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# DESIGN AND APPLICATIONS OF SLOTTED CYLINDER SPRINGS

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DESIGN AND APPLICATIONS OF SLOTTED CYLINDER SPRINGS

Wilhelm A. Schneider

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Abstract

A slotted cylinder spring offering unique characteristics of high load capacity and low deflection in extremely small size is discussed in this report. Its use as an elastic element of controllable compliance in seismic transducers is demonstrated and its performance is compared with that of conventional springs. Formulas for these devices are derived and design calculations are given for typical applications.

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## CONTENTS

| | Page |
|---|---|
| ABSTRACT | |
| INTRODUCTION | 1 |
| DISCUSSION | 1 |
| Theoretical Considerations | 1 |
| Numerical Examples for Design and Performance of Slotted Cylinder Springs | 3 |
| Numerical Examples for Comparison of Slotted vs Helical Springs | 4 |
| CONCLUSIONS | 7 |
| ACKNOWLEDGMENTS | 7 |
| REFERENCES | 7 |

TABLES

| | |
|---|---|
| I. Design Parameters and Symbols | 8 |
| II. Design Limitations | 9 |
| III. Calculation Results for Use of Compression Springs | 10 |

FIGURES

| | |
|---|---|
| 1. Slotted Cylinder Spring Complete | 11 |
| 2. Static Diagram of Slot Section | 12 |
| 3. Seismic Resonance Transducer Employing Slotted Cylinder Spring | 13 |
| 4. Load-Deflection Characteristic (Type A) | 14 |
| 5. a. Slotted Cylinder Spring (Cross Section) b. Helical Coil Spring (Cross Section) | 15 |
| 6. Load-Deflection Characteristic (Type B) | 16 |
| 7. Comparison of Slotted Cylinder Spring vs Helical Coil Spring of Same Dimension | 17 |
| 8. Diagram of Calculation Results | 18 |

# DESIGN AND APPLICATIONS OF SLOTTED CYLINDER SPRINGS

## INTRODUCTION

Among the many types and forms of springs, slotted cylinder springs are unique. They can be made extremely small in size for a very wide range of stiffness, whose upper bound exceeds by far all stiffness values which can reasonably be achieved with conventional elastic elements, and which comes close to that of the solid material.

This combination of small size and large stiffness is often required for elastic elements in electromechanical equipments and instruments used by the military and in many fields of industry. For example, a slotted cylinder spring developed by USAELRDL is successfully used as an elastic element in an experimental transducer for the generation of seismic waves.[1,2]

Other applications are currently being investigated experimentally. The range of potential applications for slotted cylinder springs is growing continuously, from simple high strength lock-washer to magnetomechanical clutches and electromechanical transducers.

## DISCUSSION

The basic principle of the elastic element consists in the creation of a spongelike structure realized by cutting slots or slot patterns into the surfaces or circumference of a body of suitable solid material such as metals, plastics or any other solids of suitable hardness and elasticity. Preferred materials are of the same type as used for conventional springs.

A typical implementation of this design concept is a slotted cylindrical tube yielding an elastic element of controllable compliance, Figure 1. It has a number of slots 2′, 2″ and 2‴ provided at one level on the circumference of the cylinder (1). The slot length is limited by vertical path (3) forming slot section (4).

Another slot section (5) is provided by the same number of slots as before, whereas slot section (5) is displaced against slot section (4) in a manner that the vertical path (6) of slot section (5) is located below the center of slots 2′, 2″ and 2‴, arranged in sequence.

Like any other spring, three groups of relations–geometric parameters, material parameters, load and deformation–characterize the design and performance of a slotted cylinder spring. The required parameters and the corresponding symbols are listed in Table I.

Structural stability and reliable operation of the cylinder spring, however, require a limitation of the values of certain design parameters as shown in Table II, where A shows the limits for intermittent operation, while B is quoted for use where continuous oscillating operation is required.

### *Theoretical Considerations*

Within the constraints of the absolute and relative magnitudes of parameters listed in Table II, the slotted cylinder spring can be considered as a parallel series assembly of coupled cantilevered beams as shown in Figure 2. Thus, the approximate load-deflection relation can be derived from the well-known load-deflection formula for a single beam: (See Table I for symbols.)

With $l_s$ the slot length

$$= \frac{D_m \cdot \pi}{n_s} \tag{1}$$

The deflection per slot is

$$f_s = \frac{l_s^3 \cdot P}{16 \cdot b \cdot h^3 \cdot E} \tag{2}$$

Using this relationship yields the deflection per slot section comprising the parallel arrangement of slots

$$f_{ss} = \frac{f_s}{n_s} \tag{3}$$

And the total deflection of $n_{ss}$ serially arranged slot sections:

$$f_{tot} = \frac{n_{ss}}{n_s} f_s \tag{4}$$

Substitution of (2) into (4) yields the total deflection

$$f_{tot} = \frac{n_{ss} \cdot l_s^3 \cdot P}{n_s \cdot 16 \cdot b \cdot h^3 \cdot E} \tag{5}$$

And vice versa, the load

$$P = \left[ 16 \cdot \frac{n_s}{n_{ss}} \cdot \frac{b \cdot h^3 \cdot E}{l_s^3} \right] \cdot f_{tot} \tag{6}$$

The total axial length of the slotted cylinder is

$$L = (n_{ss} + 1) \cdot (h + h_s) + h. \tag{7}$$

By inserting the results of (4) and (6), the static stiffness in Newton/Meter can be accurately calculated by

$$S_o = \frac{P \text{ (In Newton)}}{f_{tot} \text{ (In Meters)}} \tag{8}$$

An accurate determination of the resonance frequencies as expected for use of elastic elements in seismic transducers, for example, depends on the dynamic stiffness $S_o$ [1] and the ratio of the effective masses of the components, and on boundary conditions described in detail.[1,2]

One can see that the properties of the material, its cross-sectional dimensions, and the dimensions and configurations of the slot patterns determine the stiffness which, in turn, controls the resonance frequency of an elastic element.[1,2]

Inaccurate material characteristics and faulty construction can result in deviations from the desired stiffness and the resonance frequency of the element. In such cases, compensations can be made, for example, by locking slot sections to increase the stiffness or by grinding off material to reduce the stiffness to the desired value.

