducer, whose nominal force is 200 N and which is statically calibrated with the standard uncertainty of approximately 0.4 N (0.2% to full scale), is attached to the base. An aerostatic linear bearing is used to obtain horizontal linear motion with negligible friction acting on a mass (i.e., the moving part of the bearing).

Two optical interferometers, interferometer 1 and interferometer 2, are built to measure the velocity of the mass and the sensing point of the force transducer, respectively. The

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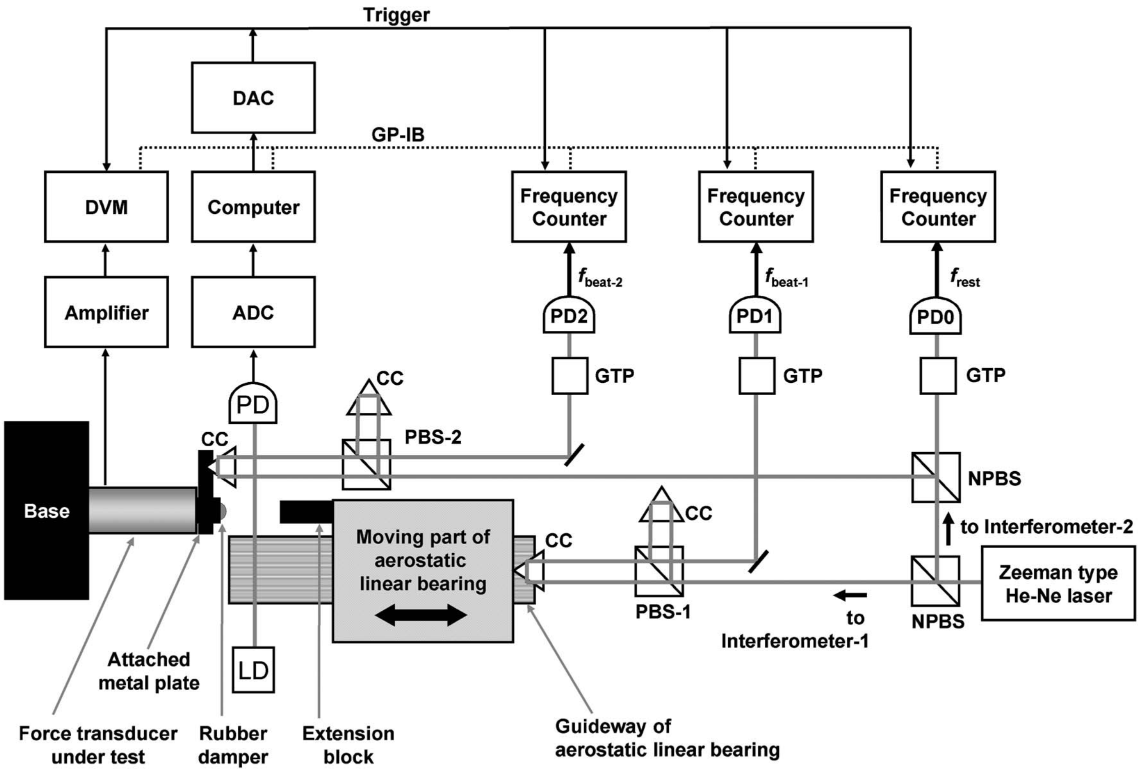

FIG. 1. Experimental setup. Code: CC=cube corner prism, PBS=polarizing beam splitter, NPBS=nonpolarizing beam splitter, GTP=Glan-Thompson prism, PD=photodiode, LD=laser diode, ADC=analog-to-digital converter, DAC=digital-to-analog converter, and PC=personal computer.

mass of the moving part, including a cube corner prism, M_1, is approximately 2.6526 kg. The mass of the metal plate attached to the force transducer including the cube corner prism, M_2, is approximately 0.0816 kg.

A Zeeman-type two-frequency He–Ne laser is used as the light source of the optical interferometers. The interferometers have three photodetectors: PD0, PD1, and PD2. The frequency difference between the two orthogonal polarization states emitted from the laser, f_{rest}, is monitored using a Glan-Thompson prism (GTP) and the first photodetector, PD0.

