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RADIOFREQUENCY-CONTACT SPRING

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ABSTRACT

A novel rf-contact spring that has been successfully used on the manhole covers of the prestripper tank and in the transmission-line connections of the heavy-ion linear accelerator is presented in this report. The rf-contact spring is helical in shape, and it is compressed from the sides. Mechanically, the spring is almost indestructible. Electrically, it has the desirable feature that the contact force rises to a maximum with a small deflection and then remains constant with further deflection. The useful elastic deflectionis one-half the radius for this spring, and springs can be designed with the deflection equal to the radius. In this report the problem of selecting the proper contact force for a given current is discussed, and sufficient graphs are presented to allow the design of a spring with a minimum of calculations. A sample calculation is included.

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I. INTRODUCTION

In the heavy-ion linear accelerator (HILAC) the prestripper and the post-stripper cavities are made of copper-clad steel. The rf cavity and the vacuum cavity are integral, and opening one cavity simultaneously opens the other. Therefore all covers must have both a vacuum gasket and an rf contact. This fact suggested the possibility of using only one set of clamping bolts for making both the rf contact and the vacuum seal, if a dependable rf contact could be designed.

An important design requirement was that the covers add a minimum to the magnetic volume of the cavity. If the covers and the rf contacts could maintain the cylindrical shape of the tank, then there would be no detuning of that portion of the cavity.

Other design requirements were that the rf contact have a large accommodation for irregular surfaces, and that it be rugged and foolproof.

It was decided that a helical spring might be designed that would meet these requirements. It was known that a current of 60 amp peak (42 amp rms) would flow through the prestripper cavity and the rf contact on a 3% duty cycle. This means that the equivalent current is $0.03 \times 42 = 1.26$ amp rms. Using William M. Brobeck's criterion for successful rf joints, there must be 10 lb of force for each amp rms per lineal inch of electrical joint.[1] (Brobeck facitly assumes that the softer material of the joint is copper.) This means that we must have $10 \times 1.26 = 12.6$ lb of force per lineal inch of electrical joint. Therefore it was necessary to calculate the spring forces, stresses and deflections in accordance with this value.

In this report is presented a sample calculation for assumed conditions that approximate those existing in the HILAC. Certain inaccuracies are inherent in the formulae that are used; these are discussed in Section IV.

For a more detailed discussion of the mechanical considerations that are involved in calculating spring forces, stresses and deflections, refer to UCRL-8108.[2]

II. A DESCRIPTION OF THE RF-CONTACT SPRING

Figure 1 is a photograph that was taken from inside the prestripper cavity of the HILAC. This photograph shows one of the manhole covers and the rf-contact springs. Figure 2 shows the edge of the manhole cover and reveals clearly the curved inner surface of the cover. The springs are placed as near the curved inner surface as practical, in order to reduce the detuning effects that are caused by any irregularity in the cavity. The individual springs (which are approximately six inches in length) are attached to the cover by inserting the ends through holes drilled in the flange and then by bending over the protruding ends.

Figure 3 shows a cross section of the cavity and the manhole cover. As the cover is moved into place, the springs are compressed between two conical surfaces. The sliding action wears the spring and is therefore undesirable, but the sliding is necessary, because the spring must be placed as near as possible to the inner surface of the cavity. (In the transmission-line connections, the springs are compressed between two parallel plates; therefore no sliding occurs.)

Machining the two conic surfaces (the cavity and the cover) was unnecessary because the springs can accommodate a one-fourth inch variation in the width of the gap. This large accommodation of the spring can absorb any warpage that might occur on the flange as a result of welding, and also absorb differential thermal expansions.

The vacuum gasket is a single hycar piece of rectangular cross section. The groove for this gasket is shown next to the bolt holes in Fig. 2. The vacuum-gasket surface is set away from the tank in order to eliminate or reduce the distortion caused by welding the flange neck to the cavity (Weld A in Fig. 3).

III. CALCULATIONS FOR A SAMPLE SPRING

The sample problem to be considered here is to design a set radial spring with 0.25 in. of useful deflection. The term "set" is used to describe a radial spring that has been plastically deformed so that the free height of the spring is less than the original pitch diameter (i.e., $h < D_o$). (See Table I for a list of abbreviations used in this discussion) Setting the spring so that cot Θ is greater than the coefficient of friction of the spring on the surface material guarantees elastic deformation of the spring without first requiring a translation of the compression surfaces. Therefore we will select a set of 0.866, and we will construct our graphs (Figs. 5, 6, 7 and 8) based on this value.

Also for this sample calculation we will assume that we use silver-plated phosphor bronze for the spring. This material is chosen because of its low modulus of elasticity, which gives a large deflection for a given stress. The spring is to be silver plated in order to improve its conductivity. Further, we will assume that the spring's ultimate strength is 160,000 psi.

For the purpose of this sample problem we will also assume that there will be a peak current of 28 amp (20 amp rms) flowing through the joint on a 3% duty cycle. Then the equivalent current is $0.03 \times 20 = 0.6$ amp rms. Using Brobeck's criterion for successful rf joints, there must be at least 10 lb force for each amp rms per lineal inch of electrical joint.[1] This gives a minimum operating force of $10 \times 0.6 = 6$ lb/in. of electrical joint. In order for the spring to have a useful deflection of 0.25 in., it will have to be designed to accommodate a somewhat higher maximum operating force. We will arbitrarily select a maximum operating force of 12 lb/in, giving a range of 6 to 12 lb/in. After we calculate the spring dimensions and stresses, we may have to revise this figure of 12 lb/in., in order to give the desired useful deflection of 0.25 in.