The formulas (1) through (8) are used in the following numerical examples. Their validity has been proved by actual construction and measurement of the load-deflection characteristics of sample springs.

*Numerical Examples for the Design and Performance of Slotted Cylinder Springs*

The following design calculations formed the basis for the fabrication of an elastic element (Figure 3) which is used in the seismic transducer model mentioned in the introduction. The relevant parameters are specified numerically:

| | | |
|---|---|---|
| $D_o = 2.000$ in. | $D_m = 1.875$ in. | $D_i = 1.750$ in. |
| $h_s = 0.031$ in. | $h = 0.094$ in. | $b = 0.125$ in. |
| $n_s = 3$ | $n_{ss} = 60$ | $P = 11$ lb. |
| Material: Steel | $E = 28.4 \cdot 10^6$ psi. | |

Using these numerical values, we obtain from formulas (1) through (8)

$$l_s = \frac{D_m \cdot \pi}{n_s} = \frac{1.875 \cdot 3.14}{3} = 1.9625 \text{ in.} \tag{1}$$

Deflection per slot is expressed as

$$f_s = \frac{l_s^3 \cdot P}{16 \cdot b \cdot h^3 \cdot E}$$

$$= \frac{7.558 \cdot 11}{16 \cdot 0.125 \cdot 8.31 \cdot 10^{-4} \cdot 28.4 \cdot 10^6} = 1.76 \cdot 10^{-3} \text{ in.} \tag{2}$$

Deflection per slot section becomes

$$f_{ss} = \frac{f_s}{n_s} = \frac{1.76 \cdot 10^{-3}}{3} = 0.587 \cdot 10^{-3} \text{ in.} \tag{3}$$

Thus, a total deflection is expressed as

$$f_{tot} = \frac{l_s^3 \cdot P \cdot n_{ss}}{16 \cdot b \cdot h^3 \cdot n_s \cdot E}$$

$$= \frac{7.558 \cdot 11 \cdot 60}{16 \cdot 0.125 \cdot 8.31 \cdot 10^{-4} \cdot 3 \cdot 28.4 \cdot 10^6} = 35.21 \cdot 10^{-3} \text{ in.} \qquad (5)$$

By checking for the given load, we obtain

$$P = \left[ 16 \cdot \frac{n_s}{n_{ss}} \cdot \frac{b \cdot h^3 \cdot E}{l_s^3} \right] \cdot f_{tot}$$

$$= \left[ 16 \cdot \frac{3}{60} \cdot \frac{0.125 \cdot 8.31 \cdot 10^{-4} \cdot 28.4 \cdot 10^6}{7.558} \right] \cdot 35.21 \cdot 10^{-3} = 11 \text{ lb.} \qquad (6)$$

The total length can then be determined as follows:

$$L = (n_{ss} + 1) \cdot (h + h_s) + h$$

$$= (60 + 1) \cdot (0.094 + 0.031) + 0.094$$

$$= 7.719 \text{ in.} \qquad (7)$$

If one inserts the results of (5) and (6), then the static stiffness may be expressed as

$$S_o = \frac{11 \text{ lb.}}{35.21 \cdot 10^{-3} \text{ in.}} \doteq \frac{5\text{kg}}{0.894\text{mm}} = \frac{50 \text{ Newton}}{0.894 \cdot 10^{-3}\text{m}} \qquad (8)$$

$$= 5.6 \cdot 10^4 \text{ Newton/Meter.}$$

A comparison of these numerical data and the resultant load-deflection characteristic with the experimentally measured load-deflection characteristic is given in Figure 6.

Calculated vs measured values of the deflection "f" show only a discrepancy of −3% from the lowest load up to maximum −4.8% to the highest load applied. This discrepancy stems largely from the inaccuracy of the elasticity modulus. Thus, the design formulas are proved to be applicable in practice. As already seen in the calculation, there is one slot section on the lower end and a ring on the upper end of the spring, which is ineffective for deflection (see Figure 1). These dimensions are therefore added to $n_{ss}$ for determination of L (see formula 7). In this way the total length L of the slotted spring can be increased to any desired value. In the case of large $h_s$, the corners of the vertical paths $l_p$ should be rounded as indicated by radius r in Figure 2.

*Numerical Comparison of Slotted Cylinder Spring vs Helical Coil Spring*

The following design calculations serve as a basis for comparison of slotted cylinder springs with helical coil springs in a typical spring application where small size and large stiffness are required. Figure 5a and 5b illustrate the situation. The requirements are specified as follows:

$D_o = 1.020$ in. $\quad D_m = 1.000$ in. $\quad D_i = 0.980$ in.

$d = b = 0.020$ in. $\quad f_{tot} = 0.192$ in. $\quad P = 6.6$ lbs.

$n_s = 3$ $\quad$ Material: Steel $\quad E = 28.4 \cdot 10^6$ psi.

For the slotted cylinder, Formula (1) yields

$$l_s = \frac{D_m \cdot \pi}{n_s} = \frac{1 \cdot 3.14}{3} = 1.047 \text{ in.} \tag{1}$$

In accordance with the constraints of Table 2,

$$h_s \max = \frac{l_s}{16} = \sim 0.065 \text{ in.}$$

For reasons of practical tooling, we chose

$$h_s = 0.064 \text{ and only } 0.6 \cdot h_s \text{ for } f_s$$

$$f_s \text{ therefore } = 0.0384 \text{ in.}$$

and (3)

$$f_{ss} = \frac{f_s}{n_s} = \frac{0.0384}{3} = 0.0128 \text{ in.}$$

The unknown h can now be determined from (2):

$$h = \sqrt[3]{\frac{l_s^3 \cdot P}{16 \cdot f_s \cdot b \cdot E}}$$

$$= \sqrt[3]{\frac{1.1484 \cdot 6.6}{16 \quad 38.4 \quad 10^{-3} \cdot 20 \cdot 10^{-3} \cdot 28.4 \cdot 10^6}} \tag{9}$$

$$= \sqrt[3]{21.72 \cdot 10^{-6}} = 0.0279 \text{ in.}$$

Then by checking, one obtains

$$f_s = \frac{l_s^3 \cdot P}{16 \cdot b \cdot h^3 \cdot E}$$

$$= \frac{1.1484 \cdot 6.6}{16 \cdot 20 \cdot 10^{-3} \cdot 21.72 \cdot 10^{-6} \cdot 28.4 \cdot 10^6} = \sim 0.0384 \text{ in.} \tag{2}$$

$$f_{ss} = \frac{f_s}{n_s} = \frac{0.0384}{3} = 0.0128 \text{ in.} \tag{3}$$

and the number of slot sections becomes

$$n_{ss} = \frac{f_{tot}}{f_{ss}} = \frac{0.192}{0.0128} = 15. \tag{10}$$

In such cases where, for example, $n_{ss} = 14.3$ or $14.8$, this must be corrected to 15 or 14, respectively. If necessary, a slight correction of $h_s$ and recalculation will lead to the exact required value of $f_{tot}$.