The total force acting on the moving part of the aerostatic linear bearing, F_{mass}, is calculated as the product of its mass M_1 and its acceleration a as follows:

$$F_{\text{mass}} = M_1 a.$$

In the measurement, the total force acting on the mass, F_{mass}, is considered to be the same as the force acting on the mass from the force transducer being tested, since the frictional force acting on the mass is negligible.[8] The acceleration is calculated from the velocity of the levitated mass. The velocity of the mass (i.e., of the moving part of the aerostatic linear bearing), v_1, is measured as the frequency Doppler shift, $f_{\text{Doppler},1}$, which can be expressed as follows:

$$v_1 = \lambda_{\text{air}}(f_{\text{Doppler},1})/2,$$

$$f_{\text{Doppler},1} = -(f_{\text{beat},1} - f_{\text{rest}}),$$

where λ_{air} is the wavelength of the signal beam under the experimental conditions and $f_{\text{beat},1}$, is the beat frequency, which is the frequency difference between the signal beam and the reference beam and appears as the beat frequency at PD1. In this case, the linear polarization transmitted through the polarizing beam splitter PBS-1, whose frequency is larger than that of the other linear polarization, is used as the signal beam. The positive direction for the velocity, acceleration, and force acting on the moving part is toward the right in Fig. 1. The position of the mass, x_1, the acceleration of the mass, a_1, and the force acting on the mass, F_{mass}, are numerically calculated from the measured velocity.

The velocity of the sensing point of the force transducer, v_2, which is attached to the right side of the transducer under tested, is measured as the Doppler shift frequency $f_{\text{Doppler},2}$ which can be expressed as follows:

$$v_2 = \lambda_{\text{air}}(f_{\text{Doppler},2})/2,$$

$$f_{\text{Doppler},2} = -(f_{\text{beat},2} - f_{\text{rest}}).$$

The beat frequency $f_{\text{beat},2}$ is measured using PD2. The position x_2 and the acceleration a_2 of the actuator are numerically calculated from the velocity v_2.

The frequency $f_{\text{beat},1}$ appearing at PD1 is measured using an electric frequency counter (model: R5363; manufactured by Advantest Corp., Japan). It continuously measures and records the beat frequency $f_{\text{beat},1}$ 1000 times at a sampling interval of $T=400/f_{\text{beat},1}$ and stores the values in its memory. This counter continuously measures the interval time every 400 periods without dead time. The sampling period of the counter is approximately 0.15 ms at a frequency of 2.7 MHz. Two other counters of the same model measure the frequencies f_{rest} and $f_{\text{beat},2}$ appearing at PD0 and PD2, respectively. The counters measure the frequencies without dead time and

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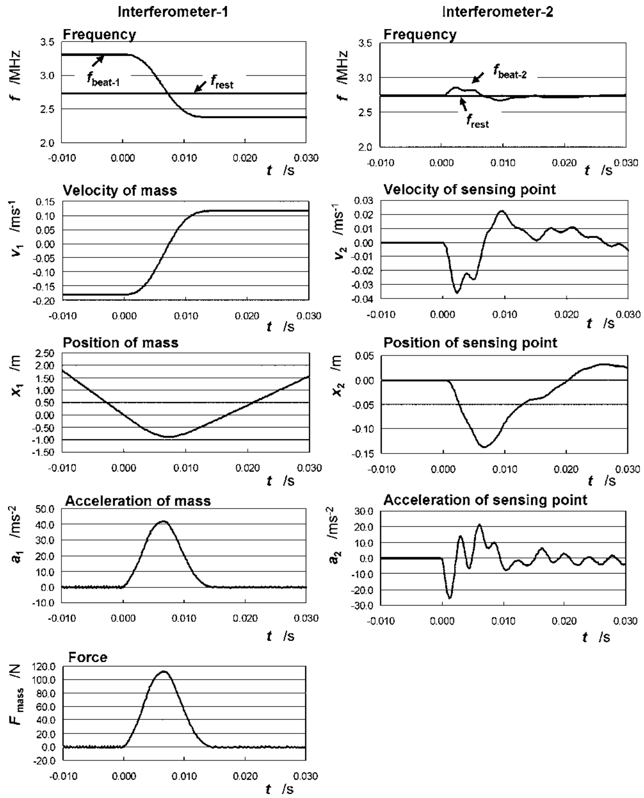

FIG. 2. Data processing procedure: calculation of the velocity, position, acceleration, and force from the measured frequency.

the sampling interval T can be exactly calculated using the measured frequency f and the expression $T=400/f$.

