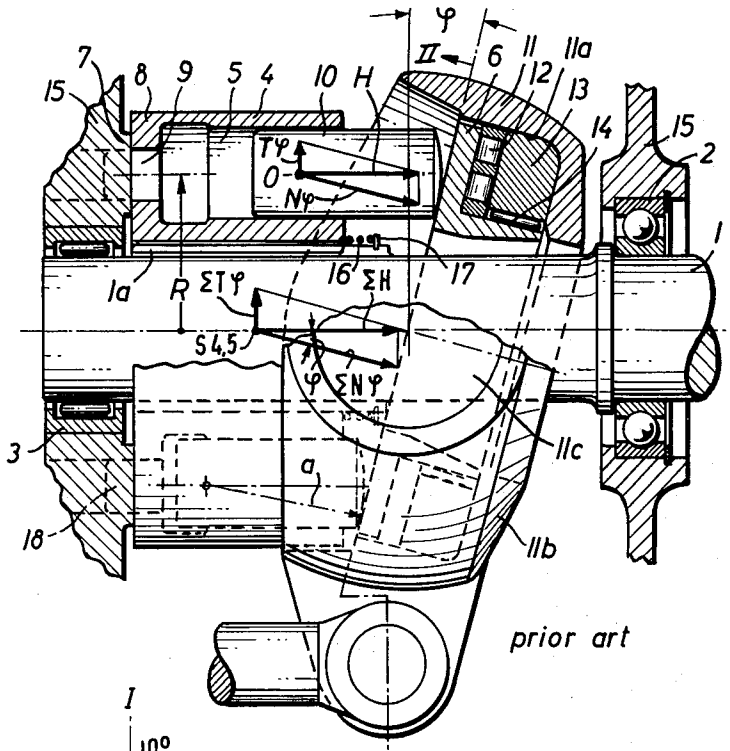
June 21, 1966 H. EBERT 3,256,782

PISTON FOR HYDRAULIC AXIAL PISTON UNITS WITH TILTABLE SWASH PLATE

Filed Feb. 17, 1964 3 Sheets-Sheet 1

Fig. 1



prior art

Fig. 2

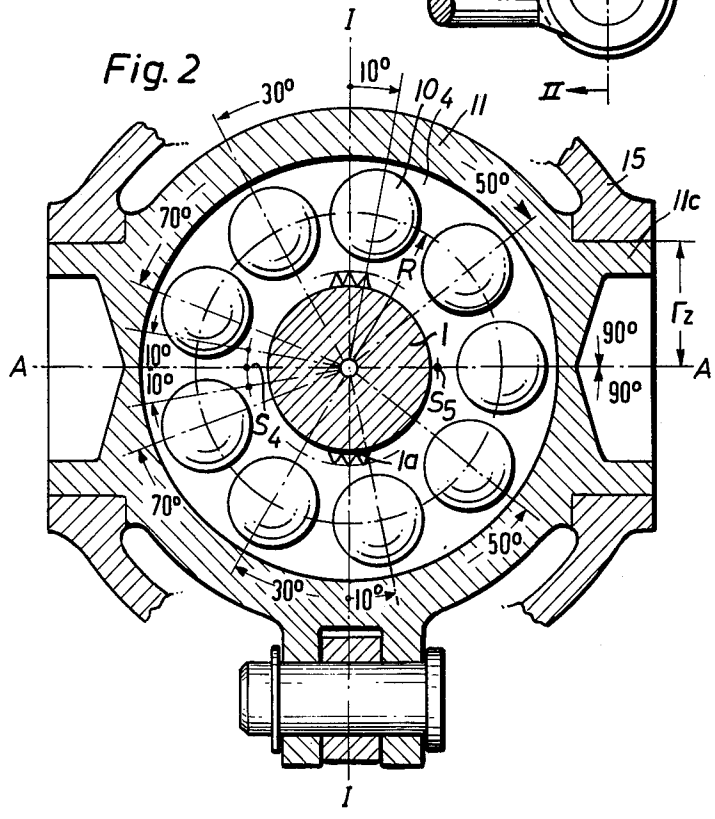
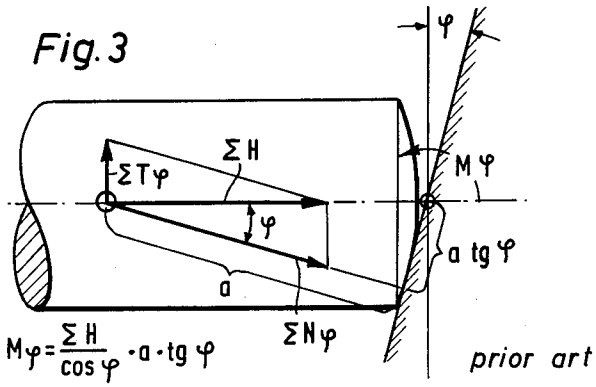
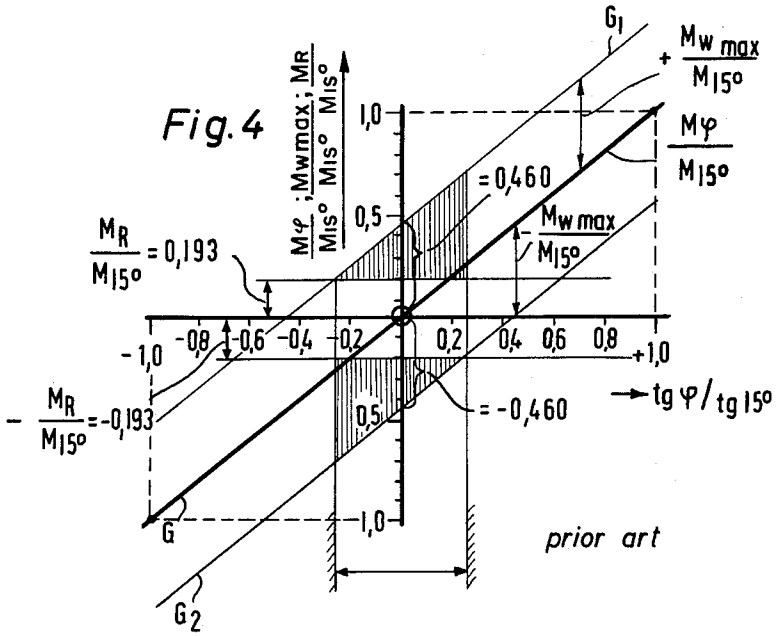


Fig. 3



prior art

Fig. 4



prior art

June 21, 1966 H. EBERT 3,256,782
PISTON FOR HYDRAULIC AXIAL PISTON UNITS WITH TILTABLE SWASH PLATE
Filed Feb. 17, 1964 3 Sheets-Sheet 3