Rechecking the total deflection, one obtains

$$f_{tot} \quad \frac{1.1484 \cdot 6.6 \cdot 15}{16 \cdot 20 \cdot 10^{-3} \cdot 21.72 \cdot 10^{-6} \cdot 3 \cdot 28.4 \cdot 10^6}$$

$$= \frac{113.69}{592.17} = \sim 0.192 \text{ in.} \tag{5}$$

This is the exact required value. A small discrepancy of about +1.6%, however, exists between calculated and measured values as shown in Figure 4.

Since, in practice, the modulus of elasticity is usually not known with such accuracy, the first result can be considered good enough for practical design.

The total length of the slotted spring can now be obtained by formula (7) in which $L = (15 + 1) \cdot (0.028 + 0.064) + 0.028 = 1.500$ in. Each end of the slotted spring was extended 0.055 in., or an added extension of 0.110 in., resulting in a total length of 1.610 in.

Figure 7 exhibits very visibly the differences in slotted cylinder springs vs helical springs which have the same diameter and equal wall thickness respective wire diameter ($b = d$). This illustration shows that it is impossible to realize an equivalent helical coil spring because the ratio of coil diameter to wire diameter–"X" in this application, Figure 5b, would require

$$X = \frac{1.000}{.020} = 50.$$

Whereas, standard engineering practice permits only "X" = 10. Thus, the coil spring assembly of Figure 5b is unrealistic, while the slotted cylinder spring assembly of Figure 5a, with the given data, is easily realized.

Table III and Figure 8 show sample results of design calculations for slotted cylinder springs of different sizes and compliances.

The data are essentially within the safety range of the parameters listed in Table II. Within these data ranges, it is possible, of course, to perform many more design variations. A slotted spring for other requirements on loading, deflection, and space can be obtained very simply by changing the dimensions of wall thickness, diameter, horizontal path, and the number of slotsections.

## CONCLUSIONS

Results of experiments have shown that slotted springs exhibit intrinsic qualities which are superior to those of conventional elastic elements. They have been incorporated successfully in mechanical transducers for generation of elastic waves in hard media (seismic waves) at frequencies where conventional elastic elements are too compliant and conventional piezostrictive and magnetostrictive transducer elements are unrealizable because of intolerably large dimension required.

A completely satisfactory design could not be obtained with other elements until the idea of this elastic element of controllable compliance was conceived. The comparison of slotted cylinder springs vs helical coil springs undoubtedly shows the superiority of the slotted spring.

It must be pointed out that no experience has been obtained regarding the thermal elastic behavior and the fatigue patterns of slotted cylinder springs. That is, no spring fatigue has been detected as yet in the seismic transducers where fatigue should manifest itself after a number of hours of operation (nominal 80 cycles per second) under otherwise fixed environmental conditions. More work will be required to establish reliability factors of this kind.

The slotted cylinder springs are more expensive than regular coil springs, with the present fabrication methods, but the former will do jobs which can never be accomplished with conventional coil springs.

May this discussion be an incentive to further considerations of the application of slotted cylinder springs.

## ACKNOWLEDGEMENTS

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1. K. Ikrath, "The Generation and Propagation of Seismic Waves," 1 March 1962, USAELRDL.

2. K. Ikrath and W. Schneider, "New Transducers for Communicating by Seismic Waves," *Electronics* (McGraw-Hill Book Company, Inc., New York, N. Y., 12 April 1963), pp. 51-55.

## TABLE I

## DESIGN PARAMETERS AND SYMBOLS

| | | |
|---|---|---|
| $D_o$ | = | Outside diameter in inches |
| $D_m$ | = | Mean diameter in inches |
| $D_i$ | = | Inside diameter in inches |
| $b$ | = | Wall thickness in inches |
| $l_s$ | = | Length of slot in inches |
| $l_p$ | = | Length of vertical path in inches |
| $h_s$ | = | Height of slot in inches |
| $h$ | = | Height of horizontal path in inches |
| $f_s$ | = | Deflection of one slot in inches |
| $f_{ss}$ | = | Deflection of one slot section in inches |
| $n_s$ | = | Number of slots per section |
| $n_{ss}$ | = | Number of slot sections |
| $f_{tot}$ | = | Total deflection at "P" lbs. in inches |
| $P$ | = | Total compression load in lbs. |
| $L$ | = | Total length in inches |
| $E$ | = | Young's modulus of elasticity in psi |
| $S_o$ | = | Static stiffness |

TABLE II

DESIGN LIMITATIONS

| CASE = | A | B |
|---|---|---|
| $n_s$ = | 3 OR MORE | |
| $h_{s\,max}$ = | $< \frac{l_s}{16}$ | $< \frac{l_s}{50}$ |
| $f_{s\,max}$ = | $0.8 \cdot h_s$ | $0.3 \cdot h_s$ |
| $f_{ss}$ = | $\frac{f_{ss}}{n_s}$ | |
| $l_p$ = | $\frac{h + h_s}{1.6} \pm 5\%$ | |

* NOTE: Structural stability and safety of the spring limit, the maximum tolerable deflection $f_s$ in case A to 80% of $h_s$ and in case B to 30% of $h_s$.