To calculate the pitch diameter (D_o), we arbitrarily select a normalized pitch (p/D_o) of 0.2. Then we consult the force vs. total height curve that we have constructed (Fig. 5). For a p/D_o of 0.2, the maximum $F_n/D_o^2(SF/D_o)$ is 0.0046. To find the corresponding value of $F_n/D_o^2(SF/D_o)$ for the minimum operating force of 6 lb/in., we first write $\frac{6 \text{ lb/in.}}{12 \text{ lb/in.}}(0.0046) = 0.0023$. Then we consult Fig. 4 and find the corresponding h/D_o to be 0.81. The spring whose characteristics are plotted in Fig. 5 will probably bottom solid at $h/D_o \cong 0.5$ to 0.6. Assuming that it will bottom solid at 0.6, we have

$$\Delta h/D_o = 0.81 - 0.6 = 0.21.$$

Because the desired useful deflection is 0.25 in., we write

$$\Delta h = 0.21 D_o = 0.25 \text{ in.}$$

Solving for D_o we obtain

$$D_o = 0.25/0.21 \cong 1 \text{ in.}$$

Thus we need a spring whose pitch diameter (D_o) is approximately one inch.

To calculate the pitch (p), first recall that we arbitrarily chose a normalized pitch (p/D_o) of 0.2. Thus we write

$$p/D_o = 0.2.$$

Letting $D_o = 1$, we obtain

$$p = 0.2 \text{ in.}$$

Thus the pitch must be 0.2 in.

To calculate the wire diameter (d), recall that the minimum operating force was assumed to be 6 lb/in. Therefore the minimum force per turn ($F_{n\ min}$) is

$$F_{n\ min} = 6 \times 0.2 \times D_o .$$

Letting $D_o = 1$ we have

$$F_{n\ min} = 1.2 \text{ lb/turn.}$$

Also recall that $F_n/D_o^{\,2}(SR/D_o)$ equals 0.0023 for the minimum operating force of 6 lb/in. of joint. Then we write

$$1.2/D_o^{\,2}(SR/D_o) = 0.0023.$$

Because D_o equals 1, this becomes

$$1.2/(SR/D_o) = 0.0023.$$

Solving for SR/D_o, we obtain

$$SR/D_o = 522.$$

Now consult Fig. 6, which is the spring-rate curve. Because our material has the same modulus of elasticity as brass, the d/D_o corresponding to an SR/D_o of 522 is 0.109. Therefore we write

$d/D_o = 0.109.$

Solving for d, we obtain

$d = 0.109$ in.

Thus the wire diameter (d) must be 0.109 in.

The nearest commercial wire size is 0.114 in. It is necessary to determine whether or not this size of wire will satisfy our design requirements. Therefore we write

$d/D_o = 0.114/1 = 0.114.$

The corresponding SR/D_o from Fig. 6 is 630. This is approximately 20% larger than the calculated value of 522, and F_n will be large in proportion. If this result is not close enough, the easiest solution is to increase the pitch diameter. Let us try $D_o = 1.063$. Then we write

$d/D_o = 0.114/1.063 = 0.107.$

From Fig. 6 the corresponding SR/D_o is 490. Recall that earlier we wrote

$F_{n\ min} = 6 \times 0.2 \times D_o$

and

$F_n/D_o^2(SR/D_o) = 0.0023.$

Combining these two equations, we find

$1.2\ D_o/D_o^2 = 0.0023\ (SR/D_o).$

Substituting the value $D_o = 1$ into this equation, and solving for SR/D_o, we obtain

$SR/D_o = 490.$

This value is in agreement with the value of SR/D_o for the wire size chosen above having a pitch diameter (D_o) of 1.063 in.

To find the solid height (h minimum), we write

$h_{min} = D_o\ (d/p) = 1.063\ (0.114/0.2).$

Thus we obtain

$h_{min} = 0.605$ in.

To find the maximum operating height (h maximum), we write

$h_{max}/D_o = 0.81.$

Solving for h_{max}, we obtain

$h_{max} = 0.861$ in.

The operating range is $h_{max} - h_{min}$. Thus the operating range is 0.861 - 0.605 = 0.256 in. The free height is equal to the set (0.866) time the pitch diameter (1.063 in.). This gives a result of 0.92 in. The distance of precompression required to enter the operating range is equal to the free height minus the maximum operating height. Thus the distance of precompression is 0.92 - 0.861 = 0.059 in.

To find the bending stress, refer to Fig. 7. For an normalized pitch (p/D_o) of 0.2, and for a normalized height (h/D_o) of 0.605/1.065, or 0.567, the corresponding value of $S/10^6(d/D_o)$ is 1.17. Therefore we write

$$S = 1.17 \times 10^6 \times d/D_o.$$

Substituting the value for d/D_o of 0.107 in this equation and solving for S, we obtain

$$S = 125,000 \text{ psi}.$$

Because the stress is less than 160,000 psi the spring fills all the design requirements.

IV. LIMITATIONS ON THE ACCURACY OF THE CALCULATIONS

Formulae for spring rate, stress and load are based on the bending of a curved beam. For a depth of beam greater than 0.1 the radius of curvature (i.e., for a value of $d/D_o > 0.1$), the length of the inner fiber differs from the length of the outer fiber by enough to affect the accuracy of the calculations.