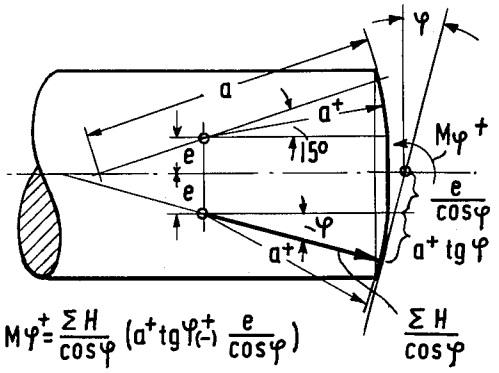


Fig.5

Fig.6

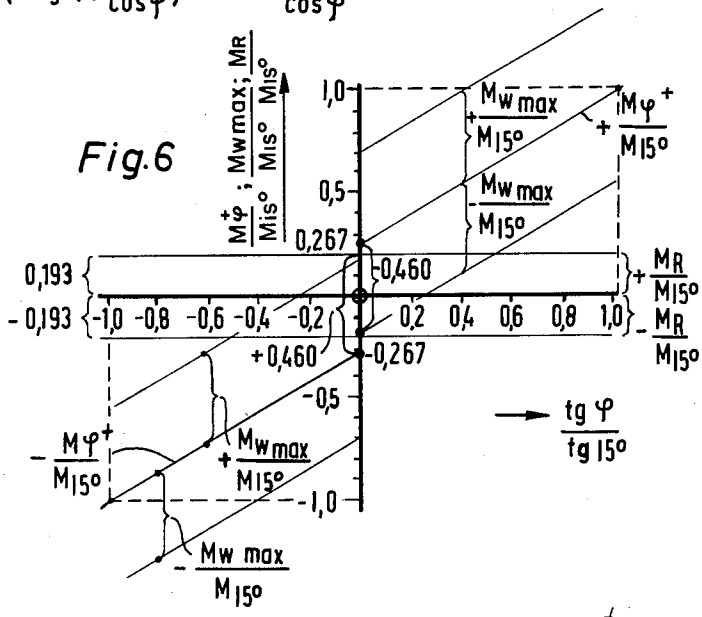


Fig.7a

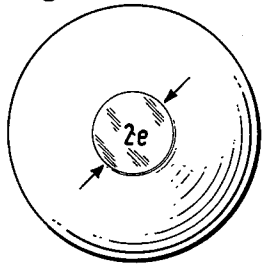
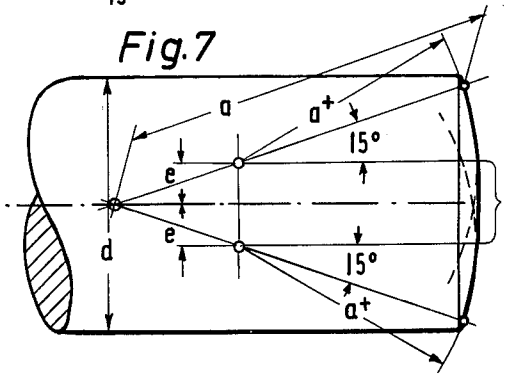


Fig.7



United States Patent Office 3,256,782
Patented June 21, 1966

3,256,782

PISTON FOR HYDRAULIC AXIAL PISTON UNITS WITH TILTABLE SWASH PLATE

Heinrich Ebert, Im Weller 2, Furth, Bavaria, Germany

Filed Feb. 17, 1964, Ser. No. 345,834
Claims priority, application Germany, Feb. 19, 1963, E 24,353
5 Claims. (Cl. 92—146)

The present invention relates to a piston for a hydraulic axial piston unit with tiltable swash plate and, more specifically, concerns the special design of pistons for multicylinder hydraulic axial piston units in which the pistons press directly upon a swash plate which in its turn is rotatably journalled in a bearing housing and together with the latter is tiltable about an axis perpendicular to the axis of rotation of the cylinder drum.

With heretofore known arrangements of this type in which generally the tilting axes are not only perpendicular to the axis of rotation of the cylinder drum but are also located in a common plane therewith, the pistons have always been so designed that that end thereof which acts upon the swash plate is a spherical cap the center point of which is located on the longitudinal axis of the otherwise cylindrical piston.

Regardless of whether or not this tilting axis is located in the plane of the swash plate, difficulties are encountered with the spherical design of the piston ends when small tilting angles φ or small piston strokes are encountered. These difficulties are due to the fact that piston forces acting upon the swash plate and the bearing housing therefor produce torques about the tilting axis which torques during the rotation of the pistons about the axis of rotation of the cylinder drum vary twice per rotating piston from a positive to a negative value. Due to these moments of changing from positive to a negative value and due to play in the adjusting means first there are caused very fast angular movements of the bearing housing as a result of which the adjusting mechanism is prematurely worn. As a result thereof, the adjusting mechanism is prematurely worn, especially in view of the always prevailing play between the individual adjusting elements. Similarly, also the tilting bearings and tilting pivots are prematurely worn in view of the tilting movements which while being small are effected in rather a quick succession so that the said tilting bearings wear out relatively quickly.

It is, therefore, an object of the present invention to provide a new design for the piston ends acting upon the swash plate in arrangements of the type involved, which will overcome the above mentioned drawbacks.

It is a further object of this invention to provide pistons for hydraulic axial piston units with tiltable swash plate, which will assure that even at a swash plate tilting angle of φ almost equalling zero no oscillating torques will be freely effective about the tilting axis of the swash plate.

It is a still further object of the present invention to provide an arrangement as set forth in the preceding paragraphs in which at all times a constant contact will be maintained between the piston stroke adjusting elements so that a wearing of the adjusting elements for the swash plate will not occur.

It is a further object of this invention to provide an arrangement as outlined above which will avoid the hammering noises which normally occur with arrangements of the above-mentioned type when the tilting angle of the swash plate approaches zero.

It is still another object of this invention to provide a piston for hydraulic axial piston units with tiltable swash plate which will greatly reduce the slip losses as they occur with heretofore known arrangements of the type involved.

These and other objects and advantages of the invention will appear more clearly from the following specification in connection with the accompanying drawings, in which:

FIG. 1 illustrates partly in section and partly in view an axial piston unit with which normally the above outlined difficulties are encountered which will be overcome by the present invention.

FIG. 2 is a sectional view indicated by line II—II on FIG. 1 looking at the piston ends when the angle φ thereof is zero, said section being taken along a plane passing through the tilting axis of the swash plate and perpendicular to the longitudinal axis of the drum supporting shaft.

FIG. 3 is a diagrammatic sketch for use in connection with the understanding of the formula for the tilting moment of the swash plate about the tilting axis thereof in connection with pistons of the heretofore known design.

FIG. 4 illustrates the conditions for the tilting moment in conformity with the tilting angle φ for heretofore known piston designs.