## TABLE. III.

| MATERIAL: STEEL $E = 28.4 \cdot 10^6$ P.S.I. | | | | |
|---|---|---|---|---|
| $f_{tot} = .063"$; $b = .020"$; $n_s = 3$. | | | | |
| $D_m$ | .500" | 1.000 | 1.500 | 2.000 |
| $D_o$ | .520 | 1.020 | 1.520 | 2.020 |
| $D_i$ | .480 | .980 | 1.480 | 1.980 |
| $l_s$ | $.523_6$ | $1.047_2$ | $1.570_8$ | $2.094_4$ |
| $l_s^3$ | $.143_2$ | $1.148_4$ | $3.875_8$ | 9.187 |
| $h_s$ max | $.032_7$ | $.065_5$ | $.098_1$ | .131 |
| $h_s$ CHOSEN | .031 | .047 | .063 | .079 |
| $f_s$ | .025 | .037 | .050 | .063 |
| $f_{ss}$ | .008 | .012 | .017 | .021 |
| $n_{ss}$ | 7.9 (8) | 5.2 (5) | 3.7 (4) | 3 |

RESULTANT DIMENSIONS OF "h" AND "L" ARE GIVEN IN CURVES FIG. 8 AT DIFFERENT COMPRESSION LOAD.

CALCULATION RESULTS FOR USE OF COMPRESSION SPRINGS.

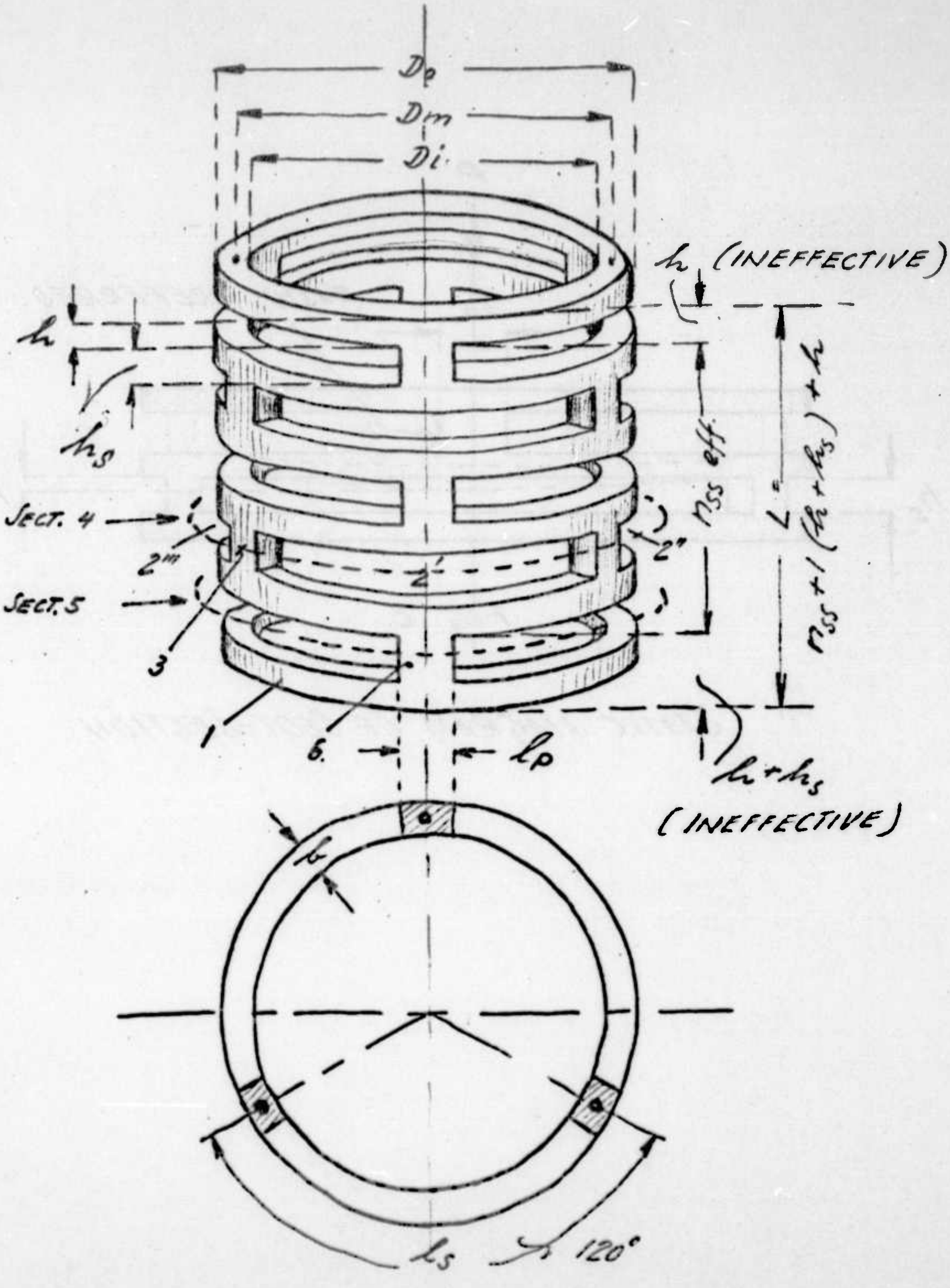



FIG. I

SLOTTED CYLINDER SPRING COMPLETE.

