POLITECNICO DI TORINO

ESAMI DI STATO PER L'ABILITAZIONE ALLA PROFESSIONE DI INGEGNERE
II SESSIONE - ANNO 1997

Ramo: Elettronica TEMA N. 1

Si vuole realizzare un sistema per la misura della massa di acqua perduta durante un processo di liofilizzazione di composti chimici. Il valore della massa di acqua sarà acquisito dal sistema ad intervalli di tempo prefissati, memorizzato dal sistema ed inviato ad un calcolatore ad esso collegato.
Le specifiche a cui deve soddisfare il sistema sono le seguenti:

- Portata: 200 g
- Massa di acqua perduta: fino al 90% della massa totale
- Incertezza: 1 g
- Campo di temperatura: da – 70 °C a + 20 °C
- Campo di pressione: da 1 Pa a $1 \cdot 10^5$ Pa
- Frequenza di acquisizione dei campioni del misurando: 0,1 Hz
- Tempo di esecuzione dell'intera procedura di misurazione: 48 ore.

Il sistema sarà interfacciato con un Personal Computer, che invierà i comandi di predisposizione all'inizio della misurazione e riceverà tutti i dati acquisiti alla fine di ogni sessione di misura.
Il sistema è costituito da un trasduttore posto nella cella di liofilizzazione, collegato ad un processore situato all'esterno della cella, come indicato schematicamente in fig. 1.
Il trasduttore è composto da un piatto mobile, su cui è posta la massa in misura, vincolato alla base con un sistema di sospensione elastica.
Il valore della massa è determinato misurando la variazione della capacità del condensatore costituito dal piatto mobile, in alluminio, e da un piatto, anch'esso in alluminio, solidale con la base. Il dielettrico è costituito dall'aria residua presente nella cella. Il coefficiente di dilatazione lineare dell'alluminio è $2{,}2 \cdot 10^{-5}$ °C^{-1}.
Si supponga che la struttura meccanica del trasduttore sia già stata progettata secondo lo schema di fig. 2. La sospensione elastica è realizzata con tre molle del tipo 'Flat Spring Steel', con costante elastica ('Spring Rate') pari a 0,15 N/mm. Le altre caratteristiche per lo svolgimento del tema sono ricavabili dalle pagine allegate (H. Neubert, Instrument Transducers, Clarendon Press, 1963, pp. 34-43).
Si richiede il progetto di massima dei circuiti elettronici che compongono il sistema, comprensivo del microprocessore per la gestione della misurazione, l'estrazione dell'informazione di interesse, la memorizzazione dei dati e l'interfacciamento con il Personal Computer.

Suggerimenti per lo svolgimento del tema.

1. E' opportuno innanzi tutto scegliere il metodo di misura che permetta la determinazione della capacità con l'incertezza voluta.
2. Si individuino eventuali procedimenti di correzione sulla base della conoscenza dei valori delle grandezze di influenza significative (si ricordi, ad esempio, che variazioni di temperatura provocano variazioni nelle caratteristiche elastiche delle molle).

3. Si indichi lo schema a blocchi dell'intero sistema, mettendo in evidenza i moduli funzionali che lo compongono.
4. Si disegni lo schema elettrico del circuito che realizza il metodo di misurazione scelto con l'indicazione dei componenti necessari. Se vi sono più possibilità, si preferisca quella di presumibile minor costo.
5. Si disegni lo schema elettrico degli eventuali circuiti ausiliari per la misurazione delle grandezze di influenza.
6. Si scelga il processore che si ritiene più adatto (microprocessore o DSP) e si disegnino i circuiti di interfacciamento fra l'elettronica di pilotaggio del trasduttore ed i canali di ingresso del processore (eventuali convertitori A/D, convertitori frequenza/tensione, ecc.).
7. Si stimi l'incertezza che si prevede di ottenere sulla base dei circuiti e dei componenti scelti. Nel calcolo dell'incertezza è opportuno tenere conto anche di elementi parassiti, come ad esempio la capacità dei cavi di collegamento, ecc.
8. Si scelgano l'hardware ed il protocollo per il collegamento del processore con il Personal Computer.
9. Si determini la procedura per la taratura del sistema di misura e si scriva il sottoprogramma (flow chart e istruzioni in linguaggio C) per l'esecuzione automatizzata della taratura.

Domanda aggiuntiva:
Si indichi il flow chart della procedura per l'invio dei comandi di predisposizione del sistema di misura da parte del Personal Computer.