FIG. 5 is a diagrammatic sketch similar to that of FIG. 3 for explaining the formula for the tilting moment of the swash plate about the tilting axis thereof in connection with pistons designed in conformity with the present invention.

FIG. 6 represents a diagrammatic showing of the conditions for the tilting moment in conformity with the tilting angle φ for pistons according to the present invention.

FIG. 7 illustrates a piston according to the present invention.

FIG. 7*a* shows an end view of the piston of FIG. 7 as seen from the swash plate.

The piston according to the present invention intended for a hydraulic axial piston unit with tilting swash plate is characterized primarily in that that end face thereof which acts upon the swash plate represents a surface portion of a body of revolution, the arrangement being such that a plane passing through the axis of rotation of the piston intersects said surface portion along a line composed of two curved line sections forming an image to each other with regard to the axis of rotation of the piston and a central straight line section arranged tangentially to said curved line sections.

Referring now to the drawings in detail and FIGS. 1 and 2 thereof in particular, the arrangement shown therein represents an axial piston unit which comprises a shaft **1** rotatably journalled in bearings **2** and **3** and representing the power input or power output depending on whether the unit is used as hydraulic pump or motor. Mounted in shaft **1** and keyed thereto by a splined connection **1***a* is a cylinder drum **4** representing a cylindrical rotary body. Drum **4** is provided with a plurality of cylinders **5** the axes of which are located along a cylinder having a diameter of 2R and being coaxially arranged with regard to shaft **1**. These cylinders are uniformly distributed about the axis of shaft **1** and are open toward a swash plate **6** while being separated from a control valve **7** by a wall **8** with control slots **9** therein. Reciprocably arranged in said cylinders **5** are fluid operable pistons **10** which during operation of the unit are acted upon by pressure fluid in the cylinders so as to have their protruding ends pressing upon the swash plate. It will be appreciated that when under these circumstances the cylinder drum rotates while the swash plate is in a tilted position, the pistons carry out a reciprocatory movement while performing work. The swash plate represents a rotary body in the form of an axial bearing ring which is rotatably

journalled in bearing housing **11** for rotation about its axis.

The forces acting parallel to the axis of rotation of said swash plate are by means of rollers **12** conveyed to the bearing ring **13**, whereas those forces acting transverse thereto are absorbed by the needles **14**.

The bearing housing **11** which has a spherical outer contour is provided with a bore **11***a* having inserted thereinto the bearing ring **13**. For purposes of absorbing axial forces, this bore **11***a* has its rear end closed by an annular wall **11***b*. The bearing housing **11** is furthermore provided with two oppositely located trunnions **11***c* coaxial with the tilting axis of the swash plate. The axis of trunnions **11***c* is so arranged that it intersects the rotation axis of the swash plate and also the axis of rotation of drum **4** while being perpendicular to both of them.

In FIG. 1 the effective surface of the swash plate is located in a plane which contains the tilting axis. This, however, must not necessarily be so. The tilting axis may also be located ahead of the effective surface of the swash plate in which instance, as is well known, smaller tilting moments are obtained.

The trunnions **11***c* are tiltably journalled in housing **15** so that the bearing housing and together therewith the swash plate **6** may be tilted about the tilting axis thereof. The titlting angle is designated φ and represents the angle formed by the effective plane of the swash plate with a plane perpendicular to the longitudinal axis of shaft **1**.

Cylinder drum **4** which is axially easily displaceable on shaft **1** is for effecting the initial sealing pressed by a spring **16** against the sealing surface of the control valve **7**. The spring **16** is interposed between the adjacent end face of cylinder drum **4** and a spring ring **17** connected to shaft **1**.

The control valve **7** is provided with two kidney-shaped cutouts **18** at the separating surface between control valve **7** and cylinder drum **4** in a manner well known and shown for instance in U.S. Patents 2,687,049; 2,931,250 and 3,074,296. One of said kidney-shaped cutouts **18** is connected to the high pressure conduit of the respective hydrostatic unit, whereas the other kidney-shaped cutout is connected to the low pressure conduit for respectively controlling the inflow and outflow through the control slots pertaining to the respective cylinders and located in the walls separating the cylinder from the control valve, all in the manner described in the above-mentioned patents.

At any rate, the control is so effected that all pistons, the axes of which are in FIG. 2 located on the left-hand side of the plane I—I passing through the tilting axis and perpendicular thereto, are acted upon by pressure of the high pressure conduit, whereas all pistons the axes of which are located on the right-hand side of the plane I—I are under the pressure of the low pressure conduit.

The pistons which in the specific example shown in FIG. 2 number nine are so designed that the four pistons on the high pressure side as well as the five pistons on the low pressure side are located symmetrically with regard to the tilting axis. The radius vectors through the axis of the unit form with the plane I—I angles of 10°, 50°, 90° on one side and 30° and 70° respectively on the other side. The axial forces H exerted hydraulically by the individual pistons, which at the tilting angle $\varphi=0$ of the swash plate act directly upon the swash plate, may in this position of the pistons be replaced by two resultant axial forces ΣH_H and ΣH_N. One of these resultant forces, namely ΣH_H passes through the center of gravity S_4 of the four pistons on the high pressure side, whereas the other resultant axial force, namely ΣH_N, passes through the center of gravity S_5 of the five pistons on the low pressure side.

If the tilting angle φ equals zero, in this position of the pistons, the resultant forces of all pistons acting upon the swash plate in axial direction are so located that they will not produce a torque about the tilting axis A—A because

their centers of gravity S_4 and S_5 are located on the high pressure side as well as on the low pressure side in the plane of the tilting axis A—A. When however, the cylinder drum is moved out of its position of symmetry in one or the other direction, the centers of gravity S_4 and S_5 rotate with the cylinder drum in a similar way in one or the other direction. The greatest distances from the tilting axis will then be obtained when either the uppermost or the lowermost right-hand piston on the low pressure side is located precisely in the plane I—I without, however, changing over to the high pressure side. This situation will be encountered at a rotation which, generally expressed, with an uneven number z of pistons equals $90°/z$, and with an even number z' of pistons equals $180°/z'$.

Since consequently an even number of pistons over an uneven number of pistons results approximately in double the distances of the centers of gravity from the tilting axis of the swash plate, which fact in the present case is rather unfavorable and since an even number of pistons also brings about a relatively considerably greater fluctuation in the delivery, the following calculations are based on an uneven number z of pistons.

