Jan. 20, 1970 M. L. GREENBERG 3,490,819

HYDROSTATIC SPINDLE

Filed Jan. 18, 1968 2 Sheets-Sheet 1

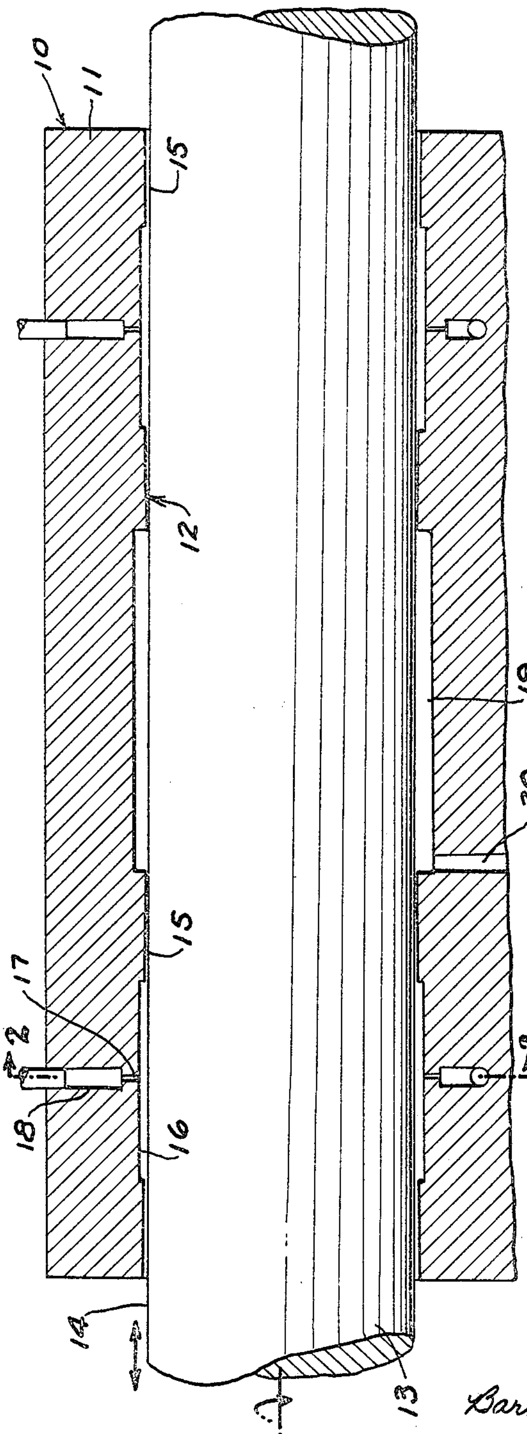


FIG. 1

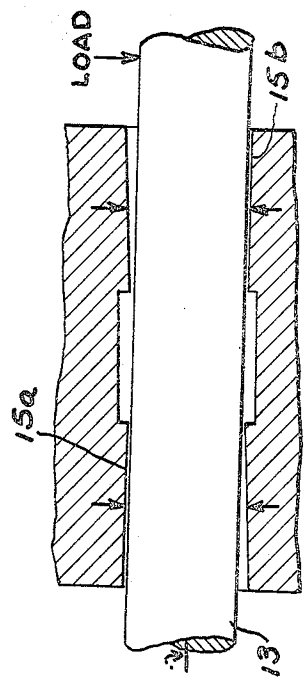
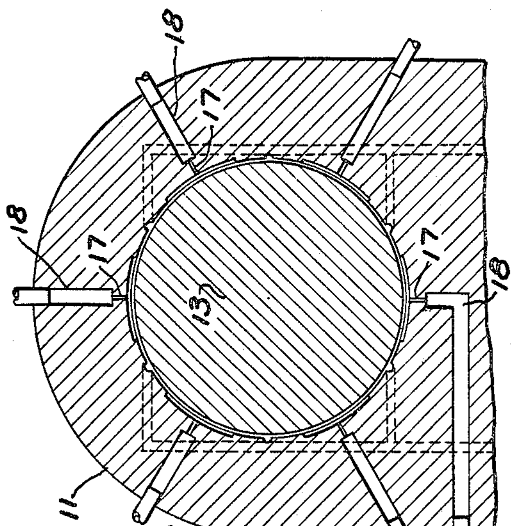