The aerostatic linear bearing, Air-Slide TAAG10A-02 (NTN Co., Ltd., Japan), is attached to an adjustable tilting stage. The maximum weight that the moving part can support is approximately 30 kg, the thickness of the air film is approximately 8 μm, the stiffness of the air film is more than 70 N/μm, and the straightness of the guideway is better than 0.3 μm/100 mm. The frictional characteristics were determined in detail by means of the developed method.[8]

The output signal of the force transducer is measured using a digital voltmeter (model: VP5481L; manufactured by Panasonic Corp., Japan) at a sampling interval of 0.2 ms.

The measurements using the three electric counters (R5363) and the digital voltmeter (VP5481L) are triggered by means of a sharp trigger signal generated using a digital-to-analog converter (DAC). This signal is initiated by means of a light switch, which is a combination of a laser diode and a photodiode. In the experiment, only one collision measurement was conducted and the electric and mechanical responses of the transducers were obtained from the result of the measurement.

III. RESULTS

Figure 2 shows the data processing procedure in the collision experiment. During the collision experiment, the frequencies f_{rest}, $f_{beat,1}$, and $f_{beat,2}$ were measured using photodiodes PD0, PD1, and PD2, respectively. The velocity v_1, position x_1, acceleration a_1, and inertial force $-F_{mass}$ of the moving part of the bearing were calculated from f_{rest} and $f_{beat,1}$, which was measured using interferometer-1. The velocity v_2, position x_2, and acceleration a_2 of the sensing point of the transducer were calculated from f_{rest} and $f_{beat,2}$, which were measured using interferometer 2.

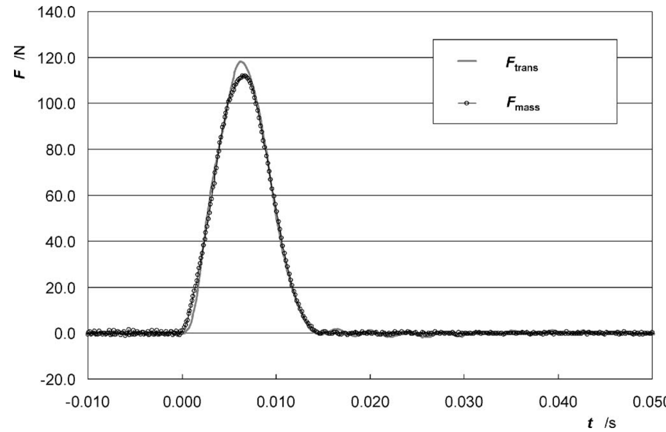

FIG. 3. Force measured by the force transducer and force measured by the proposed method.

Figure 3 shows the force calculated from the output signal of the force transducer and its static calibration results, F_{trans}, and the force acting on the mass, F_{mass}, which is calculated as the product of the mass of the moving part, M_1, and the acceleration of the mass as measured by the proposed method, a_1. The figure shows the electric response of the force transducer under the impact load. The maximum value and the full width at half maximum of the impact force were approximately 112.1 N and 6.9 ms, respectively. The collision period is approximately 14.4 ms. The difference between F_{trans} and F_{mass} derived mainly from the difference between the static characteristics and the dynamic characteristics of the transducer. The root mean square value (rms value) of the difference, $F_{trans}-F_{mass}$, during the collision period was approximately 3.8 N.

Figure 4 shows the relationship between the acceleration of the sensing point, a_2, and the difference between the values measured by the transducer and those measured by the proposed method, $F_{diff}=F_{trans}-F_{mass}$. The regression line, $F_{reg}=0.323a_2$, is also shown in the figure. Figure 4 shows the same result of the single collision experiment as shown in

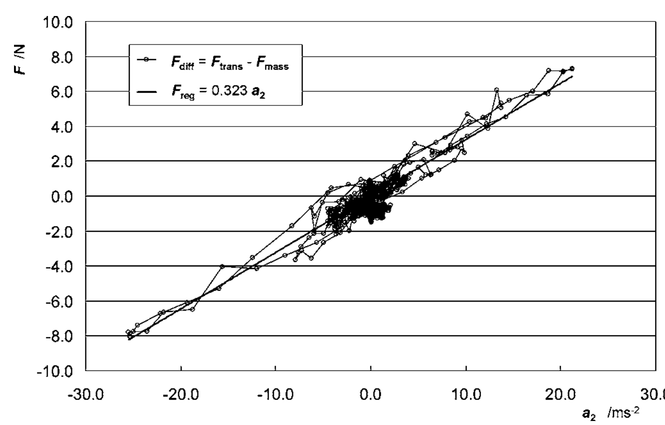

FIG. 4. Relationship between the acceleration of the sensing point and the difference between the values measured by the transducer and those measured by the proposed method.