Probably a larger source of inaccuracy is the use of the spring pitch diameter (D_o) as the diameter of the curved beam. This is justified for springs of small pitch, but for a normalized pitch (p/D_o) of 0.4 the spring rate (SR) is large by 23%.

With all ordinary extension and compression springs, a small variation in pitch diameter or wire size will materially affect the load capacity of the spring. For this reason, the commercial load tolerance on a non-special spring is ± 5%. In the case of the radially loaded spring, the load capacity (in addition to the above conditions) also varies approximately as the square of the pitch. Therefore, computations of load capacity are probably more accurate than the spring can be built.

In the case of the radially loaded spring with a permanent set, the method of setting can determine the load capacity within a variation of about 20%. If one complete turn is set at a time, the deformation of both coil halves is the same. If the whole spring is set in a jig that first stretches the spring to increase the pitch, and then radially squeezes and also axially translates it, the resulting deformation is greater in coil-half one than in coil-half two (i.e., $y_{s1} \gtrsim y_{s2}$). Using $y_{s1}/y_{s2} = 1$ as a base, the values of load (F_n) are up 10% at $y_{s1}/y_{s2} \approx 1.5$, and up 20% at $y_{s1}/y_{s2} = 2$. Figure 8 shows these relationships. (Figure 8 shows the jig that was used for setting the springs for the manhole covers of the HILAC.)

IV. CONCLUSIONS

The HILAC has been operating for about one year at rated power, and there have been no spring replacements due to burnout. One spring has been replaced because of mechanical damage, but the circumstances only prove the ruggedness of the design. One of the manhole covers was being removed from the tank, and the rigging broke when the cover was approximately one foot above the hole. The rf-contact springs piloted the 400-pound cover back into the hole and absorbed enough shock so that nothing was damaged except one 6-inch section of the spring. It took only a minute to clip off the ends of the damaged spring, insert the ends of a new spring, and bend them over.

Although this report has discussed the application of the rf-contact spring to the HILAC, it is apparent that it could be used in other situations having similar design requirements. For instance, if it were necessary to have an rf contact between a sliding position and a cylinder wall, this rf-contact spring would be ideal.

REFERENCES

1. William M. Brobeck, Summary of Radiofrequency Joint Designs, UCRL-1311, May 1951.
2. Luther R. Lucas, Heavy-Ion Accelerator Prestripper Tank Elastic Compression of Radially Loaded Helix, UCRL-8108, April 1957.

ACKNOWLEDGMENTS

I wish to thank Hayden Gordon, Chief Engineer, Radiation Laboratory, University of California, Berkeley, and former Project Engineer for the HILAC, for encouraging the design of this rf-contact spring. Also I wish to thank Neil Norris for carrying out the electrical testing of the spring.

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Table I

List of Abbreviations
(See Fig. 4)

d	diameter of wire (in.)
D_o	Pitch diameter of spring before load (in.)
F_n	external normal force to compress one coil (lb)
h	height of coil as measured to wire centers and perpendicular to the coil axis (in.)
Δh	useful deflection (accommodation) (in.)
p	pitch (in.)
S	bending stress (psi)
SR	spring rate (theoretical force to compress one inch) (lb)
Θ	angle measured between centerline and the compression coil-half of the spring
y_{s1}	permanent set in coil-half one
y_{s2}	permanent set in coil-half two
d/D_o	normalized wire diameter (unitless)
p/D_o	normalized pitch (unitless)
h/D_o	normalized height (unitless)
SR/D_o	normalized spring rate ~~(unitless)~~ (pounds per inch of deflection per inch pitch diam)

LEGENDS

Fig 1. Interior view of the prestripper cavity of the heavy-ion linear accelerator, showing a manhole cover and the rf-contact springs.

Fig. 2. A manhole cover for the prestripper cavity of the heavy-ion linear accelerator, showing the rf-contact springs and the curved inner surface of the cover. The groove for the hycar vacuum gasket is on the flange near the bolt holes.

Fig. 3. A cross section of the prestripper cavity of the heavy-ion linear accelerator, showing a manhole cover and the rf-contact spring.

Fig. 4. Diagram illustrating the abbreviations used in this report. (See Table I)

Fig. 5. Force vs total-height curve for a helical spring of set 0.866, where $y_{s1} = y_{s2}$.

Fig. 6. Spring-rate curve for a ~~helical spring of set 0.866~~ 180° curved beam, where $SR/D_o = 7.486(d/D_o)^4 \times 10^6$ for steel, and where $SR/D_o = 3.743(d/D_o)^4 \times 10^6$ for brass.

Fig. 7. Bending-stress curve for a helical spring of set 0.866. The values shown are for steel, where $E = 30{,}000{,}000$. For other materials multiply chart values by the ratio E_n/E_s (E_n is the value for the new material, and E_s is the value given for steel).

Fig. 8. A comparison of force-deflection curves for various methods of setting radial springs, where the free h/D_o is to be 0.866.

Fig. 9. The jig used for setting the rf-contact springs for the manhole covers of the heavy-ion linear accelerator.