The greatest distances of the centers of gravity will in this instance then be obtained with $z=9$, at a rotation of the cylinder drum of $\pm 10°$ from the illustrated position of symmetry of the pistons. When the cylinder drum is turned further and further, these centers of gravity will each time after reaching their maximum positive or negative distance from the tilting axis of the swash plate skip from the high pressure side to the low pressure side and vice versa, while therein between passing through all values between the maximum positive and negative distance or also vice versa. If the pressures at both sides of plane I—I were even, the forces exerted by the pistons upon the swash plate would during this alternating cycle always balance each other with regard to their torque about the tilting axis because at equalized pressure the resultant of all pistons would pass through the center point of all pistons and this center of gravity would at any rate be located in the axis of rotation of the cylinder drum which axis of rotation in conformity with the above assumption is located in one and the same plane with the tilting axis of the swash plate. Since in operation the fluid pressure on one side of plane I—I, i.e. on the high pressure side, is higher than the fluid pressure on the other side of plane I—I, i.e. on the lower pressure side, from these conditions moments about the tilting axis will result which during the rotation of the cylinder drum and at the tilting angle of $\varphi=0$ will continuously vary from a positive maximum value to a negative maximum value or vice versa. The maximum magnitude of these alternating moments $M_{w_{max}}$ is defined for an uneven number z of pistons in conformity with the above assumption by the following equation:

$$M_{w_{-max.}} = \pm\left(\frac{z-1}{2}\cdot\frac{s_{z-1}}{2}\cdot p_H - \frac{z+1}{2}\cdot\frac{s_{z+1}}{2}\cdot p_N\right)\cdot F\cdot \sin\frac{90°}{z}$$

In this equation:

z designates an uneven number of pistons of a unit;

$s_{z-1}/2$ designates the distance between the center of gravity of the smaller number of pistons on the high pressure side from the axis of rotation of the drum;

$s_{z+1}/2$ designates the distance of the center of gravity of the greater number of pistons on the low pressure side from the axis of rotation of the cylinder drum;

p_H designates the fluid pressure on the high pressure side;

p_N designates the fluid pressure on the low pressure side;

F designates the surface of each individual piston.

In this equation, according to the laws concerning the center of gravity, the following substitution may be effected:

$$\frac{z-1}{2}\cdot\frac{s_{z-1}}{2} = \frac{z+1}{2}\cdot\frac{s_{z+1}}{2}$$

so that there will be obtained the following equation:

$$M_{w_{-max.}}=\pm\left(\frac{z-1}{2}\cdot\frac{s_{z-1}}{2}\cdot F\cdot\sin\frac{90°}{z}\right)(p_H-p_N)$$

These alternating moments about the tilting axis for $\varphi=0$ are overlapped by the moments $M\varphi$ which at a tilting angle φ of the swash plate are exerted by the pistons in the position of symmetry of FIG. 2 due to the fact that at the contact point of the piston with the swash plate, the latter can exert upon the piston only a force perpendicular to the plane of the swash plate when disregarding minor friction forces. When the piston surface is designed as calotte of a ball or in other words forms a portion of a spherical surface, and the center of said calotte is located on the axis of rotation of the piston, the normal force $N\varphi$ of each piston passes through the respective center of this ball in a direction perpendicular to the plane of the swash plate as will be evident from the upper portion of FIG. 1.

If H designates the hydraulic force exerted by the pressure fluid upon the piston, which force equals the product of fluid pressure p and piston surface F, so that $H=p.F$, the piston force H is by the presence of the swash plate and the piston cylinder wall split into two components $N\varphi$ and $T\varphi$. These components are for each piston:

$$N\varphi=H/\cos\varphi=p.F/\cos\varphi$$

and

$$T\varphi=H.\tan\varphi=p.F.\tan\varphi$$

Assuming a constant tilting angle, the individual components of all pistons are parallel to each other so that the resultant forces may be considered as having been exerted by a single piston the axis of rotation of which passes through the common center of gravity of the pistons of the high pressure side as well as of the low pressure side. This means that the resultant force may be considered as having been exerted by a single piston which for purposes of calculating the tilting moments $M\varphi$ may be located in FIG. 1 in the axis of rotation of the drum and which contacts the swash plate, as shown in FIG. 3 on a larger scale.

The moments $M\varphi$ exerted by a "resultant" piston will be obtained in conformity with the following equation, in which a designates the radius of the ball which forms the calotte:

$$M\varphi=\Sigma N\varphi.a.\tan\varphi=\Sigma H\cos/\varphi a.\tan\varphi$$

When $\cos\varphi$ is nearly 1, it is possible without incurring any material mistake to write:

$$M\varphi=\Sigma H.a.\tan\varphi$$

so that the equation

$$\frac{M\varphi}{M_{15°}}=\frac{\Sigma H\cdot a\cdot\tan\varphi}{\Sigma H\cdot a\cdot\tan 15°}=\frac{\tan\varphi}{\tan 15°}$$

will be obtained.

This equation is represented by the straight line G in FIG. 4 and wherein the abscissa represents $\tan\varphi/\tan 15°$ and the ordinate represents $M\varphi/M_{15°}$. This line extends from

$$M\varphi/M_{15°}=1.0 \text{ at } \tan\varphi/\tan 15°=1.0$$

to

$$M\varphi/M_{15°}=-1.0 \text{ at } \tan\varphi/\tan 15°=-1.0$$

In the graph, the units are without dimension because each dimensioned value is divided by another of the same dimensions so the graph is applicable to all sizes of hydraulic units. The selection of the moment and tangent at 15 degrees for the divisor is made because 15 degrees is about the maximum angle of tilt of the swash plate and the numeric values dealt with in the graph are thus of a convenient size.

The previously calculated values of M_{wmax} will have to be added to or subtracted from the said moment.

Referring to FIGURES 3 and 4 which show the prior art, in the graph of FIGURE 4 the line G represents the

torque exerted on the swash plate of the pump in various tilted positions of the swash plate. The torque is either positive to the right side of the graph or negative to the left side of the graph. This is a unidirectional torque exerted on the swash plate for the particular tilted position thereof.

As has been explained, in reference to FIGURE 2 the center of action of the forces of the pistons on each side of the axis of the pump that divides it between high and low pressure is through the center of gravity of the pistons on the respective side of the said axis. Since the pistons are individual elements, the center of gravity of the pistons on each side of the axis will shift first to one side of the tilting axis of the swash plate, then to the other side.

This will obviously impose upon the swash plate an oscillating moment which varies from a maximum in one direction to a maximum in the other direction as the cylinder barrel of the pump rotates. Since this moment varies from plus to minus and is acting on the swash plate, the resultant moment on the swash plate due to the last-mentioned oscillating moment and the moment developed thereon due to the pumping action will be the sum of the said moments.

In FIGURE 4 this is represented as the region measured vertically between the lines G1 and G2. To determine the moment acting on the swash plate at any angularly adjusted position thereof, the position is located along the abscissa and the limits on the torque acting on the swash plate are then found by finding corresponding points on lines G1 and G2 vertically aligned with the said point on the abscissa. The actual torque on the swash plate will vary at a rate determined by the speed of rotation of the pump and will, thus, shift continuously between lines G1 and G2.