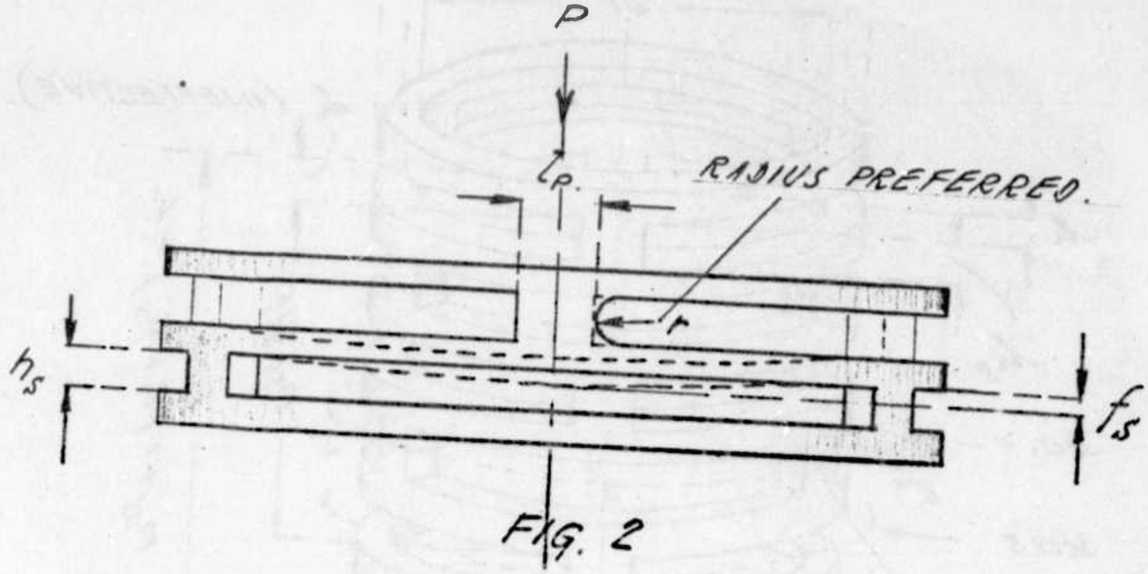
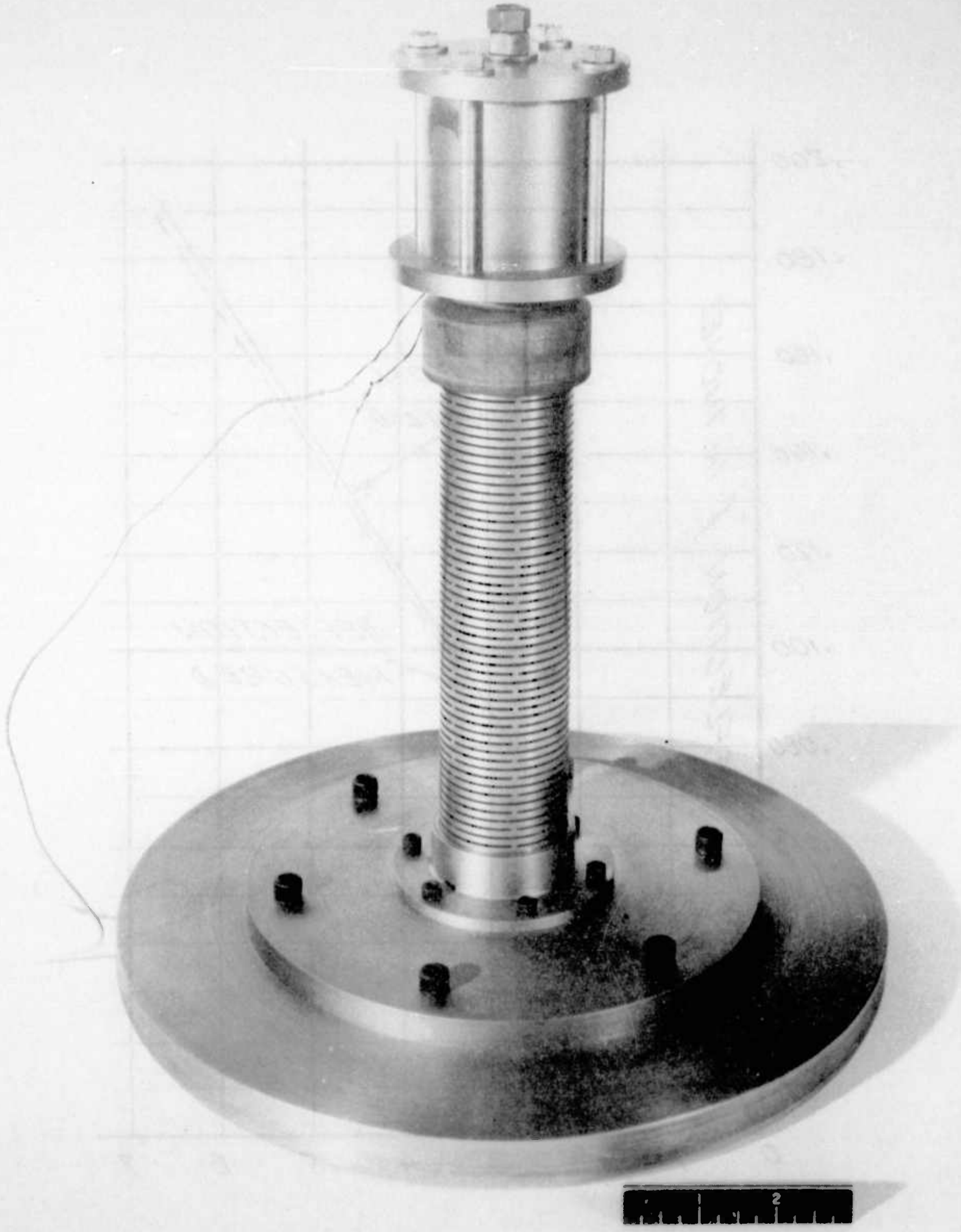



FIG. 2

STATIC DIAGRAM OF SLOT-SECTION.