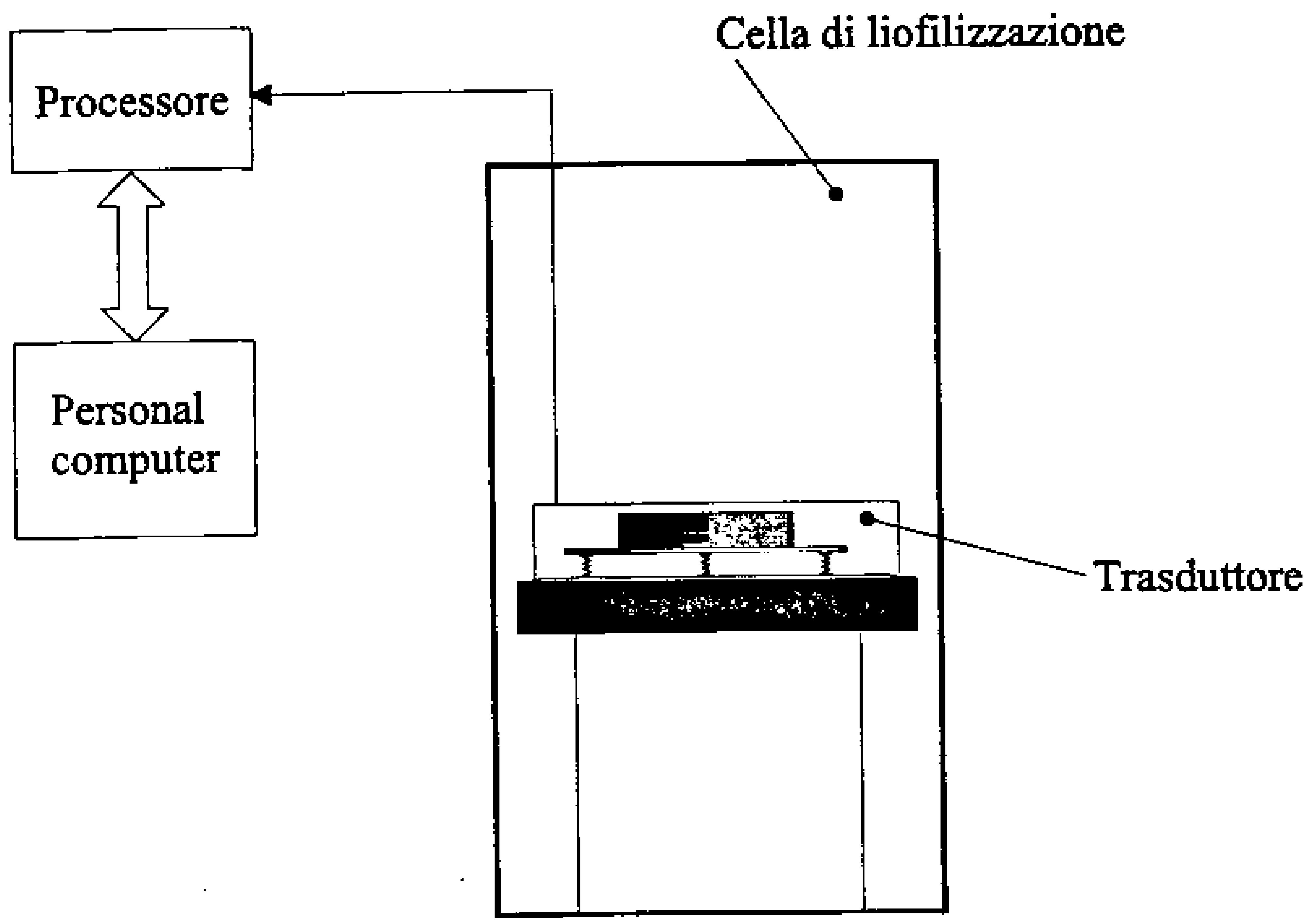
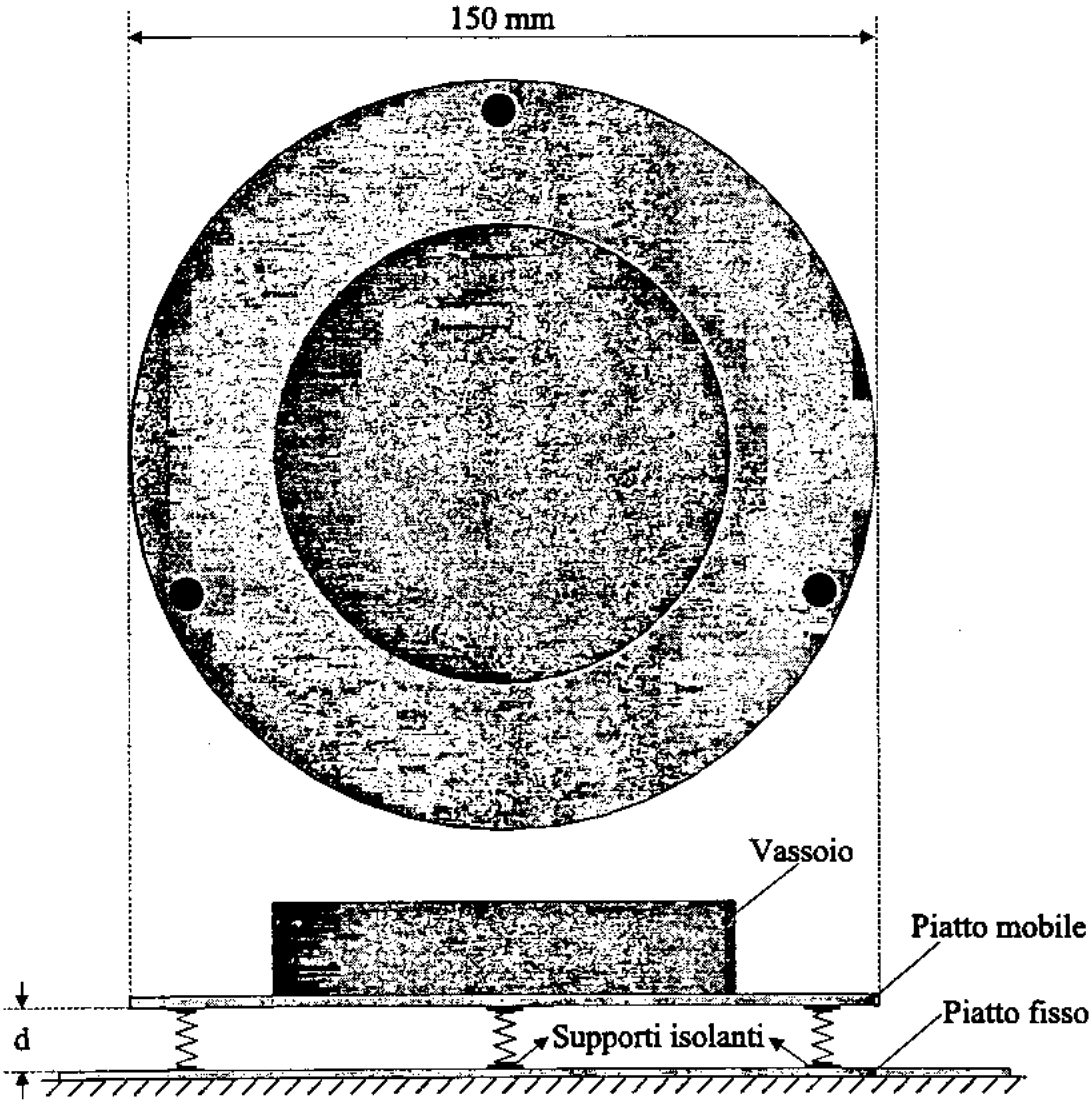


Fig. 1



d = 12 mm a vuoto (piatto mobile e vassoio vuoto) e alla temperatura di riferimento di 20 °C
massa totale a vuoto (piatto mobile + vassoio): 260 g

Fig. 2

An angular acceleration transducer employing the rotary inertia of a liquid will be described in section 4.232.2 on unbonded strain gauge type transducers (Fig. 4.2.32). A liquid-damped angular acceleration transducer of this type is subject to the same effects as a rectilinear acceleration transducer due to variation of fluid's density and the velocity of the moving fluid.