The swash plate is supported on trunnions and these introduce a certain resistance to movement of the swash plate, which resistance is represented by the horizontal lines above and below the abscissa in FIGURE 4.

It will be evident, at this point, that when the moment on the swash plate reverses, the swash plate will only tilt when the limiting torques acting thereon overcome the resistance offered by the trunnions to movement of the swash plate.

Thus, in FIGURE 4 the shaded triangular areas in the graph represent those regions in the graph in which the limits of torque exerted on the swash plate are sufficient for moving the swash plate against the resistance of the trunnions. The graph of FIGURE 4 pertains to the prior art piston of FIGURE 3, which has a spherical end cap so that as the swash plate tilts from a negative angle through zero to a positive angle, the torque G due to the pumping action will vary uniformly through zero, and conditions illustrated in FIGURE 4 will be obtained.

Referring now to a particular example with nine pistons for which $s_{z-1}/2\cdot\sin 90°/z$ will equal 0.125R (R indicating the distance of the pistons from the axis of rotation of the drum),

$$\frac{M_{w_{max.}}}{M_{15°}}=\pm\frac{4\times 0.125RF(p_H-p_N)}{(4p_H+5p_N)Fa\tan 15°}$$

Assuming R to amount to $1.642d$, and a to amount to $1.55d$ (d indicating the piston diameter), the above equation will read:

$$\frac{M_{w_{max.}}}{M_{15°}}=\pm\frac{(4p_H+5p_N-9p_N)0.125\times 1.642dF}{(4p_H+5p_N)1.55dF\tan 15°}$$

This equation can be simplified to read:

$$\frac{M_{w_{max.}}}{M_{15°}}=\pm 0.495\left(1-\frac{9p_N}{4p_H+5p_N}\right)$$

Assuming $p_H=30p_N$ by way of example, the equation will finally read:

$$\frac{M_{w_{max.}}}{M_{15°}}=\pm 0.496\left(1-\frac{9}{125}\right)=\pm 0.460$$

With this value there will be obtained the parallel lines G_1 and G_2 spaced from the line G by the distance $M_{w_{max}}/M_{15°}=\pm 0.460$. Taking into consideration that the trunnions **11***c* of the swash plate cause a frictional torque M_R in the bearing housing **15**, the magnitude of this friction torque is indicated by the equation:

$$M_R=(\Sigma H/\cos\varphi)\mu r_Z$$

in which μ indicates the friction coefficient of the trunnion with regard to the bearing and in which r_Z indicates the trunnion radius. Assuming for purposes of example that $\mu=0.06$, and that $r_Z=1.34d$, and dividing the above equation by $M_{15°}$ and furthermore assuming that cos φ is nearly 1.0, there will be obtained the following equation:

$$\frac{M_R}{M_{15°}}=\pm\frac{H0.0804d}{Ha\tan 15°}=\pm\frac{0.084d}{1.55.d\times 0.268}=\pm 0.193$$

When plotting these two lines in FIG. 4 parallel to the abscissa at a distance therefrom of $M_R/M_{15°}=\pm 0.193$, it will be noted that within the range of tan φ/tan $15°=+0.25$ to -0.25, i.e. in the tilting angle range between $\varphi=+3.8°$ and $\varphi=-3.8°$, the oscillating torques exerted about the tilting axis by the pistons may exert harmful effects inasmuch as in this connection, oscillating moments may be effective in the plus and minus range.

The tilting angle range within which free alternating moments may occur becomes even greater when the tilting axis (not as assumed above when setting up the above equations) is not located in the effective plane of the swash plate but when looking from the piston, is located in front thereof. If the tilting axis is located at a distance *a* (radius of the ball calotte) in front of the swash plate while, however, as before it is located in the same plane as the axis of rotation of the cylinder drum—desired in order to reduce the tilting moments—free alternating torques would result over the entire tilting angle range. With arrangements of this type, the tilting moments $M\phi$ for each angle would become zero so that, for instance, in a diagram according to FIG. 4, the lines represented by the $M_{w_{max}}/M_{15°}$ values ($M_{15°}$ refers to the tilting axis through the axis of rotation of the cylinder drum in the plane of the swash plate) will be parallel to the abscissa and parallel to the friction torques $M_R/M_{15°}$ and thus will never intersect. This means that over the entire tilting angle range there would prevail free oscillating alternating torques of the magnitude

$$\pm(M_{w_{max}}/M_{15°}-M/_RM_{15°})$$

so that the above mentioned defects would be encountered over the entire tilting angle range.

Turning now to FIGURES 5 and 6, FIGURE 5 shows a piston end according to the present invention and FIGURE 6 shows the moments acting on the swash plate.

As before, in FIGURE 6 the center inclined line on each side of the graph represents the moment exerted on the swash plate in respective tilted position thereof due to the pumping action. This moment is always positive on one side of the graph and is always negative on the other side. Also, as before, the inclined lines above and below the said center inclined line show the limits within which the torque on the swash plate varies due to the oscillating moment which comes about due to rotation of the cylinder barrel and periodic shifting of the center of gravity of the pistons on the high and low pressure sides of the pump across the tilting axis of the swash plate.

In FIGURES 5 and 6, however, due to the flat end on the piston, there is an abrupt off-set in the said inclined center line so that this center line does not pass through the zero point of the axis in the same manner as line G of FIGURE 4.

The amount of off-set of the said inclined center line representing the pumping torque on the swash plate is so large that when the oscillating torque is imposed thereon, as represented by the upper and lower inclined lines, there is no point along the graph where the torque acting on the swash plate will overcome the resistance of the trunnions in both directions of the torque.

It will, therefore, be seen that the prior art arrangement, as set forth in FIGURES 3 and 4, has a condition of tilt of the swash plate on opposite sides of its zero position when the swash plate is subjected to an oscillating torque in excess of the resisting moments of the trunnions in both directions of tilting about the tilt axis of the swash plate. The swash plate will, thus, wobble on its trunnions and the trunnions will wear rapidly, while at the same time the pistons will tend to take an erratic movement and interfere with smooth flow of fluid through the pump and there will, likewise, develop high stresses between the swash plate and the pistons, and the pump will also be noisy.

It is furthermore evident in the arrangement of FIGURES 5 and 6 that there is no point during the tilting movement of the swash plate that the oscillating torque thereon is sufficient for overcoming the trunnions' resistance in both tilting directions of the swash plate. With this improved arrangement there is thus no wobble of the swash plate and the pistons take the uniform action, promoting uniform fluid flow through the pump and eliminating high stress and hammering noises.

The amount of off-set between the two points where the center inclined line of the graph in FIGURE 6 intersects the vertical axis of the graph is determined by the size of the flat on the end of the position. Even a slight tilting of the swash plate from a zero position will suddenly shift the point of engagement of the pistons therewith to the periphery of the flat on the end of the piston.