FIG. 3

FIG. 2



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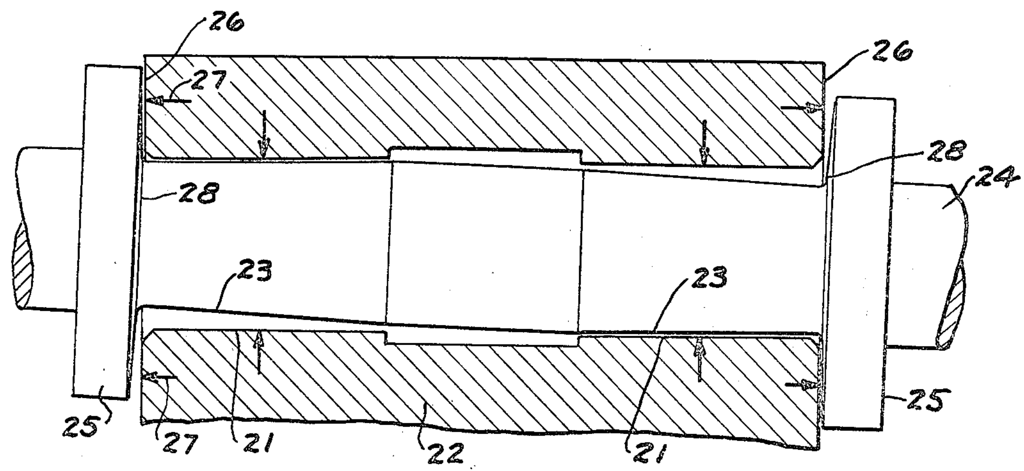

FIG. 4

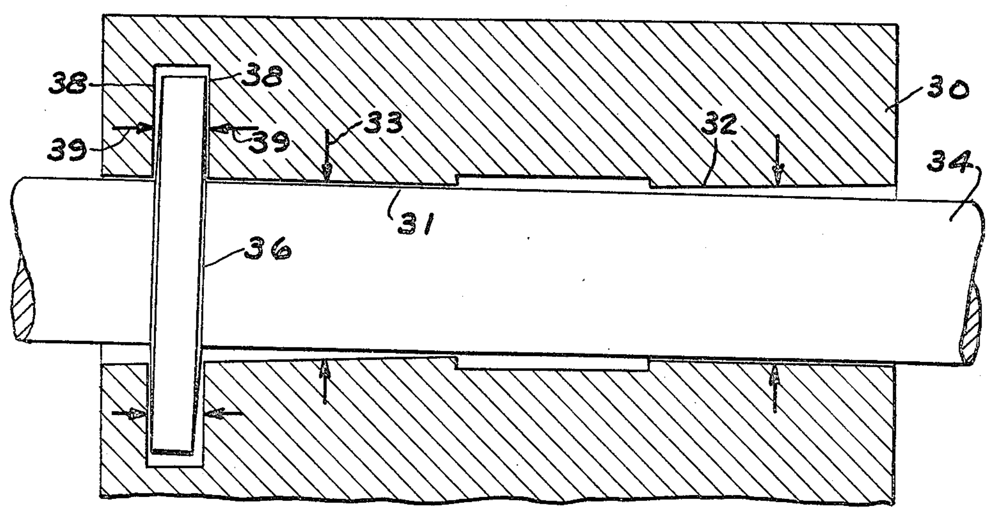

FIG. 5

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HYDROSTATIC SPINDLE

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ABSTRACT OF THE DISCLOSURE

The hydrostatic spindle disclosed herein comprises a housing having a bore and a rotor positioned in the bore. The housing bore and the rotor have two generally complementary surfaces. Two longitudinally spaced sets of circumferentially spaced hydrostatic pressure pockets and associated restrictors are provided in the surfaces of one of the members. When the bore and the rotor axes are aligned, the complementary surfaces thereof diverge axially and radially outwardly.