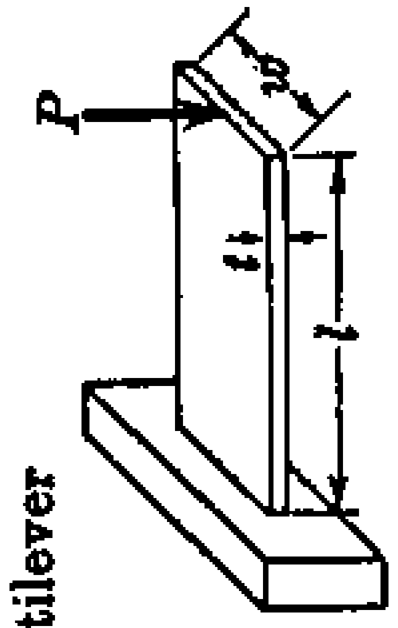
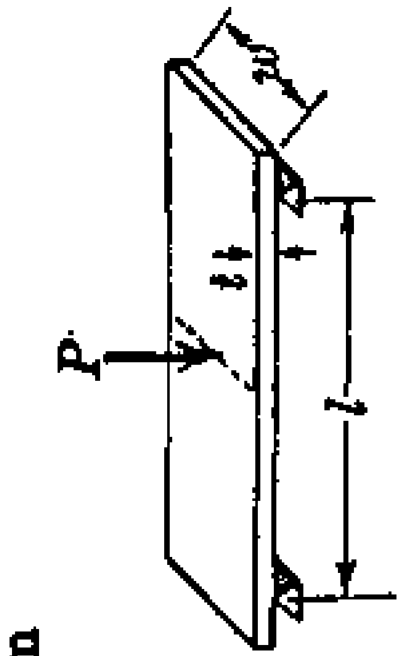
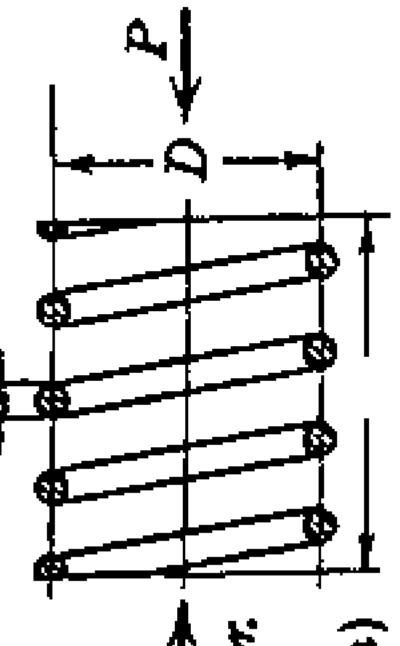
3.32. *Springs*

3.321. *Spring design.* An instrument spring is more often than not the heart of the electro-mechanical transducer and thus warrants the greatest care and consideration in design and manufacture. Frequently the primary sensing elements, such as pressure capsules or diaphragms, are simultaneously the seat of the restoring force, and their spring characteristics have to conform with the characteristics discussed in this section. Even in the design of force-balance type transducers (Chapter 5) which theoretically do not require mechanical springs, they are very much in the mind of the designer since guide elements for the moving parts are invariably required and these often constitute an undesired mechanical spring stiffness. This also applies to electrical connexions between the moving and stationary parts of this type of transducer.

The relative movements in instrument transducers are usually small and engineering approximations for small deflexions of springs are therefore applicable. Fig. 3.11 summarizes well-known design formulae for basic spring configurations with linear deflexions, and Fig. 3.12 those for angular deflexions. The dimensions are in English engineering units for convenience. E and G are the moduli of elasticity (Young's modulus) and of rigidity, respectively, both measured in lb/sq. in. Fig. 3.13 shows[24] the stress factor for helical springs as a function of the ratio D/d. Values of E and G for various spring materials are listed in Table II in the following section 3.322 on spring materials.

Fig. 3.14 shows examples of spring guides commonly used in instrument transducers, which are composed either of leaf springs[25] or of spring diaphragms.[26] As to the latter it is obvious that the multiple-start spiral cut produces a more flexible guide than the single spiral, and that the unwanted rotation of the centre is smaller. The diaphragms with segment and cantilever cuts are free from rotation, but the former has a very restricted deflexion and soon becomes non-linear. The diaphragm with the cantilever cut makes use of the second of the two