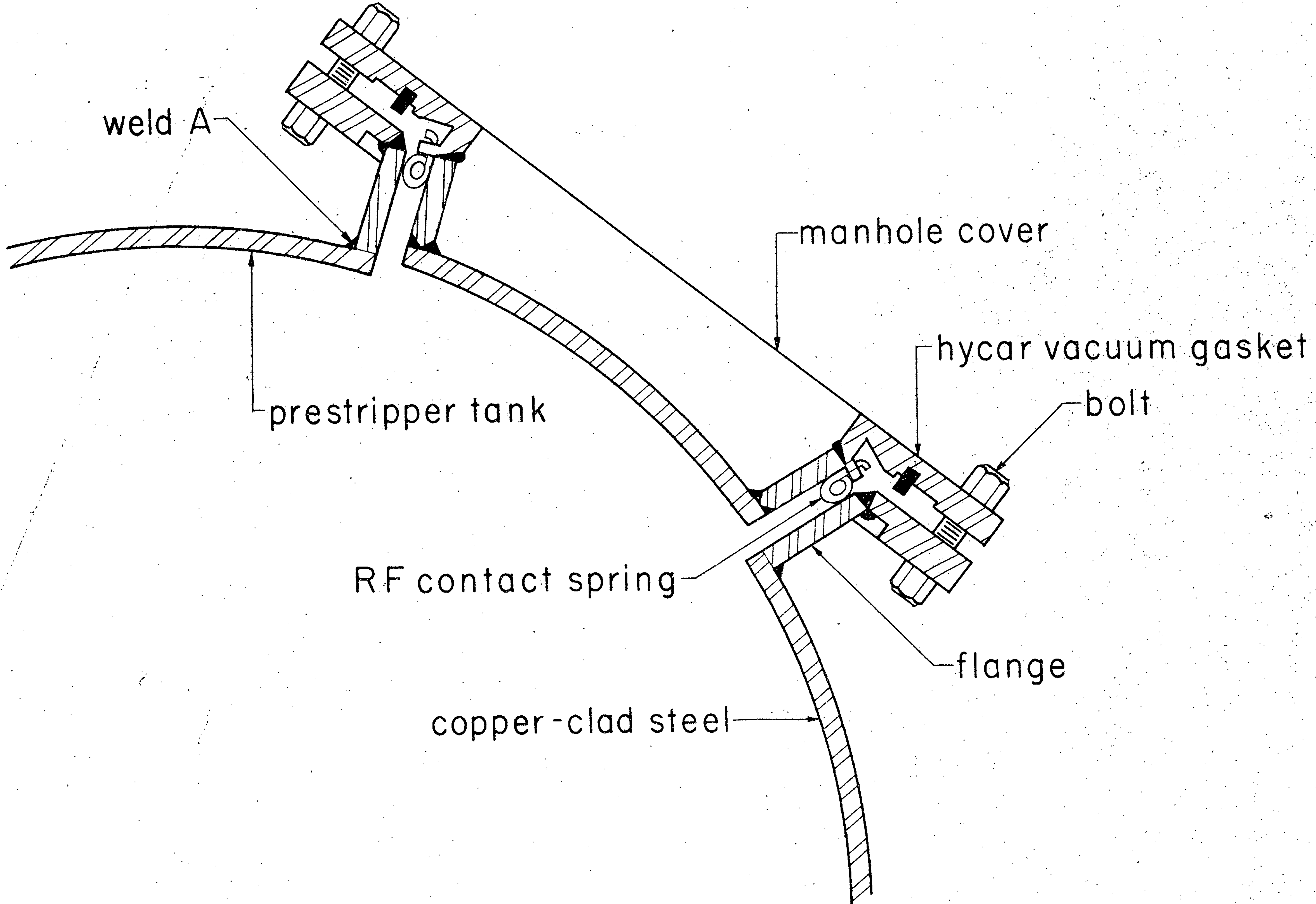
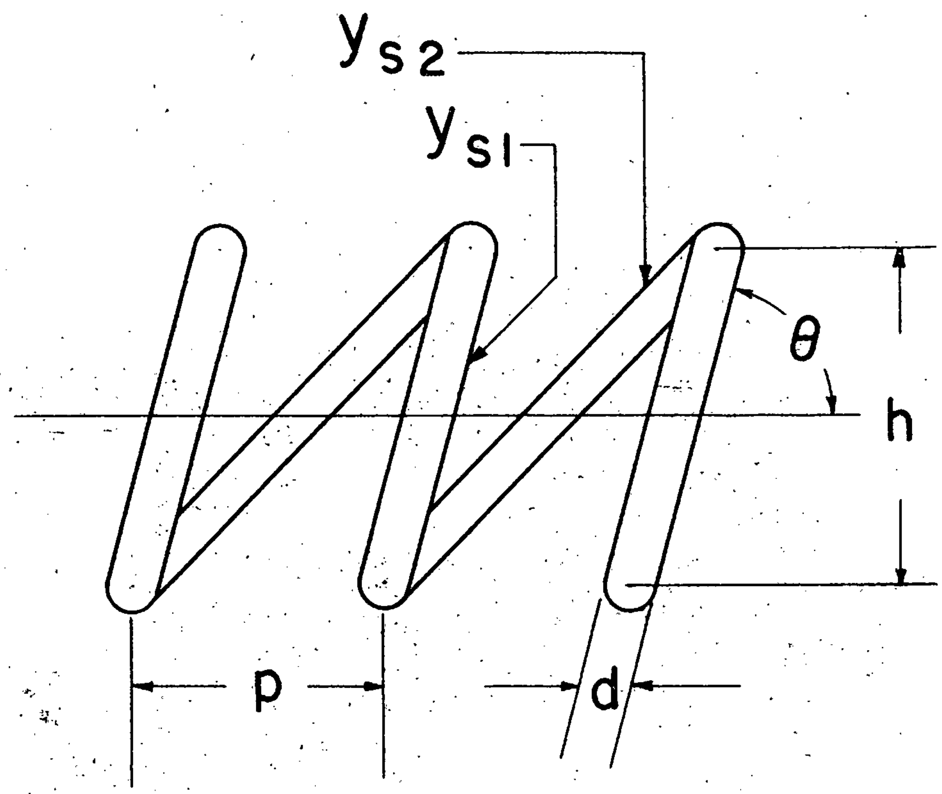

weld A
manhole cover
hycar vacuum gasket
bolt
prestripper tank
RF contact spring
flange
copper-clad steel

Fig 3.