SEISMIC RESONANCE TRANSDUCER
EMPLOYING SLOTTED CYLINDER SPRING

FIG. 3

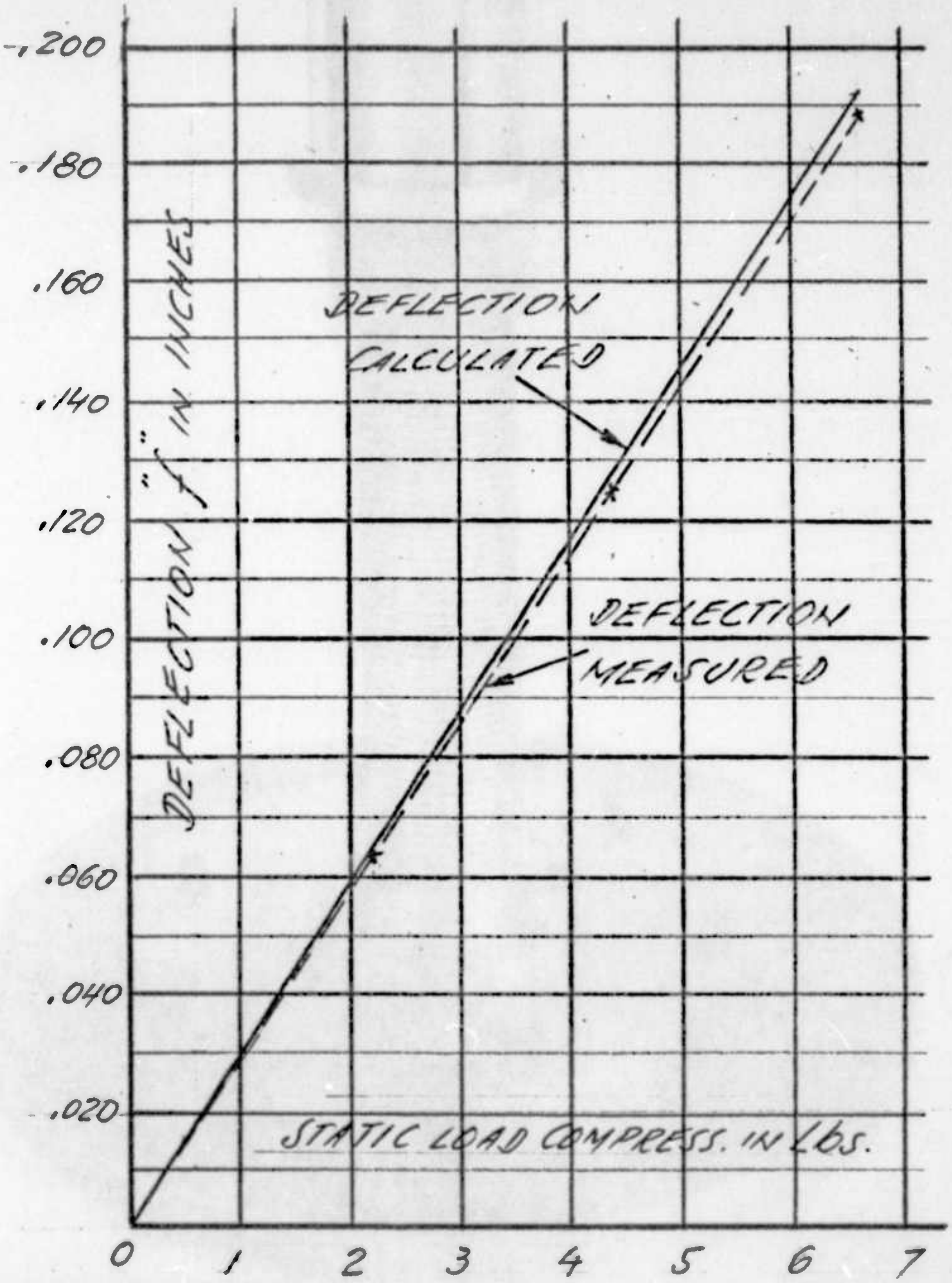



FIG. 4
LOAD-DEFLECTION CHARACTERISTIC
( TYPE A )