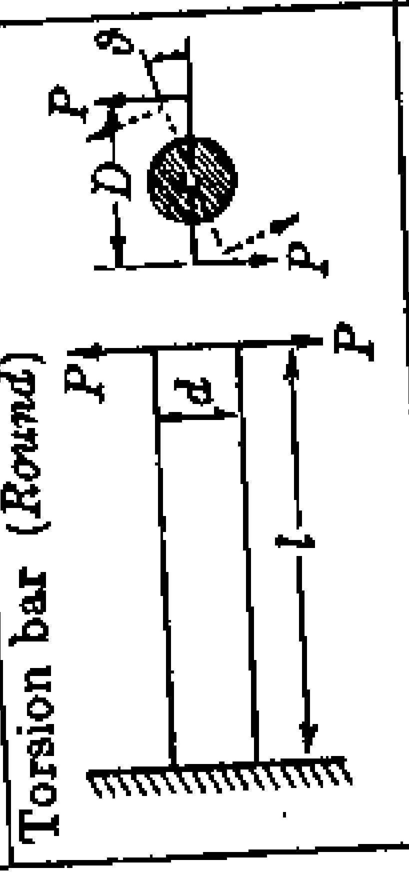
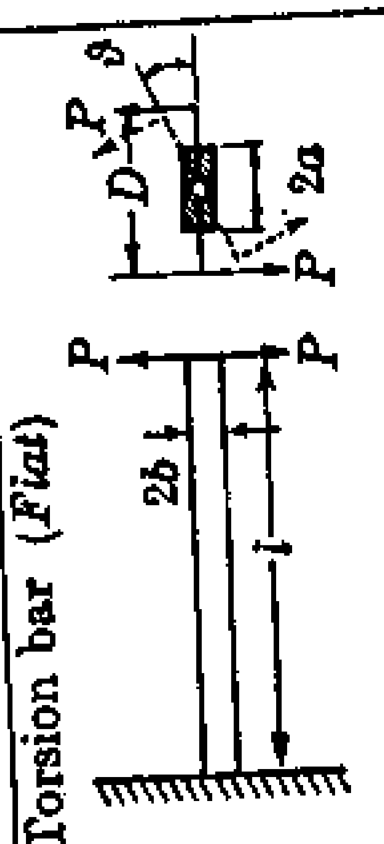
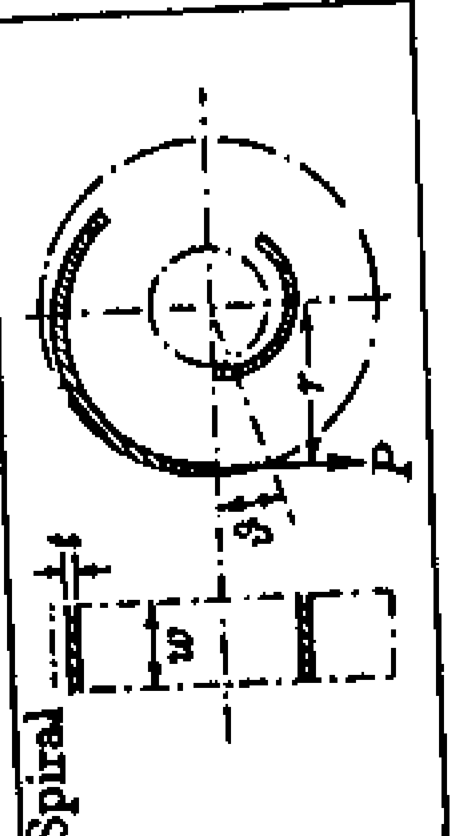
BASIC LINEAR DEFLEXION SPRINGS

Shape of spring and mode of loading	Deflexion (*in.*)	Maximum Stress (*lb/sq. in.*)	Spring rate (*lb/in.*)
Cantilever 	$f = \frac{4Pl^3}{Ewt^3} = \frac{2\sigma l^2}{3Et}$	$\sigma = \frac{6Pl}{wt^2} = \frac{3Etf}{2l^2}$	$\frac{P}{f} = \frac{Ewt^3}{4l^3}$
Beam 	$f = \frac{Pl^3}{4Ewt^3} = \frac{\sigma l^2}{6Et}$	$\sigma = \frac{3Pl}{2wt^2} = \frac{6Etf}{l^2}$	$\frac{P}{f} = \frac{4Ewt^3}{l^3}$
Helical  (Compr. or tension)	$f = \frac{8PnD^3}{Gd^4} = \frac{\pi nD^2\tau}{GdK_1}$	$\tau = \frac{8K_1DP}{\pi d^3} = \frac{Gdf\,K_1}{\pi nD^2}$	$\frac{P}{f} = \frac{Gd^4}{8nD^3}$

n = number of turns; K_1 = stress factor (see Fig. 3.13)

Fig. 3.11. Design of basic linear deflexion springs.

BASIC ANGULAR DEFLEXION SPRINGS

Shape of spring and mode of loading	Angular deflexion (*rad*)	Maximum stress (*lb/sq. in.*)	Angular spring rate (*lb in./rad*)
Torsion bar (*Round*) 	$\vartheta = \frac{32PDl}{\pi d^4 G} = \frac{2\tau l}{dG}$	$\tau = \frac{16PD}{\pi d^3} = \frac{dG\vartheta}{2l}$	$\frac{PD}{\vartheta} = \frac{\pi d^4 G}{32l}$
Torsion bar (*Flat*) 	$\vartheta = \frac{PDl/Gab^3}{\frac{16}{3}-3\cdot36(b/a)\{1-(b^4/12a^4)\}}$	$\tau = \frac{PD(3a+1\cdot8b)}{8a^2b^2}$	$\frac{PD}{\vartheta} = \frac{Gab^3}{l} \times [\frac{16}{3}-3\cdot36(b/a)\{1-(b^4/12a^4)\}]$
Spiral 	$\vartheta = \frac{12\pi P r^2 n}{Ewt^3} = \frac{2\pi r n \sigma}{Et}$	$\sigma = \frac{6Pr}{wt^2} = \frac{tE\vartheta}{2\pi r n}$	$\frac{Pr}{\vartheta} = \frac{wt^3 E}{12\pi r n}$

n = number of turns

FIG. 3.12. Design of basic angular deflexion springs.

types of parallel spring guides in Fig. 3.14. It has an excellent performance if the diaphragm is perfectly flat and free from initial buckles. The spring diaphragms of Fig. 3.14, and similar forms, can be produced from thin sheet material either by an etching process or, when stacked and clamped between solid end-plates, on an engraving machine. In all leaf springs burrs and deformations must be completely avoided in order to obtain optimum flexibility and linearity.