50970-2



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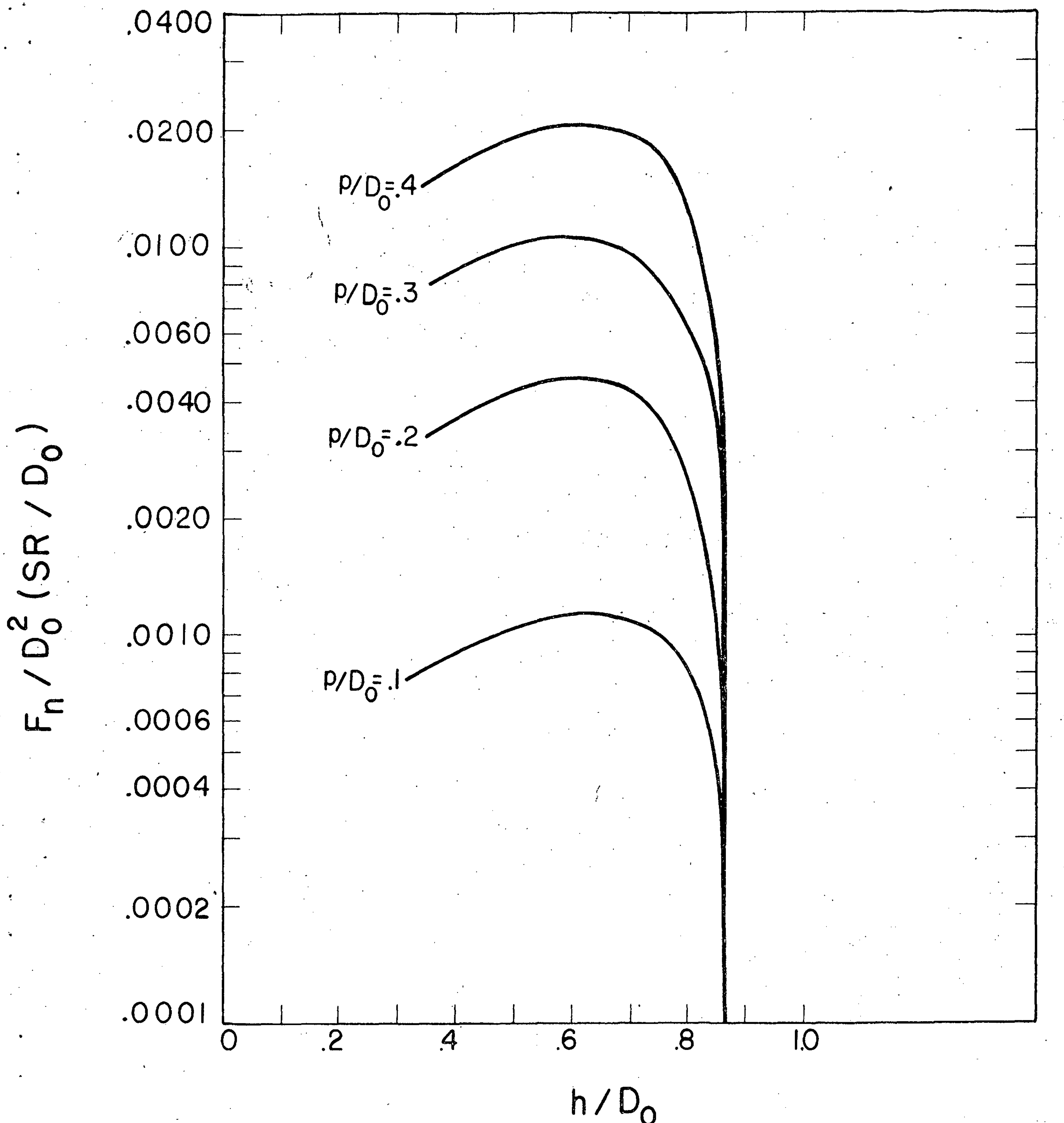
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.0200
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.0060
.0040
.0020
.0010
.0006
.0004
.0002
.0001
0
.2
.4
.6
.8
1.0
p/D0=.4
p/D0=.3
p/D0=.2
p/D0=.1
Fn / D0^2 (SR / D0)
h / D0

