HYDROSTATIC BUTTON BEARINGS FOR PUMPS AND MOTORS

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ABSTRACT

In axial-piston, hydraulic pumps and motors, the thrust loadings of the pistons on their bearing supports vary proportionally to pressures in the unit and also circumferentially around the bearing supports. By employing hydrostatic button bearings pressurized in "sets" around the bearing support with at least one set pressurized with the unit's inlet pressure and another "set" pressurized by the unit's outlet pressure, improved bearing performance is achieved. Also, certain surface configuration forming the sills and recesses of the individual buttons can enhance bearing performance.

6 Claims, 9 Drawing Figures
HYDROSTATIC BUTTON BEARINGS FOR PUMPS AND MOTORS

BACKGROUND OF THE INVENTION

Currently, axial-piston hydraulic pumps and motors are often required to operate at pressures in the range of 5,000 p.s.i. which can develop high-oblique thrust forces on the bearings that vary as the components rotate. When the units are of the variable displacement type, the angle of the oblique thrust forces can vary appreciably.

Prior art units of the above type have employed hydrostatic buttons at the socket end of the piston or connecting rod and a passage through each rod and its associated piston whereby the associated button is pressurized at the cylinder pressure of its rod. Examples of such prior art arrangements are illustrated in U.S. Pat. No. 2,241,701 issued to Dow and U.S. Pat. No. 3,121,816 issued to Firth. One of the difficulties with the prior art arrangements is that the pressures in the bearing pad vary sharply, sometimes exceeding the maximum pressure in the unit and at other times going below the minimum pressures required for proper bearing support, particularly on the intake stroke of the associated piston. This situation along with the stabilizing problems of the individual buttons experienced at high-rotational speeds has left room for measurable improvement.

Further, the manufacture of the hydrostatic button bearings and the required fluid supply system for axial piston-type hydraulic units shown in the prior art arrangements are expensive and complex. In addition, individual replacement of the button bearing is often impractical, if not impossible. Thus, considerable difficulty is experienced with correct clogging problems in the prior art arrangements and should one of the buttons fail to operate properly (clog), the thrust bearing for that piston rod is lost completely. This situation can quickly cause scoring of the associate runner surface on which the button bearings track and subsequently the failure of the complete thrust bearing system.

In addition, many of the prior art button bearing systems can not efficiently cooperate with mechanical bearings to jointly absorb the thrust forces, which vary with displacement angle, around the thrust plate or ring assembly since it is difficult to simultaneously maintain the proper clearances for the two-bearing arrangements in such systems.

SUMMARY OF THE INVENTION

Most of the above difficulties can be eliminated and improved performance achieved by the hydrostatic thrust support system of this invention which is more economical than the prior art systems. Basically, it is employed in axial piston-type hydraulic units having a plurality of connecting rods journaled in a common rotating thrust plate assembly and includes a plurality of pockets circumferentially disposed in the units structure adjacent to the thrust plate assembly, passage means connecting each pocket to a source of pressurized fluid, and a plurality of button bearing means received in each pocket, each button bearing means including a pad area cooperating with a surface on the thrust plate assembly, said pads including a recess having communication with its associated pocket and a circular sill surrounding said recess whereby pressure acting on each button bearing means will urge it toward said surface so the latter will be supported on a hydrostatic film. It is preferred that the buttons be pressurized in semicircular "s-sets," one "set" pressurized by passages connecting them to the unit's inlet, the other set pressurized by passages connecting them to the unit's outlet. This provides superior thrust compensation for the circumferentially varying thrust forces about the thrust plate assembly. Further, special pad area configuration will provide a tendency for the individual buttons to tip or tilt slightly due to the hydrodynamic film built-up so that the maximum clearance exists at the leading edge of the button which is caused by the rotation of the cooperating surface on the thrust plate assembly. The above arrangement not only provides for simple and expedient button replacement not possible in prior art designs.
the following functions and advantages: The hydrostatic but-
tons support part of the piston thrust load, thus providing
lower loads on the antifriction bearing. This allows a smaller
antifriction bearing to be used and provides longer bearing
life.

The amount of support provided by the hydrostatic buttons
is constant for a given pressure, but the axial component of
the piston thrust force varies with displacement angle. The an-
tifriction bearing supports the difference in the axial loads as
well as the radial component of the piston thrust force.