In the prior art, on the other hand, the point of contact of the end of the piston with the swash plate progresses only gradually across the central axis of the piston as the swash plate tilts through zero.

The feature of the flat on the end of the piston is, thus, effective for producing the results illustrated in FIGURE 6.

Still further, the size of the flat is such that the piston will rotate when engaging the swash plate except when the swash plate is in an exactly zero tilt position. At this time, the flats will engage the swash plate but since the flats present substantial areas there is no extreme pressure built up in the region where the pistons engage the swash plate and no undue wear of the swash plate will take place even if the hydraulic unit is standing under pressure. In practice, the zero tilted position of the swash plate is seldom achieved and is usually of only brief duration.

More specifically, if in contrast to the above, that end of the pistons which engages the swash plate is, instead of being designed as a ball calotte, designed as a surface portion of a body of revolution in conformity with the present invention, while said surface portion intersects a plane passed through the axis of rotation of the piston along a line comprising two sections of a circle the radius of which is a^+ and the center points of which are located on a line perpendicular to the axis of rotation of the piston and spaced from said axis of rotation by a distance *e*, and thus if these circular lines are interconnected by a common tangent as indicated in FIG. 5, the tilting moments plotted in FIG. 5 will be obtained according to the present invention. These tilting moments are expressed by the formula

$$M^+\varphi=\frac{\Sigma H}{\cos\varphi}\left(a^+\tan\varphi\pm\frac{e}{\cos\varphi}\right)$$

In this formula the plus sign applies to the positive tilting angle and the negative sign applies to the negative tilting angle. If, as shown in FIG. 5 $a^+=a^-e/\sin 15°$, the

same value as previously obtained for $M_{15°}$ will be obtained for $M^{+}_{15°}$, since thus there will be obtained

$$M^{+}_{15°}=\frac{\Sigma H}{\cos 15°}\left(a^{+}\tan 15°+\frac{e}{\cos 15°}\right)$$
$$=\frac{\Sigma H}{\cos 15°}\left(a\cdot\tan 15°-\frac{e\tan 15°}{\sin 15°}+\frac{e}{\cos 15°}\right)$$
$$=\frac{\Sigma H}{\cos 15°}\left(a\cdot\tan 15°-\frac{e}{\cos 15°}+\frac{e}{\cos 15°}\right)$$
$$=\frac{\Sigma H}{\cos 15°}(a\cdot\tan 15°)=M_{15°}$$

The same applies to the tilting angle $\varphi=-15°$ so that also in this instance $M^{+}\varphi/M_{15°}$ will for

$$\tan\varphi/\tan 15°=\pm 1.0$$

yield the values ± 1.0 as before. If, however, tan $\varphi/$ tan 15° becomes zero, $M^{+}{}_{0°}/M_{15°}$ will be $\pm e/(a\tan 15°)$. This means that the tilting moment $M^{+}{}_{0°}$ will in contrast to the ball calotte end not be zero but will have a more or less positive or negative value depending on the magnitude of e and depending on whether the movement toward φ=zero comes from a positive or a negative tilting angle φ.

In contrast to the course of the tilting moments as illustrated in FIG. 4, the design of the piston end in conformity with the present invention will yield a course of the tilting moments of $M^{+}\varphi/M_{15°}$ over tan $\varphi/\tan_{15°}$ in conformity with FIG. 6, i.e. an approximately straight line course between the points $M^{+}\varphi/M_{15°}=1.0$ at tan $\varphi/\tan 15°=1.0$ and $M^{+}\varphi/M_{15°}=+e/(a\tan 15°)$ at tan $\varphi/\tan 15°=0$, and furthermore between points $M^{+}\varphi/M_{15°}=-e/(a\tan 15°)$ at tan $\varphi/\tan 15°=0$ and $M^{+}\varphi/M_{15°}=-1.0$ at tan $\varphi/\tan 15°=-1.0$. When tilting the swash plate from a positive tilting angle φ to a negative tilting angle φ, or vice versa, the tilting moment $M^{+}\varphi/M_{15°}$ changes abruptly from a value $+e/(a\tan 15°)$ to a value $-e/(a\tan 15°)$, i.e. from a positive to a negative value or vice versa.

These values of the tilting moment are overlapped, in a manner described in connection with FIG. 4, by the alternating moments $M_{w_{max}}/M_{15°}$ during the continuous rotating movement of the cylinder drum out of the position of symmetry of the pistons. In this instance, the lines representing said alternating moments $M_{w_{max}}/M_{15°}$ extend parallel to the straight lines $M^{+}\varphi/M_{15°}$, namely at a distance $M_{w_{max}}/M_{15°}=\pm 0.46$, when assuming that $z=9$ and $p_H/p_N=30$.

As long as at tan $\varphi/$ tan 15°=0, the values

$$M^{+}\varphi/M_{15°}-M_{w_{max}}/M_{15°}$$

on the right-hand side of FIG. 6 exceeds the frictional torque $M_R/M_{15°}=0.193$, and as long as the value

$$M^{+}\varphi/M_{15°}+M_{w_{max}}/M_{15°}$$

on the left-hand side with regard to FIG. 6 is smaller than the frictional torque $M/M_{15°}=-0.193$, no oscillating alternating moments can act as free outer alternating moments on the tilting axis so that the above mentioned harmful effects do not occur.

In order to arrive at the above mentioned results, the following condition must be met:

$$M^{+}\varphi/M_{15°}\geqq M_{w_{max}}/M_{15°}-M_R/M_{15°}$$

or when inserting thereinto the above formula, the requirement is expressed as follows:

$$\frac{e}{a\tan 15°}\geqq$$
$$\frac{\left(\frac{z-1}{2}\right)(p_H-p_N)}{\left(\frac{z-1}{2}p_H+\frac{z+1}{2}p_N\right)a\tan 15°}\cdot\frac{s_{z-1}}{2}\cdot\sin\frac{90°}{z}-\frac{\mu r_Z}{a\tan 15°}$$

so that there will be obtained the following requirement:

$$e\geqq\left(\frac{1-p_N/p_H}{1+\frac{z+1}{z-1}p_N/p_H}\right)\cdot\frac{s_{z-1}}{2}\cdot\sin\frac{90°}{z}-\mu r_Z$$

As will be evident from the above, the spacing e of the center points from the axis of rotation of the piston (see FIG. 5) may become zero, namely if μ and especially r_Z are selected of a corresponding magnitude. It has been proved, however, that to this end too large diameters of the trunnions would be required which could be realized only if the tilting axis were provided at a considerable distance from the effective plane of the swash plate. Such an arrangement, however, results in too large a space requirement for the tilting of the swash plate, which is in most instances prohibitive.