This invention relates to hydrostatic spindles. In the design and testing of hydrostatic spindles which have longitudinally spaced sets of pressure pads, it is common procedure to predict lateral moment stiffness of the spindle by approximately the following method:

(1) The lateral load capacity of a journal element is determined by a conventional approximation, usually that of projected area (journal diameter × length of one pad) times an "effective pressure" which can be determined from pocket pressure and a geometric constant associated with the pad configuration. These procedures are well known and need no elaboration.

(2) The journal element forces are assumed to be acting at the center of the pads, and the moment arm is assumed to be the distance between centers.

(3) The moment stiffness is then the angular deflection divided into the product of load capacity of a journal element and moment arm.

Assuming that,

S=pad length (in.)
l=moment arm (in.)
d=journal diameter (in.)
a_p=pad efficiency coefficient (dimensionless)
F=load capacity of one journal element (lb.)
P_0=pocket pressure
A=projected area of one journal element
h=gap (in.)

Then:

M=moment capacity
$A=ds$
$F=a_p ds P_0$
$M=Fl=a_p lds P_0$
x=possible angular deflection (radius)

then:

$$x=\tan x=\frac{2h}{1+s}$$

and

$$\text{moment stiffness}=\frac{a_p ldsP_o}{\frac{2h}{(l+s)}}$$

However, this is an instantaneous expression. If we assume that the compensation is linear up to 85% of full deflection, then the greatest possible value of $P_0=.85P_m$, and the maximum stiffness, which is the figure we are after, becomes:

$$S_m=\frac{.85a_p ldsP_m}{\frac{2h}{l+s}}\text{ in.-lb./radian}$$

In a typical case,

$a_p=.6$
$s=4''$
$d=2.5''$
$l=8''$
$P_m=300$ p.s.i.
$h=.001''$

$$S_m=\frac{.85\times .6\times 8\times 2.5\times 4\times 300}{2\times .001}$$
$$=73.44\times 10^6\text{ in.-lb./rad.}$$

This appears on the surface to be a simple and straightforward way of calculating, which should produce fairly accurate predictions of compliance. In fact, however, predictions made by this method vary from twice to five times too high. Although I do not wish to be bound by the theory involved, it is believed that at least two effects contribute to this regrettable situation; the nonlinearity of pocket pressure change with respect to deflection (assuming a rigid shaft), and deflection of the shaft, which introduces curvature about an axis perpendicular to the spindle axis and further increases this nonlinearity.

A hidden and infrequently noted assumption always made in the design of hydrostatic bearings of this type is that all of the above calculations, and all of the considerably more complex ones of which these are a working simplification are based on the assumption that the gap between shaft and housing is constant over the full extent of any given pad, especially those in the plane of the applied moment. In some situations this is a legitimate approximation, but in the case of a spindle I have determined that the gap can easily vary 30% along the open side and to an even greater degree on the short side. This situation produces, in place of the uniform flow and pressure gradient around a pad usually assumed, a very large variation in these parameters as the perimeter of the pad is traversed. Since flow is proportional to the cube of the gap, the average flow will be much too high and will produce a much lower maximum pocket pressure than is predicted by the simplified theory and so a much lower actual stiffness.

It is an object of this invention to overcome this difficulty by providing a configuration whose stator-to-rotor gap becomes more uniform as the deflection increases, which can be manufactured with reasonable facility, and which has a higher moment stiffness at large deflections.

In the drawings:

FIG. 1 is a longitudinal sectional view through a rotary spindle embodying the invention.

FIG. 2 is a fragmentary sectional view taken along the line **2**—**2** in FIG. 1.

FIG. 3 is a diagrammatic view of the spindle shown in FIG. 1 showing the action of the spindle under lateral load.

FIG. 4 is a partly diagrammatic longitudinal sectional view through a modified form of rotary spindle.