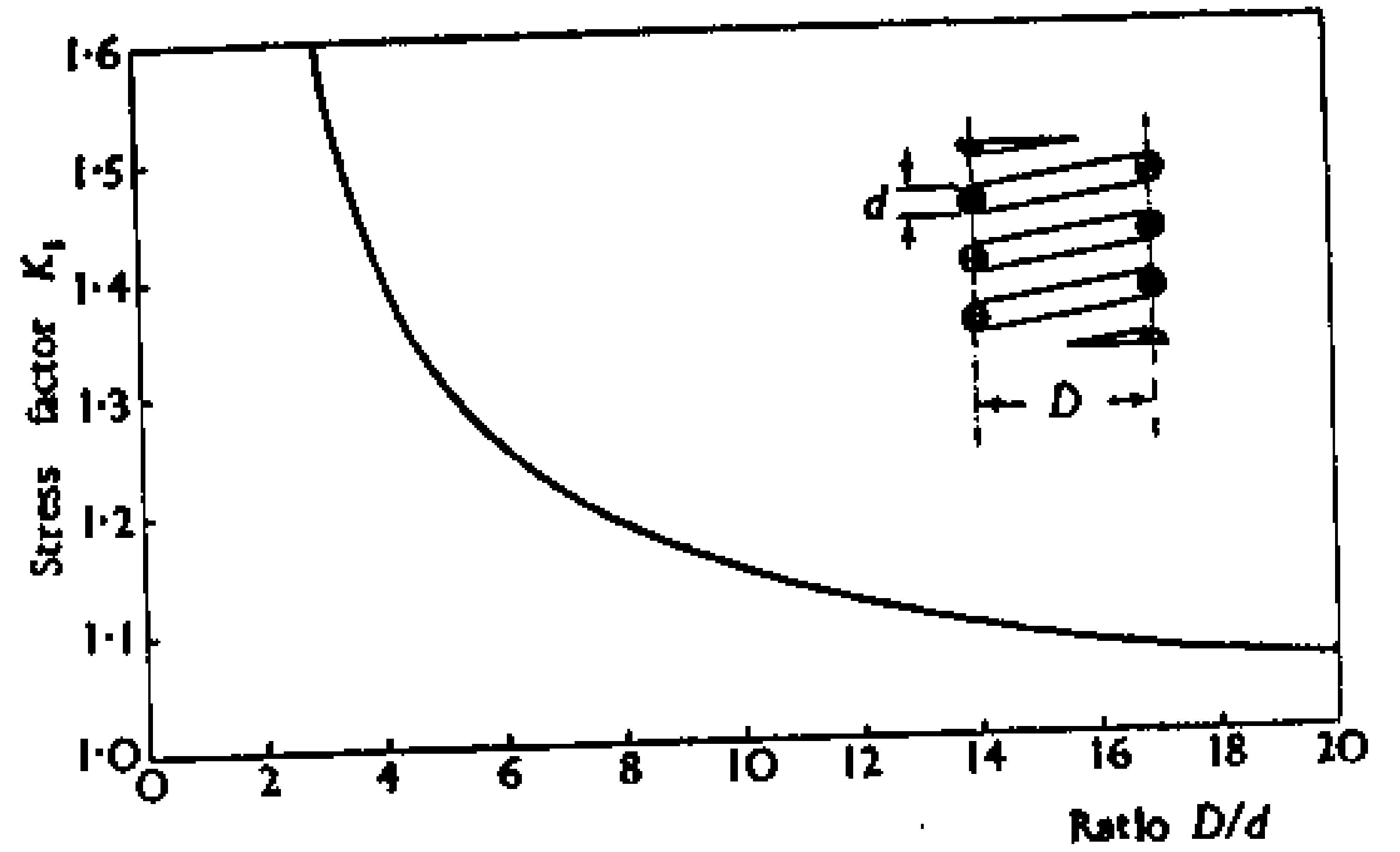


FIG. 3.13. Stress factor K_1 of helical springs as a function of the ratio of mean diameter D to wire diameter d.

Fig. 3.15 shows spring pivots, either consisting of a single-leaf spring (which can also be machined from solid material) or of two pairs of crossed-leaf springs. The crossed spring pivot is widely used in instrument work and its properties have been investigated both theoretically[27,28,29,30] and experimentally[30,31]. There is a second-order displacement of the axis of rotation, especially at large angular deflexions, which can be computed from

$$s = (3\cdot57-4\cdot44l)10^{-4}\vartheta+(5\cdot5-2\cdot4l)10^{-6}\vartheta^2 \qquad (42)$$

where l = free length of spring strip (inches),

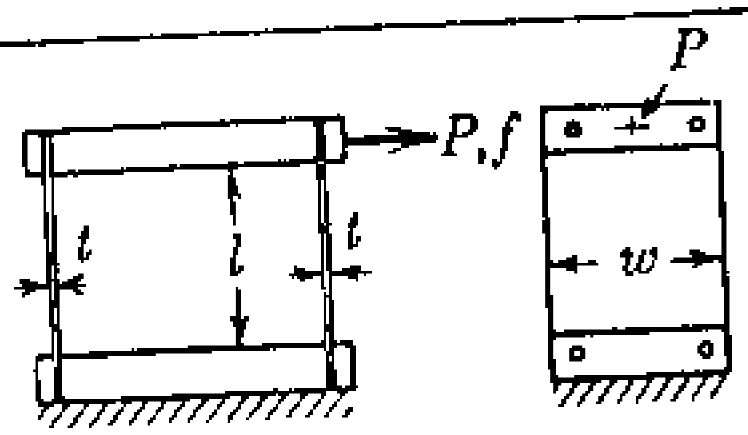
ϑ = angular deflexion (degrees).

At the small angular deflexions commonly used in instrument transducers this error is negligible. As compared with other types of pivots the crossed spring pivot is virtually frictionless.[32,33]

3.322. ***Spring materials.*** An ideal material for instrument springs should have the following properties (see also section 3.1):

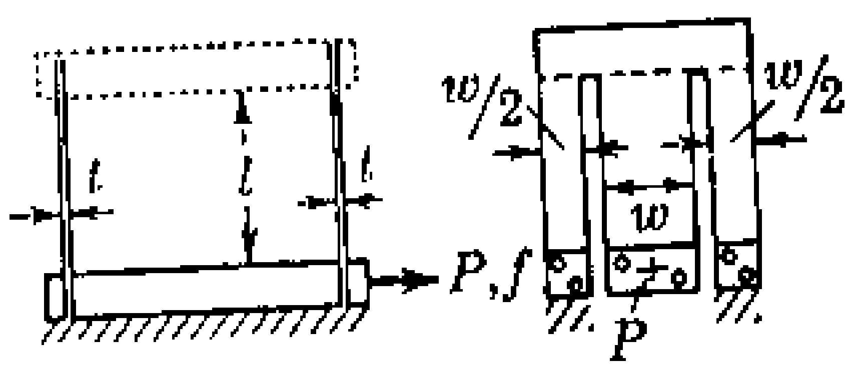
(1) Low values of E and G (moduli of elasticity and rigidity).