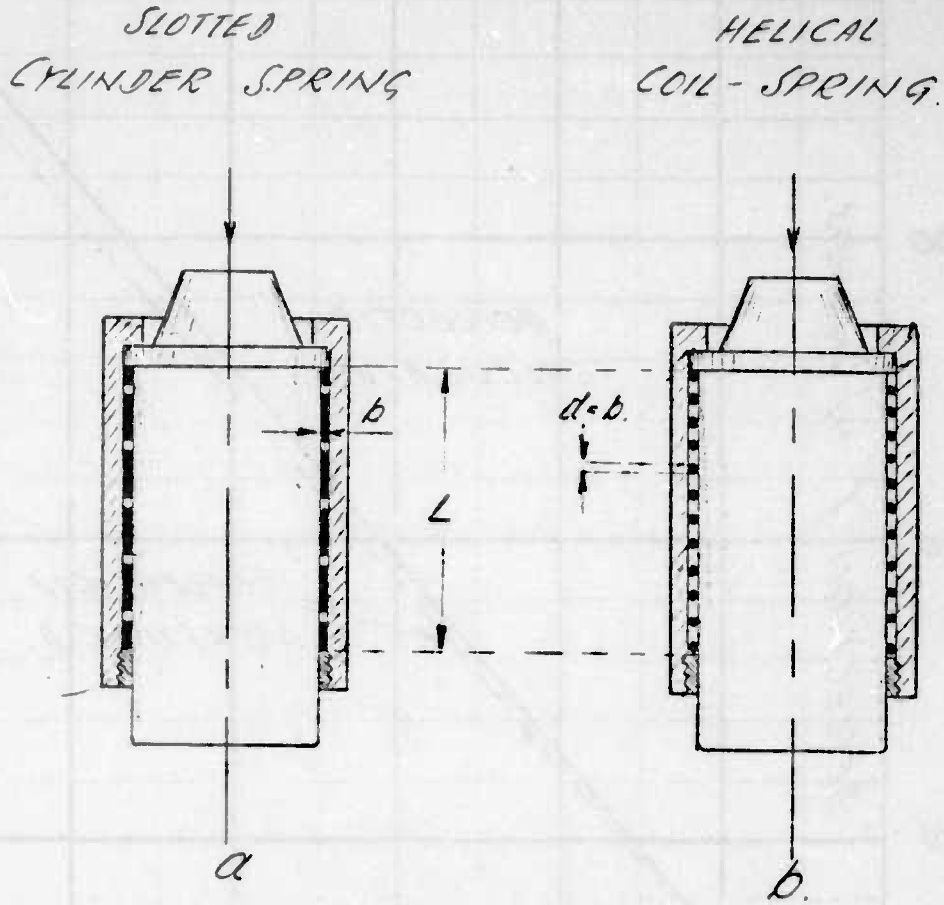



FIG. 5

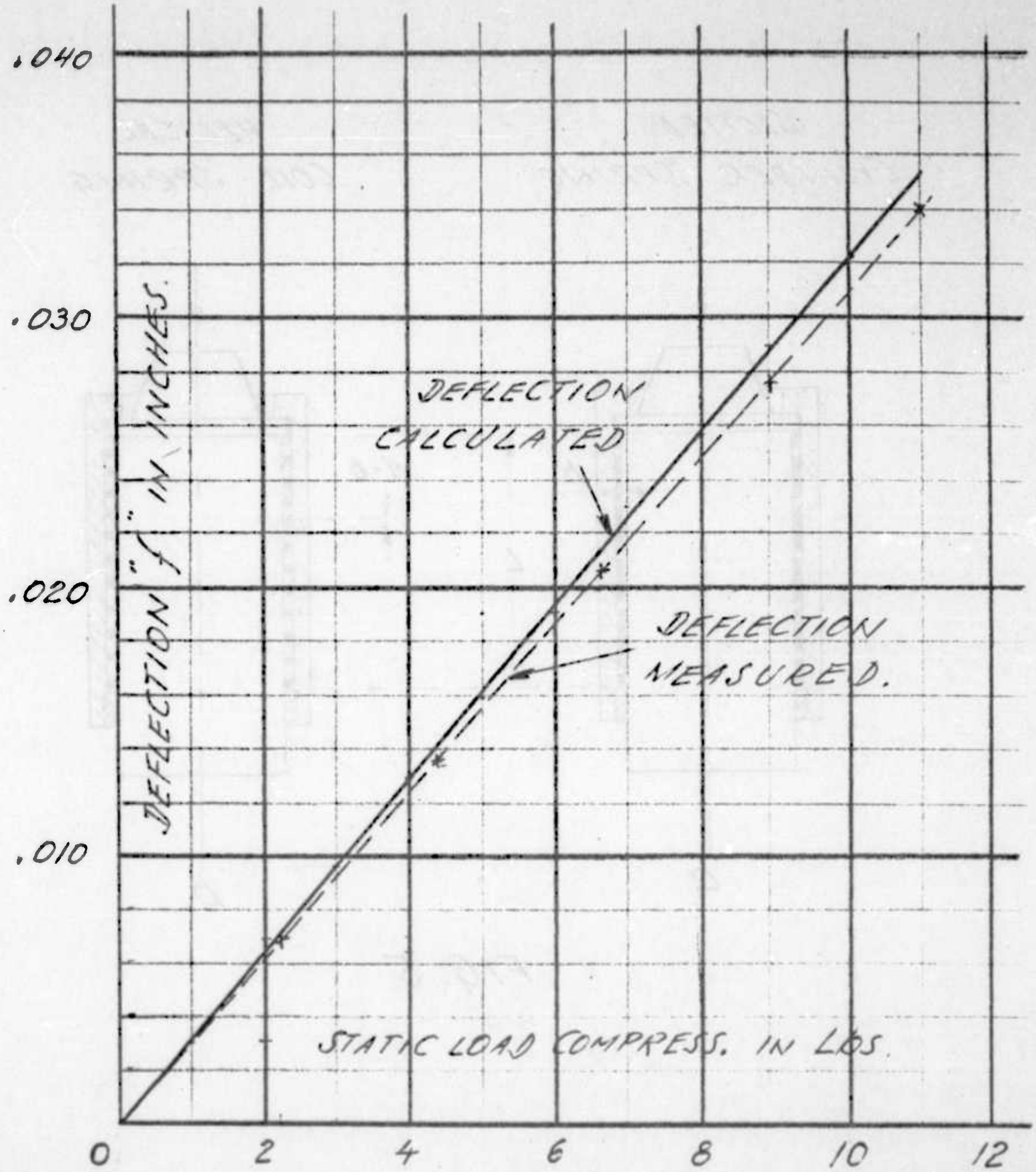



FIG. 6

LOAD-DEFLECTION CHARACTERISTIC

(TYPE B.)

