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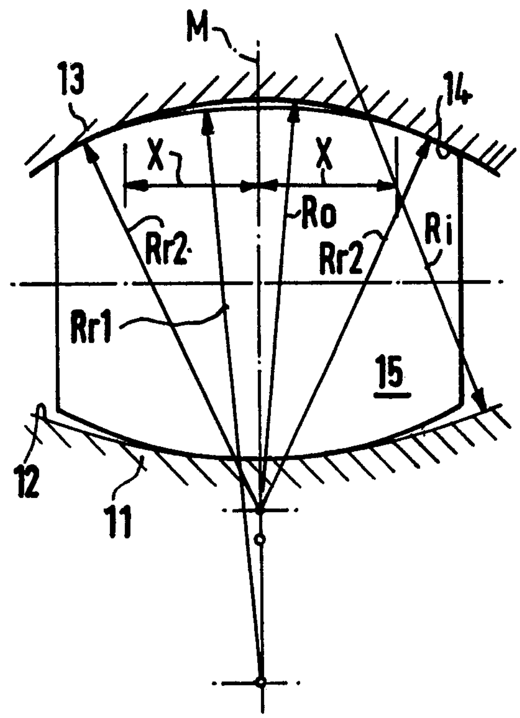
(54) **Spherical roller bearing**

(57) A spherical roller bearing comprises inner and outer raceways (12, 14) and a plurality of rollers (15). Each raceway (12, 14) is formed as the rotation of an arc of a circle R_o, R_i) about an axis. The external surface of each roller (15) is formed as the rotation of another arc about an axis, which said another arc is composed of at least two different curves (R_{r1}, R_{r2}), all such that in use of the bearing there is a resultant frictional moment on each roller causing it to be skewed to a skew angle which is positive.

FIG.19c.



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Certain of the mathematical formulæ appearing in the printed specification were submitted in formal form after the date of filing.
This print embodies corrections made under Section 117(1) of the Patents Act 1977.

FIG. 1.

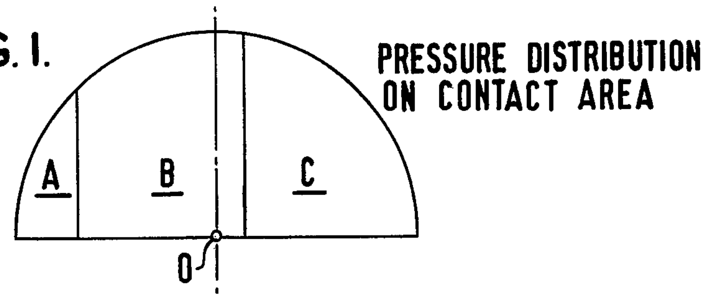


FIG. 2.

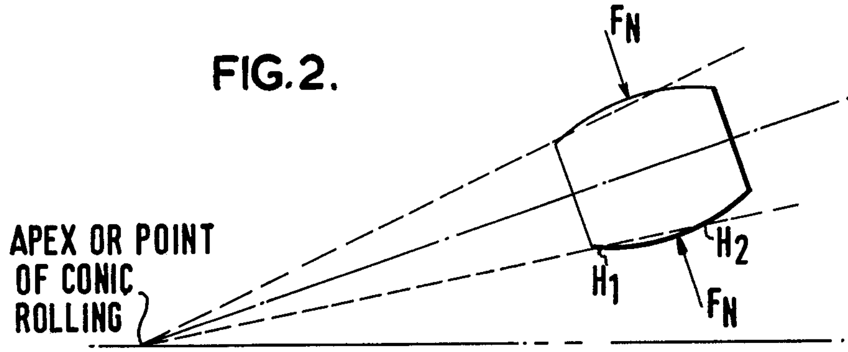


FIG. 3.

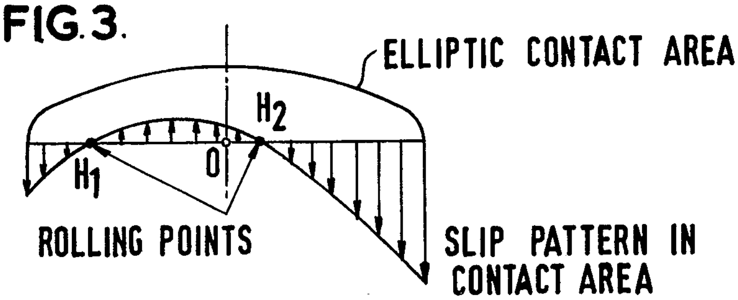


FIG 4.