PARALLEL SPRING GUIDES



For complete guide (*2 springs*)

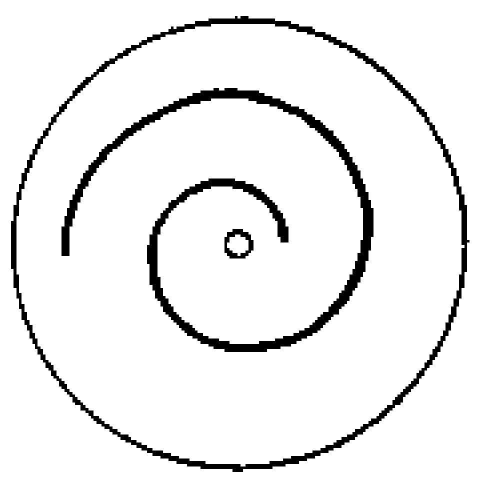
$$\frac{P}{f} = \frac{2Ewt^3}{l^3} \text{ (lb/in.)}$$



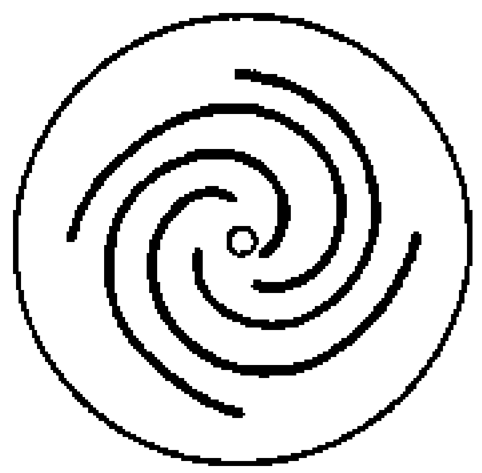
$$\frac{P}{f} = \frac{Ewt^3}{l^3} \text{ (lb/in.)}$$

SPRING DIAPHRAGMS

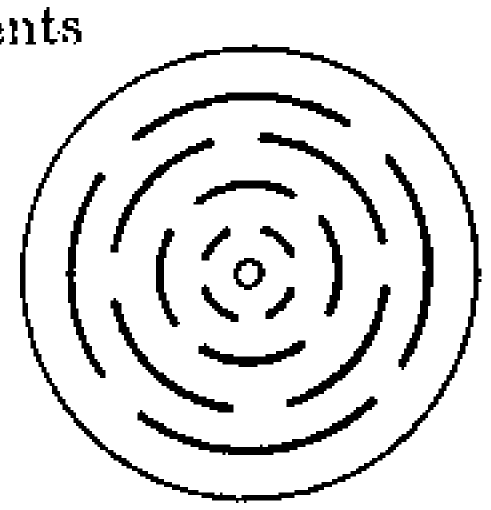
Spiral



Multiple start spiral



Segments



Cantilevers

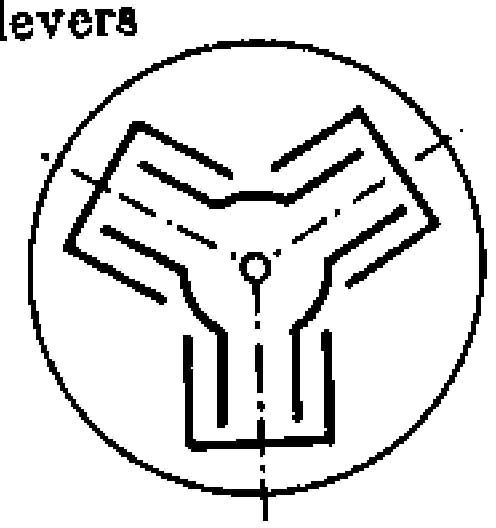
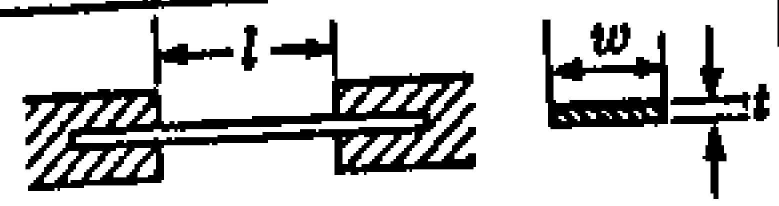
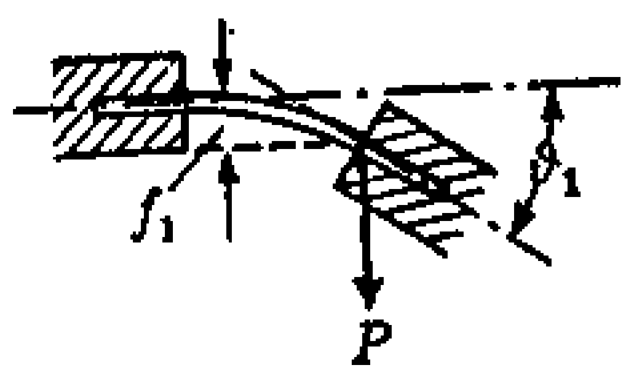


FIG. 3.14. Design of parallel spring guides.