When in the last formula substituting

$$\frac{1-p_N/p_H}{1+\frac{z+1}{z-1}p_N/p_H}$$

for K, assuming $z=9$, the following table for various values of p_H/p_N will be obtained:

$p_H/p_N=$	5	10	15	20	25	30	00
K=	0.6550	0.810	0.868	0.900	0.920	0.935	1.000

For a standard hydrostatic axial piston unit with nine pistons each having a diameter d, the distance R of the pistons from the longitudinal axis of the cylinder drum may be assumed to amount to $R=1.642d$ and the radius of the trunnions may be assumed to amount to $r_Z=1.34d$. With $s_4\sin 10°=0.125R$ and with $K=0.935$ assuming a maximum pressure ratio of $p_H/p_N=30$ and a friction coefficient $\mu=0.06$, as it is well justified in view of the prevailing conditions, the following numerical values will be obtained for e:

$$e\geqq 0.935\times 0.125\times 1.642d-0.06\times 1.34d$$

or

$$\frac{e}{d}\geqq 0.192-0.0804=\underline{0.0116}$$

For the sake of completeness it may be mentioned that the above obtained requirement exists not only when the tilting axis through the axis of rotation of the cylinder drum is located in the effective plane of the swash plate but in general for any position of the tilting axis in front of or behind the effective plane of the swash plate. Furthermore, this requirement exists also regardless of whether or not the tilting axis is located in one and the same plane with the axis of rotation of the cylinder drum.

It is, however, advisable to arrange the tilting axis of the swash plate at least in one and the same plane with the axis of rotation of the cylinder drum inasmuch as this arrangement at any rate yields the smallest e-values independently of the position of the tilting axis with regard to the effective plane of the swash plate.

However, the smallest tilting moments over the entire tilting range are obtained, when the tilting axis of the swash plate is located at a distance a^{+} in front of the effective plane of the swash plate, when viewed from the pistons, and in one and the same plane with the axis of rotation of the cylinder drum. Referring, as before, $M_{15°}$ to the tilting axis of the swash plate passing through the axis of rotation of the cylinder drum in the effective plane of the swash plate, the mean tilting moment $M\varphi/M_{15°}$ will be constant over the entire tilting angle range and will amount to $\pm e/(a\tan 15°)$ so that for the values of $\pm M_{w_{max}}/M_{15°}$ there will be obtained lines parallel to the abscissa in a diagram according to FIG. 6 and spaced from said abscissa by a distance:

$$+\frac{e}{a\tan 15°}\pm M_{w_{max.}}/M_{15°}\text{ or }-\frac{e}{a\tan 15°}\pm M_{w_{max.}}/M_{15°}$$

If the spacing of the tilting axis of the swash plate from the effective plane of the swash plate were still larger than a^+, for instance $\underline{a}$, (the radius of the ball as before), the mean tilting moment $M\varphi/M_{15°}$ would, at tan φ/tan 15°=0, again amount to $\pm e/(a \tan 15°)$, but $M\varphi/M_{15°}$ would be zero if tan φ/tan/15°=±1.0. Consequently, in a diagram according to FIG. 6, the values $M\varphi/M_{15°}$ plotted over tan φ/tan 15° would yield a line declining from $+e/(a \tan 15°)$ at tan φ/tan 15°=0 toward the right to the value zero at tan φ/tan 15°=1.0 when tan φ/tan 15° is positive, and would yield a line ascending from a value of $-e/(a \tan 15°)$ at tan φ/tan 15°=0 to the value zero at tan φ/tan 15°=1.0 when tan φ/tan 15° is negative.

The lines representing the values of $M_{w_{max}}/M_{15°}$ would in this instance already at tan φ/tan 15°=0 drop below the curve of $-M_R/M_{15°}$ and extend beyond the curve $+M_R/M_{15°}$ so that again no harmful alternating moments would occur at tan φ/tan 15°=0, but these alternating moments would occur over the entire remaining range. In order to avoid this, e would have to be increased correspondingly at the distance a^+ from the effective plane of the swash plate retained. This increase, however, may have the result that the swash plate can no longer be tilted by larger angles inasmuch as it lacks a definite point of contact with the pistons. In addition thereto it may be mentioned that the line of intersection of the piston end face with a plane extending through the axis of rotation of the piston does not necessarily have to be composed of two circular lines of constant radius the center points of which are located at the same distance from the axis of rotation of the piston while the connecting line between said center points is perpendicular to said axis. Instead, this line of intersection may consist of lines curved in any desired manner, i.e. lines with variable radius of curvature. It is, however, important that these curved lines tangentially engage a plane perpendicular to the axis of rotation of the piston at a distance e from said axis and that the smallest radii of curvature of said lines are not selected too small in view of the Hertz' specific pressure between the piston and the swash plate, Hertz' specific pressure being a customary expression indicating the stress of a material as results, for instance, when a ball under pressure presses on a plate, and more in general, when curved surfaces are pressed against each other.

As will be evident from the preceding description, for the heretofore customary pistons according to FIG. 3, as indicated in FIG. 3, the following value for the momentum is obtained:

$$M\varphi=\Sigma H/\cos\varphi\ a \tan\varphi(=\Sigma N\varphi\ a \tan\varphi)$$

In contrast thereto, for the piston according to the present invention (FIG. 5) the following value applies:

$$M^+\varphi=\Sigma H/\cos\varphi(a^+ \tan\varphi \pm e/\cos\varphi)$$

Mφ is, therefore, the total tilting moment exerted by all pistons according to the embodiment of FIG. 3 in the position of symmetry of the pistons with regard to the tilting axis according to FIG. 2, at a tilting angle φ°. In contrast thereto, Mφ^+ is the tilting moment exerted by all pistons in the embodiment according to FIG. 5 of the present invention in the position of symmetry of the piston with regard to the tilting axis according to FIG. 2 at a tilting angle of φ°.

When rotating the pistons or the cylinder drum respectively out of the position of symmetry of the pistons according to FIG. 2, the center of gravity S_4 as well as the center of gravity S_5 moves out of that plane which is formed by the tilting axis and the axis of rotation. Their largest spacings from this plane then amount to the spacing of the center of gravity from the axis of rotation of the cylinder times sin (90°/z), i.e. with z equalling 9:times sin (90°/9)=sin 10°. These spacings may be found for S_4 in FIG. 2 (the points on the radii 10° with regard to the tilting axis A—A).

Inasmuch as in the center of gravity of the piston on one pressure side as well as on the other pressure side (p_H, p_N) the resulting force of the pistons located on one or the other pressure side acts parallel to the axis of rotation, during rotation of the cylinder drum oscillating moments are obtained the maximum values $M_{w_{max}}$ of which equal the resulting piston force of one or the other pressure sides times the largest spacing of the respective center of gravity of the pistons from the tilting axis.