Fig 5

50,969-1

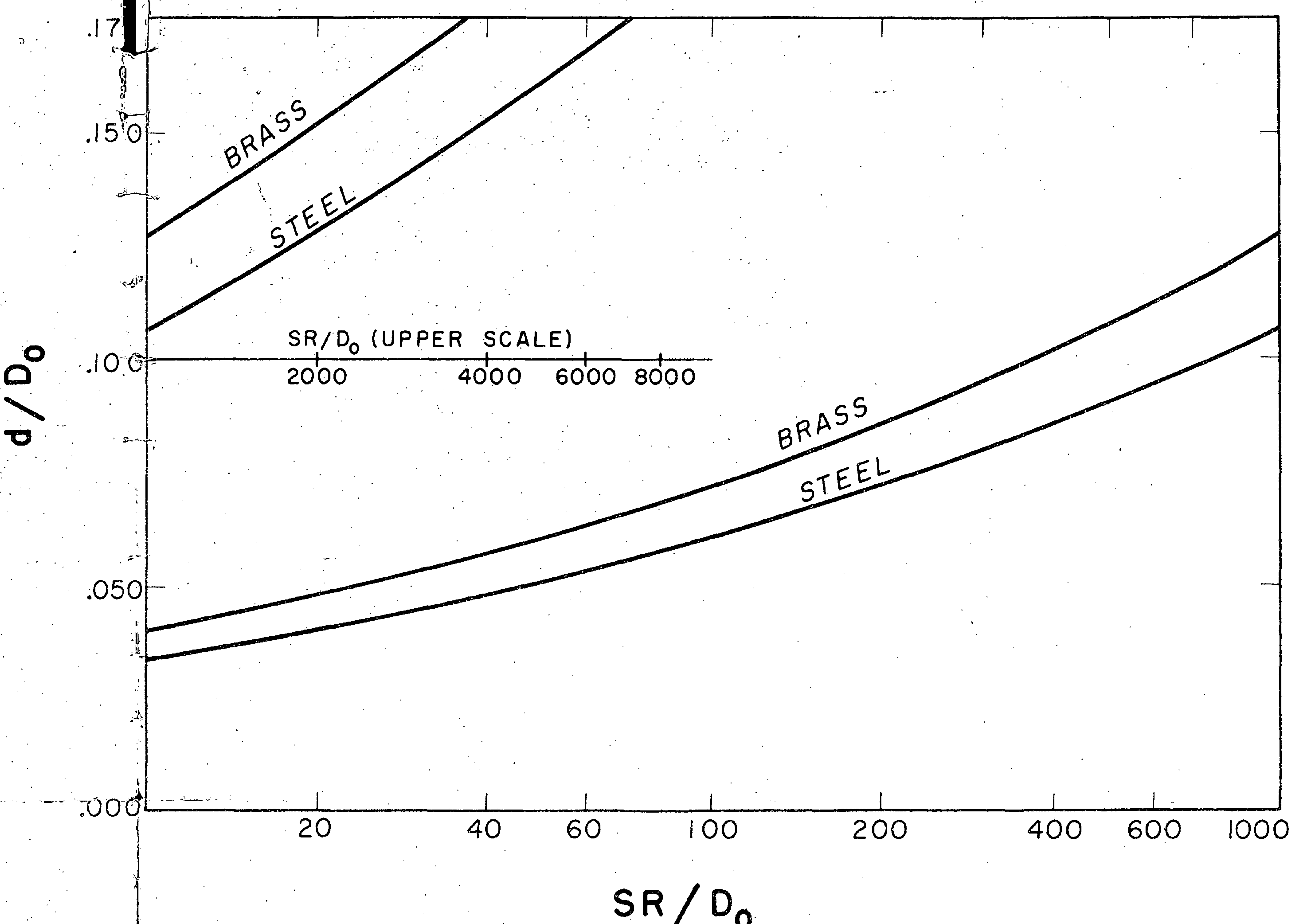
.17
.150
.100
.050
.000
d / D_o
BRASS
STEEL
SR/D_o (UPPER SCALE)
2000
4000
6000
8000
BRASS
STEEL
20
40
60
100
200
400
600
1000
SR / D_o