SINGLE SPRING PIVOT

Loaded by a force P

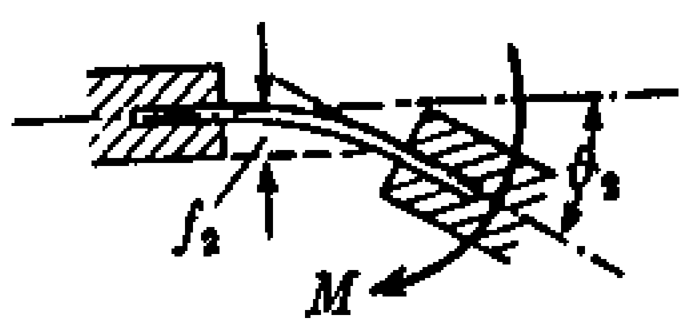


$$\frac{P}{f_1} = \frac{Ewt^3}{4l^3} \text{ (lb/in.)}$$

$$\vartheta_1 \simeq \frac{6Pl^2}{Ewt^3} = \frac{3f_1}{2l} \text{ (rad)}$$

$$f_1 = \frac{4Pl^3}{Ewt^3} \text{ (in.)}$$

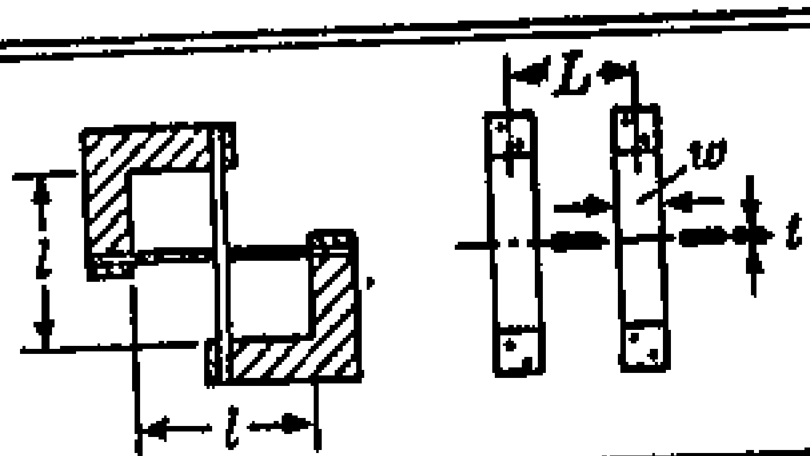
Loaded by a moment M



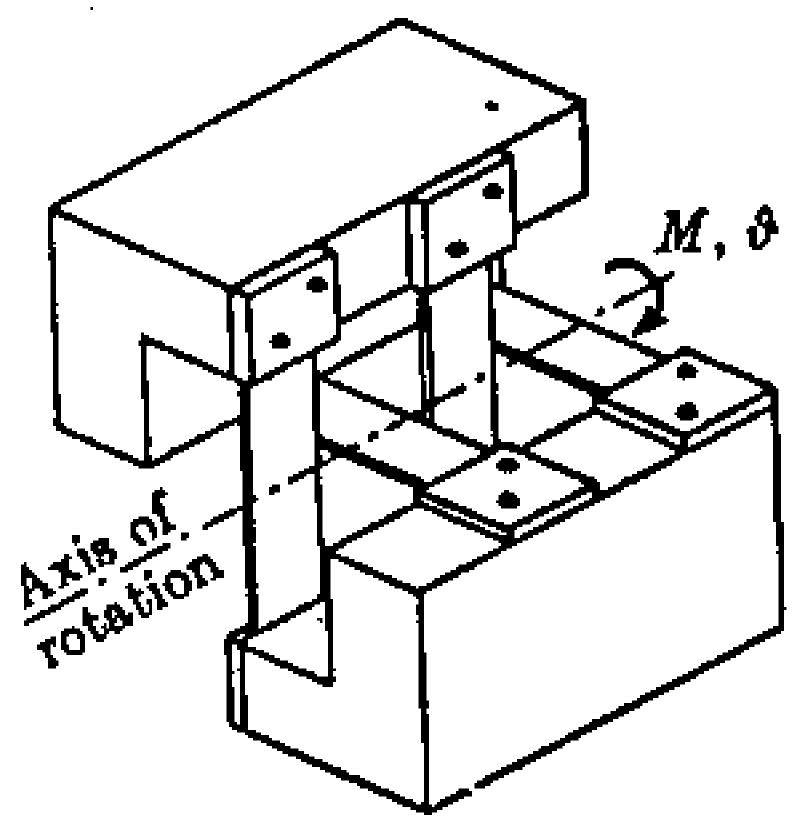
$$\frac{M}{\vartheta_2} \simeq \frac{Ewt^3}{12l} \text{ (lb in./rad)}$$

$$\vartheta_2 \simeq \frac{12Ml}{Ewt^3} = \frac{2f_2}{l} \text{ (rad)}$$

$$f_2 = \frac{6Ml^2}{Ewt^3} \text{ (in.)}$$



CROSS SPRING PIVOT



Angular spring rate of complete pivot:

$$\frac{M}{\vartheta} \simeq \frac{Ewt^3}{3(l+0{\cdot}035)} \text{ (lb in./rad)}$$

Angular deflexion:

$$\vartheta \simeq \frac{3(l+0{\cdot}035)M}{Ewt^3} \text{ (rad)}$$

$\vartheta \leqslant 15°$ (0·25 rad)
$L \geqslant 3w$

FIG. 3.15. Design of spring pivots.

TABLE II

Elastic and thermal properties of some spring materials

Material	*Composition*	*Modulus of elasticity, E* (lb/sq. in.)	*Modulus of rigidity, G* (lb/sq. in.)	*Thermal coeff. of linear expansion, α* (in./in. °C) (0–100° C)	*Thermal coeff. of mod. of elasticity, c* (ΔE/E °C) (−50 to +50° C)	*Thermal coeff. of mod. of rigidity, m* (ΔG/G °C) (−50 to +50° C)	*Remarks*
Multiply by		10^6	10^6	10^{-6}	10^{-6}	10^{-5}	
Flat spring steel	Fe, C, Mn	30	..	11·8	−24	−24	General purpose
Piano wire	Fe, C, Mn	30	11·5–12	11·7	−27	−36	Small helical springs
German silver	Cu, Zn, Ni	16	5·5	16·2	−35	−37	Corrosion resistant
Phosphor bronze	Cu, Sn	15	6·25	17·3	−36	−40	Corrosion resistant, good electrical conductor
Beryllium copper	Cu, Be	16–18·5	6–7	16·6	−35	−33	High fatigue life, low hysteresis, good electrical conductor
Invar	Ni, Fe	21·4	8·1	1·08	+4·8	..	Low thermal expan.
Modulvar	Ni, Fe, C, Mn, Cr, Si	21	8	0	+4·8	..	Low thermal expan.
Elinvar	Ni, Cr, Si, W, Mn, Fe, C	21·5	8·3	0·6	−0·66	−0·72	Constant moduli
Isoelastic	Ni, Cr, Mn, Fe	26	9·2	7·2	−3·6 to +2·7	..	Fairly constant moduli
Ni-span-C	Ni, Cr, Ti, Al, C, Mn	24–27	10	8·1	−1 to +1	..	Constant moduli, high strength

(2) High values of stress at limit of proportionality and of yield strength.
(3) Low values of hysteresis, creep, and elastic after-effect.
(4) High values of endurance and fatigue strength.
(5) Low values of thermal expansion.
(6) Low values of variation with temperature of E and G.
(7) Resistance to corrosion.
(8) Ease of machining and/or joining techniques.
(9) Simple heat treatment or none required after manufacture.
(10) Low and stable values of electrical resistivity.
(11) Ease of electrical connexion to spring by way of clamping or soft soldering.