The resulting force is determined by the number of pistons located on one of the pressure sides times the surface of one piston times the pressure. This force is furthermore to be multiplied with the respective lever arm of this force at which the same is effective with regard to the tilting axis, and which amounts to $S_4\times$ sin 10° or S_5 sin 10° in the example according to FIG. 2. When considering that with a positive moment of one pressure side there is present a negative moment of the other pressure side, there is finally obtained after combining the individual values the formula:

$$M_{w_{max}}=\pm(4S_4p_H-5S_5p_N)F\cdot\sin 10°$$

This oscillating moment $M_{w_{max}}$ is superimposed upon Mφ or M$^+\varphi$.

From the above it will be evident that pistons designed in conformity with the present invention and employed in hydrostatic axial piston units will not bring about any oscillating moments about the tilting axis of the swash plate even when the swash plate tilting angle φ equals zero. At all times the piston stroke adjusting elements will remain in contact with each other so that the elements of the adjusting mechanism of the swash plate will be prevented from prematurely wearing out. In addition thereto, a fast oscillating movement of the swash plate is avoided and, therefore, its pivots and pivot bearings will have a long life.

Due to the design of the pistons according to the present invention also the hammering noises normally occurring in such angle ranges are eliminated. Such hammering noises are not only due to the impact of the adjusting elements upon each other in view of the play which always exists between these elements, but they are also due to the lifting off of the pistons from and their subsequent impact upon the swash plate in view of the heretofore prevailing very fast oscillating movements about the tilting axis of the swash plate, especially at the low pressure side.

In addition thereto, the specific design of the pistons will, during the first few moments of the passage of the piston from the high pressure side to the low pressure side, prevent the displacement of a more or large quantity of fluid from the high pressure side to the low pressure side. Such a displacement was heretofore caused by the oscillating movement of the swash plate and normally results in high slip losses.

The specific design of the pistons according to the present invention will, therefore, in addition to avoiding harmful noises in the critical angle range also bring about a decrease in slip losses, thereby improving the volumetric degree of efficiency of the unit.

Furthermore, due to the measures suggested by the present invention, the heretofore encountered wear of the piston and the swash plate at small tilting angles is avoided. Such wear was due to the fact that the point of contact between the piston and the swash plate at small tilting angles was located within the so-called "friction circle of the piston." In this way, at best, differences in speed between the piston and the swash plate could be compensated for only by corresponding sliding movements between the piston and the swash plate under very high Hertz' specific pressures (ball against plate) so that considerable wear was unavoidable. The above mentioned differences in speed between the piston and the swash plate occur even at a minor inclination of the axis of rotation of the swash plate with regard to the axis of rotation of the cylinder drum and also at a minor un-

avoidable eccentricity of one of these axes relative to the other.

In contrast thereto, the design of the piston ends in conformity with the present invention will bring about that the contact points of the pistons with the swash plate are, even up to the smallest tilting angles, located outside the "friction circle of the piston." The diameter of said friction circle amounts to approximately $0.1d$, inasmuch as the distance between the said contact points and the axis of the piston can never be smaller than e, and inasmuch as $2e$ is always in excess of $0.1d$ as may be seen from the above mentioned numerical examples. Also differences in speed between the piston and the swash plate which might occur during the operation of the unit are compensated for by rotary movements of the pistons about their axes of rotation and by corresponding rolling movements of the pistons on the swash plate, which in heretofore known arrangements would have been possible only at larger tilting angles.

Merely at tilting angles φ which are absolutely zero, a contact between the pistons and the swash plate occurs within the said so-called "friction circle." In this instance, however, the pistons and the swash plate are in contact with each other over a circular area the diameter of which equals $2e$. If in this connection the axis of rotation of the swash plate is eccentric with regard to the axis of rotation of the cylinder drum, the swash plate and the pistons will also in this instance slide upon each other. However, since the contacting area is plane and finite, this contacting area is no longer subjected to a high Hertz' specific pressure but is subjected to much smaller specific pressures. Moreover, a situation in which the tilting angle φ is absolutely zero occurs very seldom, especially with automatic speed control, and if it occurs, it will last for a very short period of time only.

It is, of course, to be understood that the present invention is, by no means, limited to the particular construction shown in the drawings but also comprises any modifications within the scope of the appended claims.

What I claim is:

1. A piston for a hydraulic axial piston unit with a tiltable swash plate, said piston having its one end face which engages said swash plate convex toward the swash plate and formed as a surface of rotation the intersecting line of which with a plane containing the axis of rotation of the piston comprises two curved lines on opposite sides of the said axis of rotation of the piston and being mirror images of each other and located within the diametral limits of the piston, said lines meeting a plane forming a part of said one end face of the piston and perpendicular to the axis of rotation of the piston at a circular region of the plane concentric with said axis of rotation of the piston, the centers of curvature of all regions of said lines being respectively located on the same side of the axis of rotation of the piston as the pertaining said line, the radius of curvature of each said line at every region therealong being at least as large as the diameter of the piston.

2. A piston according to claim **1** in which the two curved lines are circular arches the centers of which are located on a line perpendicular to and intersecting the axis of rotation of the piston and are positioned on the line a predetermined distance from the said axis of rotation, and said plane forming a part of said one end face of the piston being tangent to said circular arches.

3. A piston according to claim **2** in which said predetermined distance is smaller than one-fourth of the piston diameter.

4. A hydrostatic axial piston system having cylinder drums each with an uneven number of pistons reciprocable therein and also having a high pressure side and a low pressure side, said hydrostatic axial piston system furthermore including swash plates adapted to be engaged by said pistons and also including pivot means pivotally supporting said swash plates and furthermore including bearing means for supporting said pivot means, each of said pistons having one end face for contact with the respective swash plate, said end face having its marginal surface portion formed by an annular curved surface area representing a portion of the outer surface of a body of revolution the axis of revolution of which coincides with the longitudinal axis of said piston while the center of curvature of said curved surface is laterally spaced from said piston axis by a distance defined by the formula

$$e \geqq \left(\frac{1 - p_N/p_H}{1 + \frac{z+1}{z-1} p_N/p_H} \right) \cdot \frac{s_{z-1}}{2} \cdot \sin \frac{90°}{z} - \mu_Z r_Z$$

in which p_H represents the pressure of the working fluid at the high pressure side,

p_N represents the pressure of the working fluid at the low pressure side,

z indicates an uneven number of pistons in said cylinder drum,

$s_{z-1/2}$ stands for the distance of the center of gravity of $z-1$ consecutive pistons in one and the same cylinder drum from the axis of rotation of the latter,

μ_Z indicates the coefficient of friction of said pivot means and said supporting bearing means therefor,

r_Z indicates half the diameter of the pivot in said supporting bearing means.

5. An arrangement according to claim **4**, in which the radius of curvature a^+ said curved surface is defined by the formula $a^+ = a - e/\sin \alpha$ in which $a = d/2 \sin \beta$ and in which d represents the diameter of the pistons of one and the same drum and α and β are angles within the range of from 13 to 17°.

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