On the basis of these ferrodynamic transducers the Khar'kov KIP plant [4] has developed an equipment which provides automatic monitoring and control of various thermo-technical parameters. A prolonged use of this equipment in an operating metallurgical plant showed a very high reliability of the ferrodynamic transducers as well as of the instruments and regulators with which they operate.

Conclusions. The ferrodynamic transducers compare favorably with differential transformer transducers:

1) by their better linear relationship between the output emf and the angle of rotation (displacement) of the coil, which makes it possible to correct the zero of the indicating instrument without an additional transducer and without introducing any additional errors;

2) by the possibility of changing continuously and without using any contacts (by means of the adjustable plunger), and over a wide range, the transducer transformation ratio (the slope of the characteristic), by means of simple adjustments;

3) by the possibility of obtaining characteristic curves with a parallel displacement with respect to each other;

4) by the small moment required for rotating the moving coil and the absence of a moment of reaction in it;

5) by the stability of the output voltage phase in the effective angle range of the moving coil;

6) by the simple linear kinematic coupling between the transducer coil and the rotating elements of the instrument;

7) by the possibility of using ferrodynamic transducers for adding, subtracting, multiplying and dividing circuits without any rewiring or the use of matching dividers;

8) by the fact that ferrodynamic transducers can fully compensate the effects of transmission lines; the easy setting of a transducer for a left-hand or right-hand rotation makes the direction of rotation of the driving mechanism immaterial.

LITERATURE CITED


DIAPHRAGM VIBRATOR TEST RACK

A. P. Pleshko and V. V. Perfil'ev

Translated from Izmeritel'naya Tekhnika, No. 2, pp. 12-14, February, 1961

Until recently the main obstacle to the wide application of vibration transducers and accelerometers in studying complex mechanico-hydraulic systems under vibration conditions consisted of the lack of a reliable method for testing the transducers' frequency characteristics, for their dynamic calibration, and for determining their sensitivity to side (transverse to the transducer axis) vibrations. The existing vibration test racks provide vibration accelerations not exceeding 30-60 g at frequencies up to 300 cps and only of a few g at frequencies of 2000-3000 cps. Vibration test stands of the tuning-fork type provide vibration accelerations of some 800 g at fixed frequencies in the range of 200-1500 cps.
A further extension of the range of such racks cannot be considered advisable, since for higher frequencies the height \( h \) of the tuning fork arm becomes comparable to its length \( l \). For instance, a tuning fork arm 100 mm long has at 1500 cps a height of 25 mm, for the same length and a tuned frequency of 3 kc its height amounts to 50 mm, and for a frequency of 4.5 kc to 75 mm. It should be noted that for tuning-fork lengths less than 100 mm its oscillating amplitude becomes extremely small, and it becomes increasingly affected by the place of measurement, since the angular displacement of each end has an increasing effect on the oscillations. An attempt to use a tuning fork 65 mm long tuned to 3 kc was only partially successful: the maximum oscillation amplitude with unloaded arms amounted only to 15-20 \( \mu \), which provided an amplitude of vibration accelerations of 600-720 g. Thus, it should be considered that the maximum amplitude obtainable by a tuning-fork vibrator is found at a frequency of 1500 cps.

A theoretical investigation of the possibility of making vibrators for higher frequencies shows that it is necessary to use for this purpose a tuned, balanced mechanical oscillating system of a diaphragm type. This investigation was reduced to finding the natural frequency (first harmonic) of three different mechanical systems, with the parameters determining the frequency being selected the same for the three systems.

The following types of mechanical oscillating systems were considered: a beam with a fixed end (cantilever, tuning-fork vibrator), a beam with two fixed ends (frame vibrator), a round plate with fixed edges (diaphragm vibrator).

The schematics of these systems, formulas for determining their natural frequencies and the ratio of the natural frequency of each system to that of the cantilever beam are given in Fig. 1. From this data it will be seen that in order to obtain large vibration amplitudes at high frequencies it is necessary to use diaphragm type vibrators.

The type of construction we adopted for a diaphragm vibrator is shown in Fig. 2.

<table>
<thead>
<tr>
<th>System type</th>
<th>Natural frequency</th>
<th>Frequency ratio</th>
<th>Type of construction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tuning-fork vibrator</td>
<td>( f_1 )</td>
<td>1</td>
<td><img src="image" alt="Tuning-fork vibrator" /></td>
</tr>
<tr>
<td>Frame vibrator</td>
<td>( f_2 = 0.63 \sqrt{\frac{l}{E}} )</td>
<td>0.63</td>
<td><img src="image" alt="Frame vibrator" /></td>
</tr>
<tr>
<td>Diaphragm vibrator</td>
<td>( f_3 = 12.2 \sqrt{\frac{h}{l}} )</td>
<td>12.2</td>
<td><img src="image" alt="Diaphragm vibrator" /></td>
</tr>
</tbody>
</table>

Fig. 1.

Contrary to a frame and cantilever vibrators with electromagnetic excitation, the excitation system of a diaphragm vibrator was made in a single unit with the core, and suspended in such a manner that the natural frequency of the suspension was a small fraction of the operating vibration frequency.

Another essential difference of diaphragm vibrators consists in the use of four separate damping feet for each vibrator, thus making it possible to stand the vibrator on an ordinary table.

During the assembly and adjustment testing of finished vibrators the following work was carried out:

1. The natural frequency of each half of the diaphragm casing was measured and adjusted to be equal to the other half by varying the thickness \( h \) of one of the diaphragms. Oscillations over a wide frequency range were established by means of the electromagnetic system of ordinary headphones and an audio-frequency oscillator type ZG-2A. The maximum amplitude of the diaphragm oscillations corresponded to its resonance and was determined by means of a piezoelectric probe and a cathode-ray oscilloscope.

2. The conditions of joining and securing the two halves of the vibrator casing along a line of perfect abutment (see Fig. 2) were determined in order to obtain a maximum Q-factor for the system, For this purpose the degree of compression by means of nuts was varied and packing of various thicknesses was inserted. The maximum Q-factor was