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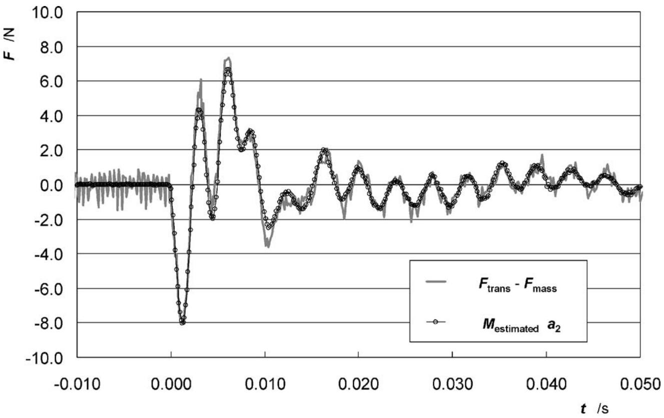

FIG. 5. Difference between the values measured by the transducer and those measured by the proposed method, and the estimated inertial force of the sensor element.

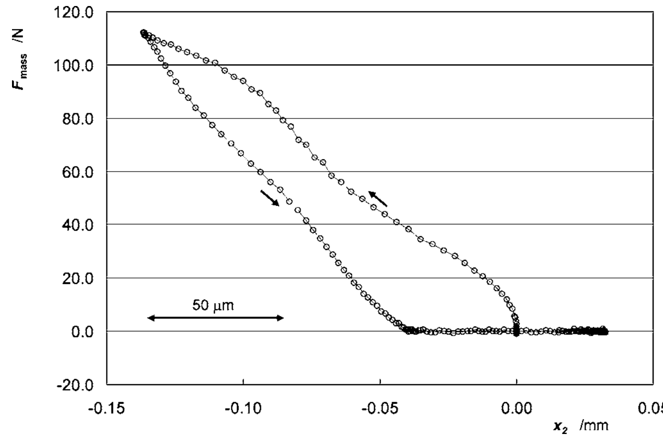

FIG. 6. Mechanical response of the transducer against the impact force.

Figs. 2 and 3 but in different manner. A strong correlation between $F_{diff}=F_{trans}-F_{mass}$ and a_2 was observed from the figure. If the transducer is considered as a mechanical structure consisting of an inertial mass and a spring element, and if the output signal can be considered to represent the deformation of the spring, the inclination of the line, 0.323, can be considered as the estimated effective inertial mass of the transducer, $M_{estimated}$. The value $M_{estimated}-M_2$ was calculated to be 0.241 kg, which corresponds to 57% of the total mass of the transducer itself. Since the transducer is S shaped and its deformation is designed to concentrate around the center of the transducer, the result that estimates the effective inertial mass at about half of the mass of the transducer is reasonable.

Figure 5 shows the difference between the values measured by the transducer and those measured by the proposed method, $F_{trans}-F_{mass}$, and the estimated inertial force of the sensor element, $M_{estimated}a_2$. The two curves coincide well with each other. This indicates that the difference in the electric response of the transducer between the static and dynamic conditions can be well explained as the inertial force of the part of the transducer itself. The rms value of the residual, $F_{trans}-F_{mass}-M_{estimated}a_2$, during the collision period was approximated at 0.7 N.

Figure 6 shows the mechanical response of the transducer against the impact force. The viscoelastic hysteresis, which was caused by the viscosity of the mechanical structure of the transducer, is clearly observed.

IV. EVALUATION OF UNCERTAINTY

In this section, the uncertainties both in measured force and the moment at which the dynamic force is measured are evaluated as follows.

A. Uncertainty of the force measured

The uncertainty components for the determination of the instantaneous value of the force acting on the force transducer, $-F_{mass}$, are as follows:

(1) *Electric counter* (*R5363*). The uncertainty originated from the electric counter R5363 with the sampling interval of $dt=400/f$ (s) is estimated to be approximately 100 Hz. These uncertainties in the beat frequency, $T=400/f_{beat,1}$, and the rest frequency, $T=400/f_{rest}$, correspond to an uncertainty in the velocity of the moving part of approximately 4.5×10^{-5} m/s, according to the expression $v=-\lambda_{air}(f_{rest,1}-f_{rest})/2$. This corresponds to an uncertainty in the acceleration and force of approximately 4.3×10^{-1} ms^{-2} and 1.1 N, respectively.