In all applications there is, of course, the demand for low cost and easy availability in convenient form of spring materials. For example, beryllium copper, one of the best spring materials with respect to (1) to (7) above, falls short of the ideal with respect to (8), (9), and (11), besides being rather expensive. It is therefore obvious that the choice of the most suitable spring material for a particular application will be a compromise between conflicting requirements.

The main reason for inaccuracies in the performance of instrument springs is the variation with temperature of the spring dimensions (coefficient of expansion) and of the moduli of elasticity, E, and rigidity, G. In order to help the designer in his choice of a suitable spring material with respect to temperature stability, Table II has been compiled, though in specific cases more detailed information may be extracted from specialized sources.[34,35,36,37,38] If the basic expressions for the spring rates of different spring configurations are written in a generalized form we have

$$\text{Cantilever:} \quad \frac{P}{f} = \text{const}\,\frac{wt^3E}{l^3}, \tag{43 a}$$

$$\text{Helical:} \quad \frac{P}{f} = \text{const}\,\frac{d^4G}{D^3}, \tag{43 b}$$

$$\text{Torsional:} \quad \frac{M}{\vartheta} = \text{const}\,\frac{D^4G}{l}, \tag{43 c}$$

and the corresponding percentage errors in spring rate can be shown to be[35]

$$\text{Cantilever:} \quad \left[1-\frac{1}{(1+\alpha\Delta T)(1+c\Delta T)}\right]100 \simeq 100\Delta T(\alpha+c), \tag{44 a}$$

Helical: $$\left[1-\frac{1}{(1+\alpha\Delta T)(1+m\Delta T)}\right]100 \simeq 100\Delta T(\alpha+m), \qquad (44\text{b})$$

Torsional: $$\left[1-\frac{1}{(1+\alpha\Delta T)^3(1+m\Delta T)}\right]100 \simeq 100\Delta T(3\alpha+m), \qquad (44\text{c})$$

where ΔT = temperature variation (°C),

α = thermal coefficient of linear expansion,

c = thermal coefficient of modulus of elasticity, E,

m = thermal coefficient of modulus of rigidity, G.

The percentage errors per degree C of spring rate for a number of spring materials, when employed in cantilever, helical-, and torsional-type springs, have been computed from equation (44 (a–c)) and are listed in Table III.[35] In addition to the spring rate errors of equation (44) and Table III, there is a zero-shift error with temperature which is proportional to $(\alpha\Delta T)$. Its magnitude is a function of the length of the spring, its geometry, and the manner in which it is being used.

TABLE III

Percentage errors of spring rate per degree Centigrade of cantilever, helical, and torsional-type springs for various spring materials (−50 to +50° C)

	Cantilever	*Helical*	*Torsional*
Piano wire	0·026	0·025	0·022
Phosphor bronze	0·034	0·038	0·035
Beryllium copper	0·032	0·031	0·028
Ni-span-C	0·0008	0·0008	0·002

3.33. *Damping*

The significance of damping in the dynamic performance of transducer systems has been discussed in section 3.2. The differential equation of a linear second-order system with sinusiodal excitation was (equation (26)):

$$\ddot{z}+\frac{c}{m}\dot{z}+\frac{k}{m}z = \frac{k}{m}s(t) \qquad (45\text{a})$$

$$\ddot{z}+2\delta\dot{z}+(\omega_d^2+\delta^2)z = \frac{k}{m}s(t), \qquad (45\text{b})$$

$$\ddot{z}+2h\omega_0\dot{z}+\omega_0^2 z = \frac{k}{m}s(t), \qquad (45\text{c})$$

$$\frac{m}{k}\ddot{z}+\frac{c}{k}\dot{z}+z = s(t), \qquad (45\text{d})$$

and

$$T_2^2\ddot{z}+T_1\dot{z}+z = s(t) \qquad (45\text{e})$$

where

undamped circular natural frequency: $\omega_0 = \sqrt{(k/m)}$, (46)

damped circular natural frequency: $\omega_d = \sqrt{(\omega_0^2-\delta^2)}$, (47)

ratio: $\omega_d/\omega_0 = \sqrt{(1-h^2)}$ (for $h < 1$), (48)

damping factor: $\delta = c/2m$, (49)

'critical' damping factor: $\delta_k = \omega_0 = \sqrt{(k/m)}$, (50)

damping ratio: $h = \delta/\delta_k = \delta/\omega_0 = c\omega_0/2k = c/2\sqrt{(km)}$, (51)

logarithmic decrement: $\vartheta = 2\pi h/\sqrt{(1-h^2)}$ (for $h < 1$), (52)

time constants: $T_1 = 2h/\omega_0$; $T_2^2 = 1/\omega_0^2$. (53)

The ratio ω_d/ω_0 of the damped natural frequency to the undamped natural frequency (equation (48)), and the logarithmic decrement ϑ (equation (52)) have been plotted in Fig. 3.16 against the damping ratio h, the most commonly used damping notation. At critical damping, h is unity.

In a practical layout of damping devices the designer usually starts by computing the damping force $c\dot{z}$ for a given system, and with the additional information of the undamped circular natural frequency, $\omega_0 = \sqrt{(k/m)}$, the system performance can conveniently be analysed with the parameters ω_0 and h given (see section 3.2). Equations (45) to (53) and Fig. 3.16 will be helpful if alternative parameters have been quoted.

3.331. *Air damping.* The conventional air-damping devices, such as vanes and dashpots, are seldom used in instrument transducers, since they tend to be large and bulky for sufficiently large damping forces. Design formulae and typical arrangements of vane- and piston-type air dampers can be found in textbooks on electrical measuring instruments.[39] However, appreciably larger damping forces can be obtained in a small space, if the air is forced through narrow capillary tubes, or better still, through a porous ceramic plug. Fig. 3.17 shows schematically the latter arrangement in use with acceleration transducers of the unbonded strain-gauge type.[40] The movement of the seismic mass pumps air from the chamber through the porous plug, and if the volume of the air chamber is small, a high air pressure, and thus an effective flow of air in the plug, can be obtained. The temperature stability of this type of damper is better than that of an equivalent oil damper, and the