To accomplish this objective the thrust ring 32 is mounted
on a face of the radial flange 18 with dowels 33 so its thrust
face 34 can form the runner part of a separate hydrostatic
bearing formed with each of the buttons. The individual but-
tons 35 are circumferentially disposed in pockets 36 and 37
in the trunnion 10, as can be best seen in FIG. 2. Pockets 36,
through passages 38 and 39, are in fluid communication with
one side of the unit while pockets 37, through passages 40
and 41, are in fluid communication with the other side. Thus,
pockets 36 form one semicircular "set" of buttons while
pockets 37 form another semicircular "set" whereby one -
set will always be pressurized at the unit's inlet pressure and
the other "set" always will be pressurized at the unit's outlet
pressure.

Utilizing two semicircular "sets," pressurized at the inlet
and outlet pressures respectively, avoids the individual pres-
sure peaks occurring on the individual pistons and allows the
rigidity of the thrust plate assembly to distribute the thrust
loading more evenly on several of the individual button
bearings composing the "set." Use of the buttons in high-
and low-pressure sets allows the centroid of the hydrostatic button
forces to be located close to the centroid of the axial piston
thrust forces, thus minimizing moment unbalance on the
thrust plate. With the buttons in sets and located as indicated,
pressure in each set is continuous and does not fluctuate
between inlet and discharge pressure as it would if the button
rotated with the piston and rod were lubricated by a
passage through each piston rod. The passages are arranged so
the higher pressure will be on the "set" with the highest load-
ing, whether the high pressure is the unit's inlet or outlet. It is
through such an arrangement that substantial advantages can
be gained and button replacement greatly simplified.

The general configuration of each button 35 is best shown in
FIGS. 4 to 9 wherein it can be seen that each button includes a
head portion 50 and a cylindrical skirt 51 having a groove 52
for a seal 53 which prevents leakage of pressurized fluid
around the skirt from pocket 36 (or 37) and passage 38 or 40.
It can be seen that the individual button has considerable
freedom of movement within its associated pocket.

The skirt 51 of each button is preferably hollow and may in-
clude a strainer or screen 54 over its mouth or opening 55
communicating with pressurized fluid in its associated pocket,
which is best seen in FIGS. 3 and 9. The skirt fits loosely in its
pocket so the button can tip or tilt about the centerline of the
pocket. This arrangement allows the individual button to track
smoothly on the associated surface 34 and the bearing formed
by the button to be less effected by distortion occurring in
the thrust plate assembly due to high-thrust loadings. The buttons
having a degree of freedom to tilt and move axially can adjust
for such things as nonparallelism between the thrust plate and
trunnion, variation in assembled distance between thrust plate
and trunnion, difference in running clearance required with
changes in pressure, and/or temperature.

Basically each button 35 forms a separate hydrostatic bear-
ing with the pad or face 56 of the button having a recess 57 in
combination with the pressurized fluid in its pocket through an orifice or restrictor and a raised circular sill 58
which mates with the flat face or surface 34 of the thrust ring
32 which forms the runner of the bearing.

Since the thrust ring is rotating, there is some hydrodynamic
bearing effect along with the hydrostatic bearing formed by
the flow of pressurized fluid over the sill 58 of the pad or face
56 and acting in the recess. Movement of the thrust ring in the
direction of arrow "D" tends to increase the flow of the fluid
passing over the trailing edge of the sill and results in the slight
tip of the button illustrated in FIG. 4. The effects of these ac-
tions are reflected in the force diagram illustrated in FIG. 5,
wherein the pressure buildup and dropoff across the pad area
is illustrated.

Because of these effects it is desirable to design the pad or
face so this undesirable tipping or tilting of the pad or face
area can be corrected.

More particularly FIGS. 8 and 9 illustrate the simplest but-
ton construction. Basically the construction includes the face
or pad area 56 having a recess 57 and a raised circular sill 58,
which are more clearly depicted in FIG. 9 which is a section
along line IX—IX of FIG. 8. The head portion 50 of the button
is larger than the skirt portion 51 which is received in pocket
36. Pressurized fluid is supplied through passage 38 (or 40),
screen 54 to the mouth 55 of the button unit 35. From inside
the skirt the fluid passes through orifice 59 into recess 57.
Thus, as pressure increases in passage 38, the pressure acting
against the runner provided by surface 34 will increase, sup-
porting the increased thrust loads acting on thrust ring 32
through the thrust plate assembly, etc.