(2) *Optical alignment*. The major source of uncertainty in the optical alignment is the inclination of the signal beam of 1 mrad, and it results in a relative uncertainty in the velocity of approximately 5×10^{-7}, which is negligible.

(3) *Mass*. Mass of the moving part is calibrated with a standard uncertainty of approximately 0.01 g, which corresponds the relative standard uncertainty in force determination of approximately 4×10^{-6}. This is negligible.

(4) *External force acting on the mass*. For the external force acting on the moving part, the friction force acting inside the pneumatic linear bearing is dominant under the condition that the air film of approximately 8 μm thickness inside the bearing is not broken. The frictional characteristics of the air bearing are determined using the developed method.[8] The dynamic frictional force acting on the moving part, F_{df}, is estimated by

$$F_{df}=Av,$$

$$A=8\times10^{-2}/\mathrm{kg\ s^{-1}}.$$

This is calculated to be approximately 0.02 N at a velocity of approximately 0.2 m s^{-1}, which is negligible.

(5) *Combined standard uncertainty*. Therefore, the standard uncertainty in the determination of the force acting on the force transducer under test is estimated to be 1.1 N. This corresponds to 1.0×10^{-2} (1.0%) of the maximum force applied to the force transducer under test of approximately 1.1×10^2 N.

B. Uncertainty of the moment at which the force is measured

The velocity v_1, position x_1, acceleration a_1, and inertial force $-F_{mass}$ of the moving part of the bearing were calculated from f_{rest} and $f_{beat,1}$. Therefore the shift of the moment at which the varying force is measured is estimated to be less than the sampling interval of the frequency counters of ap-

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proximately 0.15 ms. Therefore, the standard uncertainty in the moment at which the varying force is measured is estimated to be 0.05 ms (50 μs).

V. DISCUSSION

The possible causes for the difference between the static and dynamic characteristics of the output electric signal of strain-gauge-type force transducers are as follows: (1) Inertial mass: the effect of the inertial force of the inertial mass of a part of the transducer, (2) strain gauge: the difference between the static and dynamic characteristics of the strain gauge, (3) elastic body: the difference between the static and dynamic characteristics of the elastic body, and (4) signal processor: the difference between the static and dynamic characteristics of the electric signal processing system.

From the experimental result, it can be said that the effect of the inertial mass was dominant for the case of the S-shaped strain-gauge-type force transducer used in the experiment. The rms value during the collision period, $F_{\mathrm{trans}} - F_{\mathrm{mass}} - M_{\mathrm{estimated}} a_2$, is approximately 0.7 N and it is comparable to the measurement uncertainty of 1.1 N. The simple spring-mass model explains the difference between the static and dynamic characteristics of the output electric signal well. This indicates the possibility of developing a high static and dynamic performance force transducer by combining a strain-gauge-type force transducer with an accelerometer. A strain-gauge-type force transducer can be used to measure static force accurately but its low frequency characteristic is inadequate for dynamic force measurements. The proposed method can be used to evaluate both the electrical and mechanical responses of a force transducer under varying forces and will be applicable for the development of such force transducer.

As shown in Fig. 6, the moving part separated from the transducer at $x_2 = 39$ μm. This residual displacement approaches zero due to the damped oscillation after the impulse. This indicates that the structure is a viscoelastic one and cannot be fully explained as a simple spring-mass model. The mechanical response of a force transducer will be useful for developing an appropriate numerical model of the transducer in its design process.

In this experiment, the uncertainties in measured force and in the moment at which the dynamic force is measured mainly come from the performance of the frequency counters. In general, there is a trade-off between resolution and sampling rate. Recently, a method in which the entire output signal wave form from the optical interferometer is recorded using a high-speed analog-to-digital converter and the frequency is calculated from the recorded wave form by calculating the time interval between the zero-crossing points of the wave form has been developed.[15] Using this method, both the resolution and the sampling rate will be significantly improved.[15]

With the proposed method, the electrical and mechanical responses of force transducers against impact loads can be simultaneously evaluated. This information is necessary for the proper understanding of the electrical response of a transducer to varying load, and it will be useful not only for evaluating the uncertainty of the transducer under dynamic conditions but for developing a method of correcting the output signal of the transducer under varying load.

ACKNOWLEDGMENTS

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