Fig 6

50,966-2

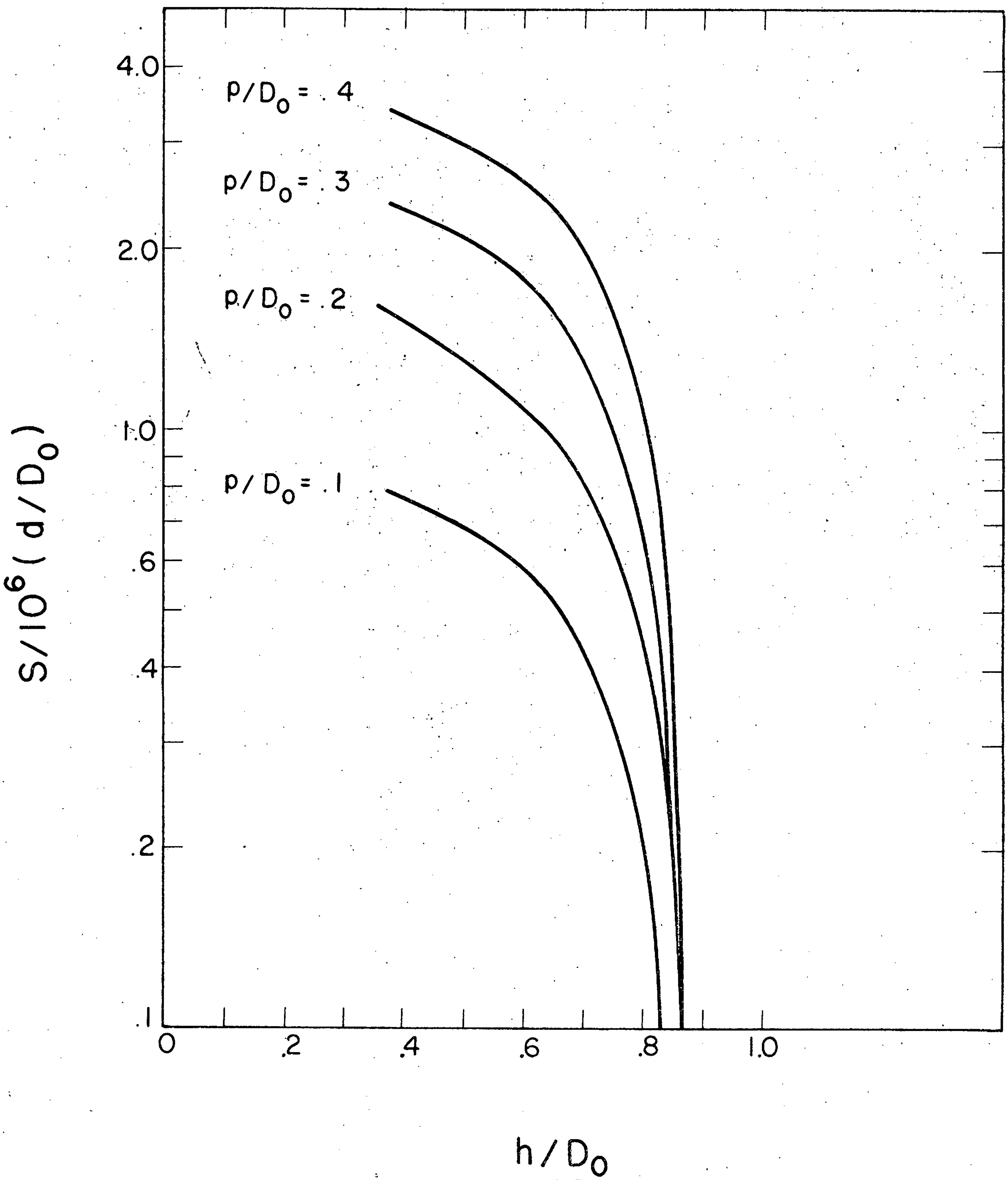

Fig 7

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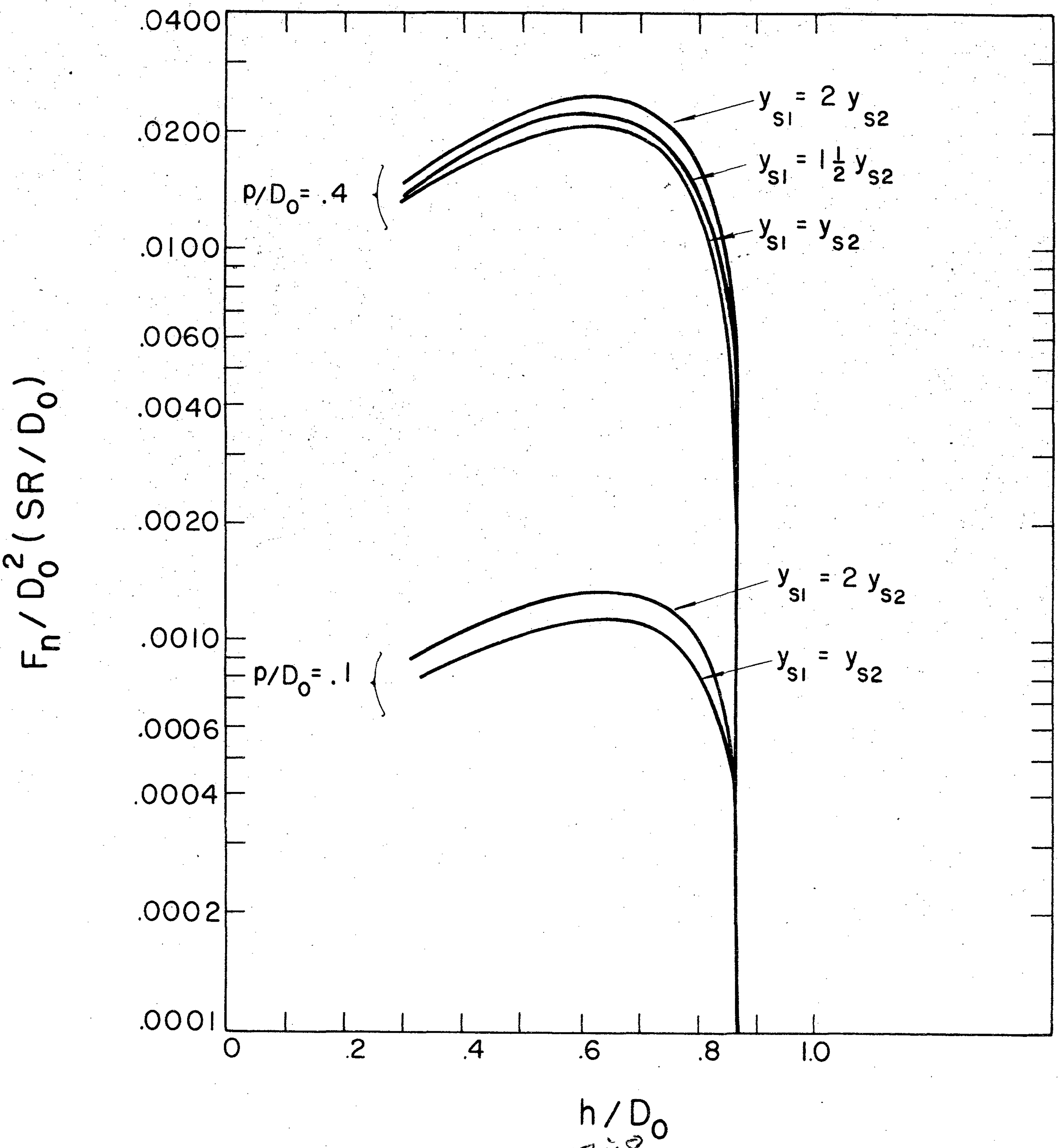