Since the button is free to wobble and move axially in its
pocket, the fluid pressure from passage 38 (or 40) acting on
the bottom of the button will tend to urge the button toward
the thrust face or surface 34 for optimum clearance between
the sill and the thrust face or surface for the pressures involved
and best hydrostatic film support. By utilizing this arrange-
ment, the effects of both thermal and mechanical distortion
are greatly reduced. Also, the hydrostatic button bearings
can be "matched" with cooperating mechanical bearings for shar-
ing the thrust loading so that problems of over and under
compensation can be minimized. Having the sill raised centrally
on the pad area also tends to increase the hydrodynamic buildup
of pressure on the leading edge of the button pad area, result-
ing in a favorable tilt of the button. Use of an overbalanced
button (face force higher than hub force with recess at full
pressure) with a restriction or flow control to reduce recess
pressure provides a stable arrangement capable of compensat-
ing for changes in pressure profile over the face of the button
with changes in speed and oil viscosity. If the button separates
too far from the thrust plate, recess pressure drops and the
button recontacts. If the button tries to contact the thrust plate,
recess pressure increases and separates the two surfaces.

FIGS. 6 and 7 illustrate another pad configuration for the
button 35 with FIG. 7 being a section along lines VII—VII
of FIG. 6. In this design multiple sills are employed to lessen
the hydrodynamic effects referred to above. As can be seen in
FIGS. 6, 7, and 8, the face or pad includes an outer minor circular
sill 60 which includes a plurality of notches 61 communicating
with an inboard groove 62. This minor sill does not provide
support but next is a concentric outer major sill 63 followed by
a groove recess 64 and then a concentric inner sill 65 with an
annular recess 57 located centrally in the pad face. Only the
groove recess 64 is supplied with pressurized fluid through in-
terior passages 66 and oil entering the central recess 57 can
cress via passages 67 and 68 to drain. This arrangement tends
to change the force diagram, distributing it away from the
center area of the button and toward its outer diameter in a
ringlike pattern. Since the narrower "force ring" will tend to
have equal hydrodynamic pressure buildup at diametric op-
opposite sides, it will tend to stabilize the button and lessen its
tilt or tipping.

Obviously in the above arrangements the bearings 12
and 13 assist the buttons in stabilizing the thrust plate assembly
and provide additional support if the buttons fail to fully com-
pensate or over compensate for the axial thrust loadings, espe-
cially at high-swing angles in the unit. Of course, these bearings
absorb the radial component 31 of the oblique thrust forces
without assistance from the button bearings.

What is claimed is:
1. In combination with an axial piston-type hydraulic unit
having two ports for fluid inlet and outlet, a thrust plate as-
assembly rotatably mounted within the unit and having a plurality of connecting rods journaled in said thrust plate assembly, a hydrostatic thrust bearing system between the unit and said thrust assembly for thrust forces transmitted by the connecting rods to said thrust plate assembly comprising: a plurality of stationary pocket means in said unit adjacent to said thrust plate assembly and circumferentially disposed within the circular periphery of said assembly; a first common passage means in fluid communication with one port of said hydraulic unit commonly connecting a group of pocket means and a second common passage means in fluid communication with the other port of said hydraulic unit commonly connecting the remaining pocket means whereby one group of said pocket means will be connected to the inlet and the other group will be connected to the outlet of said hydraulic unit; a plurality of unitary individual button bearing means and each including a bearing pad and a stem portion, each of said stem portions being received in respective pocket means and being of a size smaller than the size of said pocket means to permit relative axial and tilting movement of said individual button means with respect to said unit, said bearing pad having a central recess in communication with the source of pressurized fluid in its associated pocket means and a raised circular sill area; and a cooperating flat ringlike runner surface mounted with said thrust plate assembly to rotate therewith, said surface disposed parallel to the bearing pads of the several button bearing means whereby pressurized fluid in said pocket means will urge its associated button bearing means toward said rotating runner surface for supporting thrust loads on said thrust plate assembly on a plurality of individual hydrostatic bearings.

2. The combination as defined in claim 1 wherein the thrust plate assembly is supported for rotation on roller bearings and the hydrostatic thrust bearing system cooperates with said roller bearings to compensate for thrust loadings.

3. The combination as defined in claim 1 wherein the pocket means are circular blind bores circumferentially disposed in the body of the hydraulic unit near the circular periphery of the thrust plate assembly whereby axial thrust loadings can be independently compensating by each button bearing means received in said pocket means.

4. The combination as defined in claim 3 wherein both groups of pocket means connected respectively to the inlet and the outlet are in a semicircular arrangement.

5. The combination as defined in claim 1 wherein the pad of each individual button bearing means includes an outer non-load bearing minor circular sill with bleed notches therein, an outer and inner circular sill having a pressurized circular recess therebetween and a center recess having a passage communicating to bleed fluid from said center recess.

6. The combination as defined in claim 1 wherein the group of pocket means commonly connected to the port acting as the inlet of the hydraulic unit are circumferentially disposed behind the pistons of the cylinders communicating with the inlet and the group of pocket means commonly connected to the port acting as the outlet of said hydraulic unit are circumferentially disposed behind the pistons of the cylinders communicating with the outlet.