FIG. 5 is a partly diagrammatic part sectional view through a further modified form of rotary spindle.

Referring to FIG. 1, a rotary spindle **10** embodying the invention comprises a housing **11** having a bore **12** in which a rotor **13** is positioned for rotation and axial movement relative to the housing. The rotor has a cylindrical outer surface **14**. The housing has generally complementary annular surfaces **15** each of which has circumferentially spaced pressure pads or pockets **16** therein with associated restrictors **17**, that may be in the form of capillary tubes, to which fluid under pressure is supplied through inlet line **18**. An annular drain groove **19** is provided for collecting the fluid that flows inwardly from the pockets **16**. External drains and seals, not shown, complete the system. The fluid flows from pocket **19** to a passage **20** and then to the reservoir. This configuration is not part of this invention and is shown in the patent to Porath 3,200,671.

In accordance with the invention, the annular surfaces **14, 15** are so related that when the axis of the rotor **13** is aligned with the axis of the bore **12**, the surfaces **14, 15** diverge axially and radially outwardly. As a result when the moment load is applied laterally on the rotor **13** as shown in FIG. 3, the surface **14** of the rotor is brought into substantially parallel relation to the surface **15** at the areas **15***a*, **15***b* resulting in considerably improved stiffness. As shown diagrammatically in FIG. 3 wherein arrows represent the pressure pockets and restrictors, the tapering relation of the surfaces **14, 15** is exaggerated. In practice it is small being on the order of 0.0003" in 2½" of axial length of the surface.

Although I do not wish to be bound by the theory involved, in my opinion, the increased stiffness is achieved by providing the relationship such that upon lateral load on the rotor the two complementary surfaces are brought into nearly parallel relation resulting in the more effective pressure compensation.

In the form of the invention shown in FIG. 4, (see Porath 3,223,463) the pressure pads and associated restrictors are shown in the form of arrows for purposes of clarity. In this form the surfaces **21** of the housing **22** are parallel to the axis of the bore in the housing and the complementary surfaces **23** of the rotor **24** are tapered to produce the tapering relationship of the surfaces **23, 21** when the axis of the rotor **24** is aligned with the axis of the bore in the housing. This is possible since in such a trapped spindle, there is no axial motion.

In addition radially extending flanges **25** are provided at longitudinally spaced points on the rotor **24** and are associated with surfaces **26** on the housing that have circumferentially spaced pressure pockets and restrictors **27**. In this form the surfaces **28, 26** taper outwardly relative to one another when the axis of the rotor **24** and the bore in the housing are aligned so that when there is a lateral moment load on the rotor as shown in exaggerated position shown in FIG. 4, not only are the surfaces **23, 21** brought into more nearly parallel relation but in addition portions of the thrust surfaces **26, 28** are brought into more nearly parallel relation.

In the form of the invention shown in FIG. 5, the housing **30** has surfaces **31, 32** in which the pressure pockets and restrictors **33** are provided, as in the form of the invention shown in FIGS. 1 and 3, while the rotor **34** has a radial flange **35** with surfaces **36, 37** that are tapered with relation to radial surfaces **38** in the housing **30**. Pressure pockets and associated restrictors **39** are provided in the surfaces **38**. When the axis of the rotor **34** is aligned with the axis of the bore in the housing **30**, the surface of the rotor and the surfaces **31, 32** diverge outwardly and radially while the surfaces **36, 38** and **37, 38** diverge outwardly.

It will be understood that some sacrifice in axial thrust load is made by tapering surfaces **36, 37**. If thrust capacity is more important than the comparatively minor additional moment capacity of the thrust surfaces, then the taper on surfaces **36, 37** can be omitted.

I claim:

1. In a hydrostatic spindle, the combination comprising

a housing having a bore therein,

a rotor in said bore,

said bore and said rotor having two longitudinally spaced generally axially extending complementary surfaces,

each surface of one of said housing and said rotor having circumferentially spaced pressure pockets therein,

a restrictor associated with each said pressure pocket,

means for supplying fluid under pressure to each said restrictor,

each surface of one of said bore and said rotor tapering axially and radially outwardly with respect to the complementary surface of the other of said bore and said rotor when the axis of the rotor and the axis of the bore are aligned whereby when fluid is supplied to each of said pressure pockets and a lateral moment load is provided on said rotor and provides improved maximum level of fluid pressure within the pressure pocket.