Fig 8

50,968,21

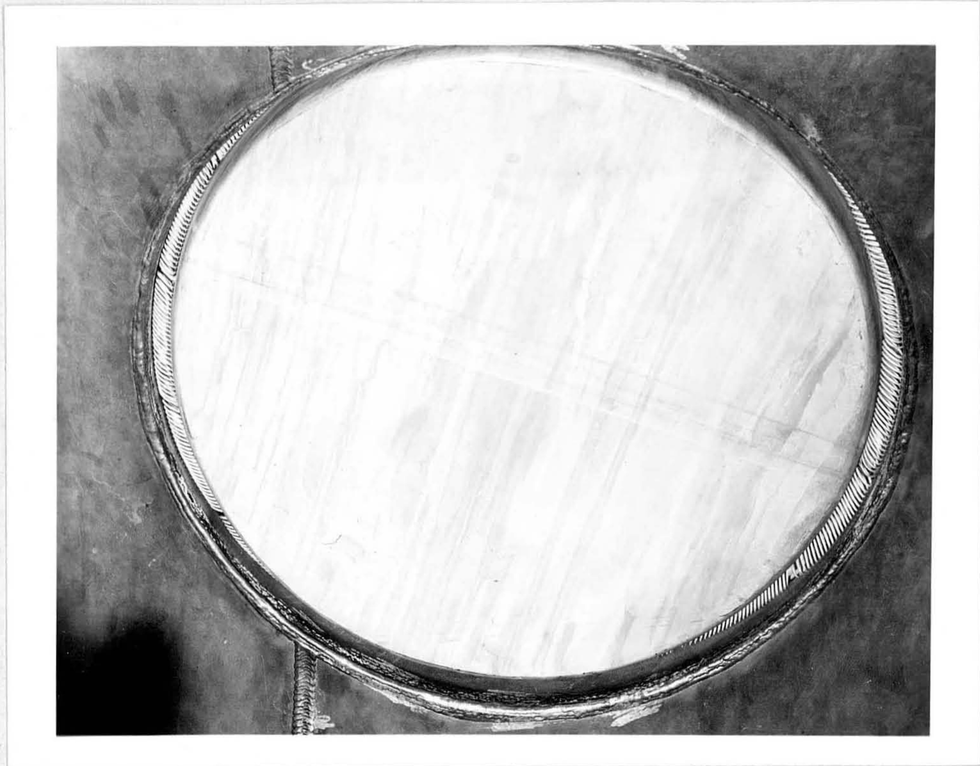

fig. 1

Fig. 2.

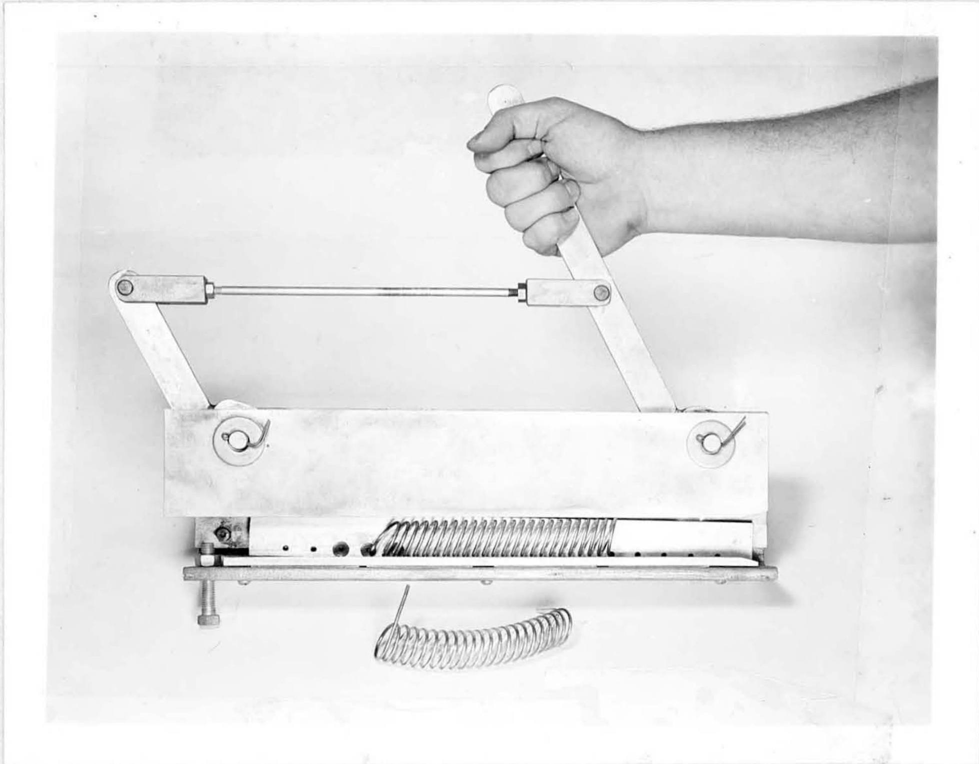

Fig. 9