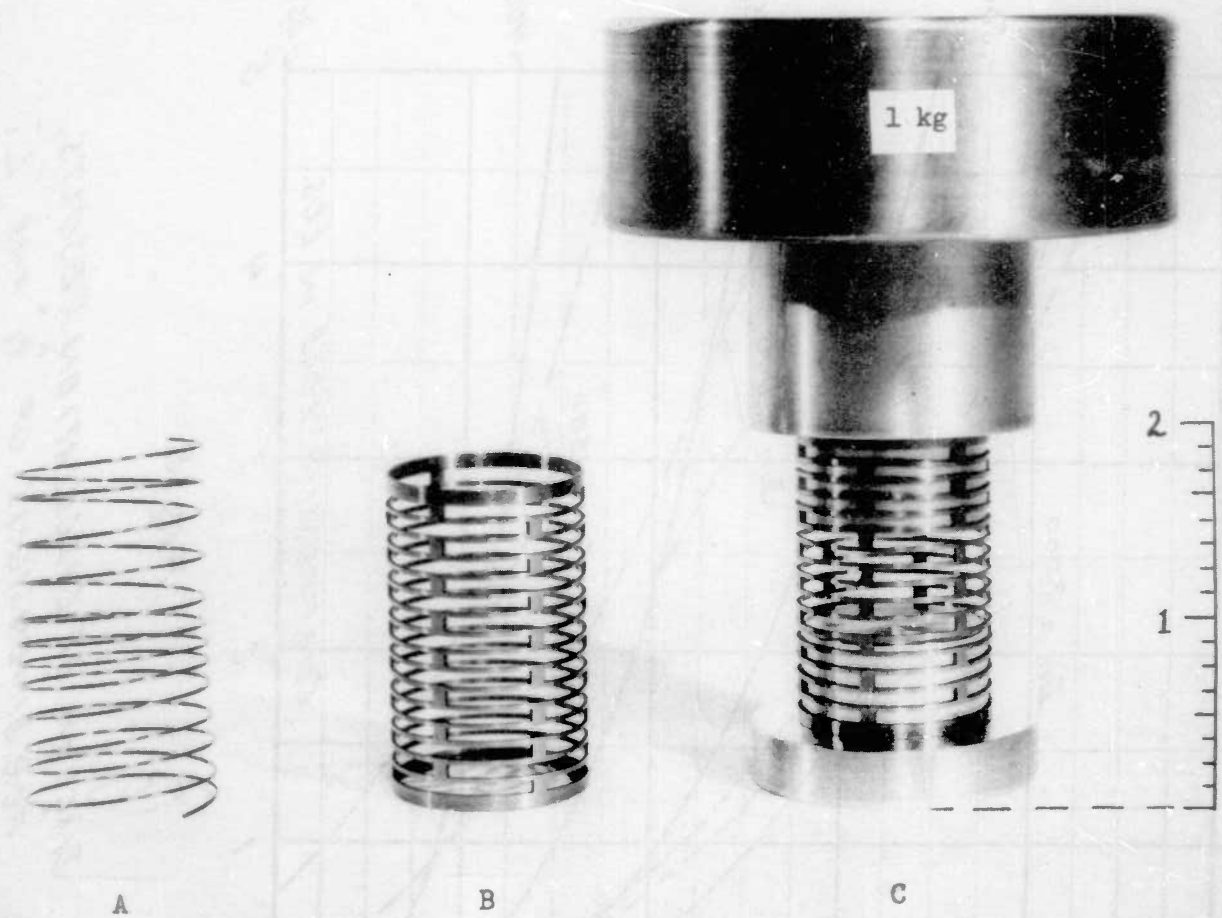



Fig. 7

COMPARISON OF SLOTTED CYLINDER SPRING VS HELICAL COIL SPRING OF SAME DIMENSION
A. Helical Coil Spring, B. Slotted Cylinder Spring, C. Slotted Cylinder Spring Compressed by 1 kg = 2.2 lbs

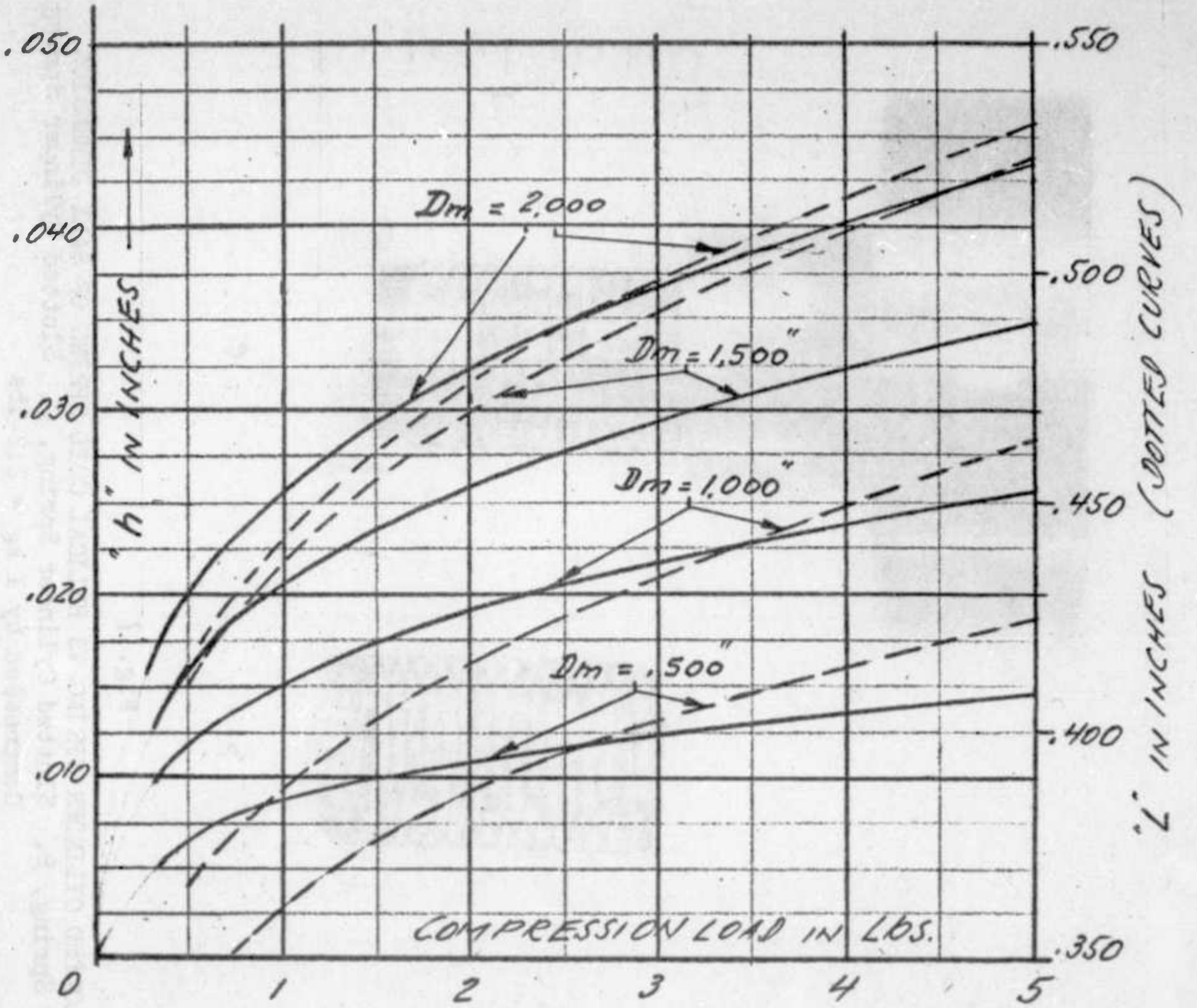



FIG. 8

DIAGRAM OF CALCULATION RESULTS.
FOR DIMENSIONS OF "h" AND "L".

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Army Electronics Research and Development Laboratories, Fort Monmouth, N. J.
DESIGN AND APPLICATIONS OF SLOTTED CYLINDER SPRINGS, by Wilhelm A. Schneider. May 63, 18p. incl. illus. tables, 2 refs.
(AEIRDL Technical Report 2327)
(DA Task 3A99-25-004-02) Unclassified report

A slotted cylinder spring offering unique characteristics of high load capacity and low deflection in extremely small size is discussed in this report. Its use as an elastic element of controllable compliance in seismic transducers is demonstrated and its performance is compared with that of conventional springs. Formulas for these devices are derived and design calculations are given for typical applications.

1. Slotted Cylinder Springs
2. Springs, Slotted Cylinder
3. Elastic Elements
4. Power Springs

I. Schneider, Wilhelm A.
II. Army Electronics Research and Development Laboratories, Fort Monmouth, N. J.
III. DA Task 3A99-25-004-02

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