2. The combination set forth in claim **1** wherein said pressure pockets are provided on said housing.

3. The combination set forth in claim **1** wherein said surfaces of said housing are tapered outwardly and radially.

4. The combination set forth in claim **1** wherein said surfaces of said rotor are tapered.

5. The combination set forth in claim **1** including a pair of longitudinally spaced radially extending flanges on said rotor,

said housing having generally complementary radially extending surfaces,

one of said surfaces having a plurality of circumferentially spaced pressure pads thereon,

restrictors associated with said last-mentioned pressure pads,

and means for supplying fluid under pressure to said last-mentioned restrictors,

the radial surfaces of said flanges and the radial surfaces of said housing normally diverging outwardly relative to one another when the axis of the bore and the axis of the rotor are aligned.

6. The combination set forth in claim **1** wherein said rotor has a radial flange thereon,

said flange having generally radial surfaces thereon,

said housing having generally complementary radial surfaces,

curcumferentially spaced pressure pockets on one of said surfaces,

a restrictor associated with each said pressure pocket,

and means for supplying fluid under pressure to said last-mentioned restrictors,

said radial surfaces of said flange and said radial surfaces of said housing being tapered outwardly relative to one another when the axis of said bore and the axis of said rotor are aligned.

7. In a hydrostatic spindle, the combination comprising

a housing having a bore therein,

a rotor in said bore,

said bore and said rotor having two longitudinally spaced generally axially extending complementary surfaces,

each surface of one of said rotor and said bore being cylindrical,

each surface of said housing having circumferentially spaced pressure pockets therein,

a restrictor associated with each said pressure pocket,

means for supplying fluid under pressure to each said restrictor,

each surface of one of said bore and said rotor tapering axially and radially outwardly with respect to the complementary surface of the other of said bore and said rotor when the axes of said rotor and said bore are aligned whereby when fluid is supplied to each of said pressure pockets and a lateral moment load is provided on said rotor and provides improved maximum level of fluid pressure within the pressure pocket.

8. The combination set forth in claim **7** wherein said pressure pockets are provided on said housing.

9. The combination set forth in claim 7 wherein said surfaces of said housing are tapered outwardly and radially.

10. The combination set forth in claim 7 wherein said surfaces of said rotor are tapered.

11. The combination set forth in claim 7 including a pair of longitudinally spaced radially extending flanges on said rotor,

said housing having generally complementary radially extending surfaces,

said radially extending surfaces of one of said rotor and housing forming a right angle with respect to the axes of said one of said rotor and said housing,

one of said surfaces having a plurality of circumferentially spaced pressure pads thereon,

restrictors associated with said pressure pads,

and means for supplying fluid under pressure to said last-mentioned restrictors,

the radial surfaces of said flanges and the radial surfaces of said housing normally diverging outwardly relative to one another when the axis of the bore and the axis of the rotor are aligned.

12. The combination set forth in claim 7 wherein said rotor has a radial flange thereon,

said flange having generally radial surfaces thereon,

said housing having generally complementary radial surfaces,

a circumferentially spaced pressure pocket on said radial surfaces of said housing,

a restrictor associated with each said pressure pad,

and means for supplying fluid under pressure to said last-mentioned restrictor,

said radial surfaces of said flange and said radial surfaces of said housing being tapered outwardly relative to one another when the axis of said bore and the axis of said rotor are aligned.

13. The combination set forth in claim 7 wherein said rotor has a radial flange thereon,

said flange having generally radial surfaces thereon,

said housing having generally complementary radial surfaces,

a circumferentially spaced pressure pocket on said radial surfaces of said housing,

a restrictor associated with each said pressure pad,

and means for supplying fluid under pressure to said last-mentioned restrictor.

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