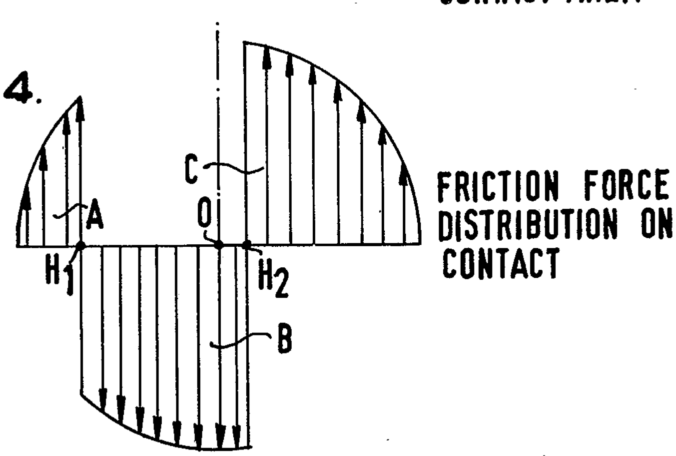


FIG. 5.

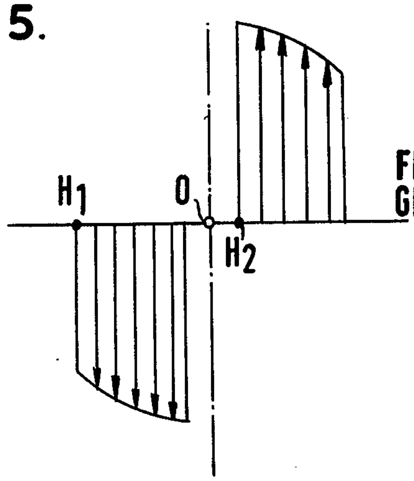


FIG. 6.

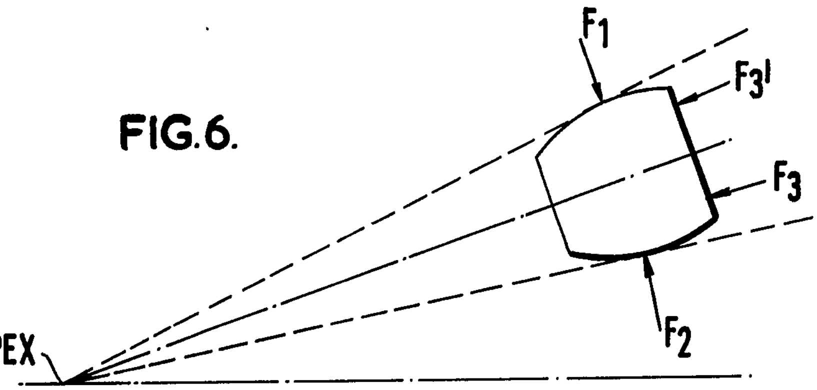


FIG. 7.

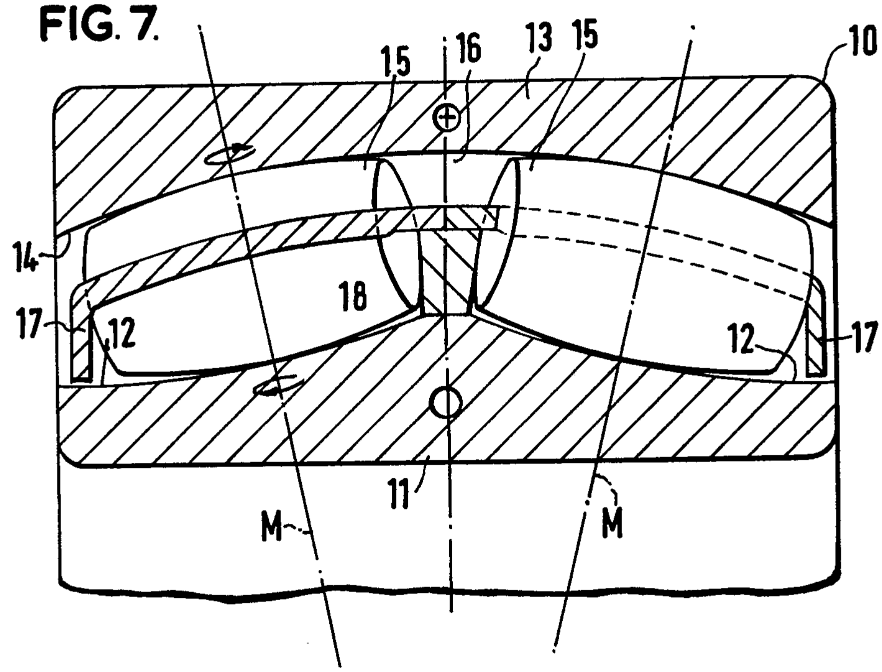


FIG.8.

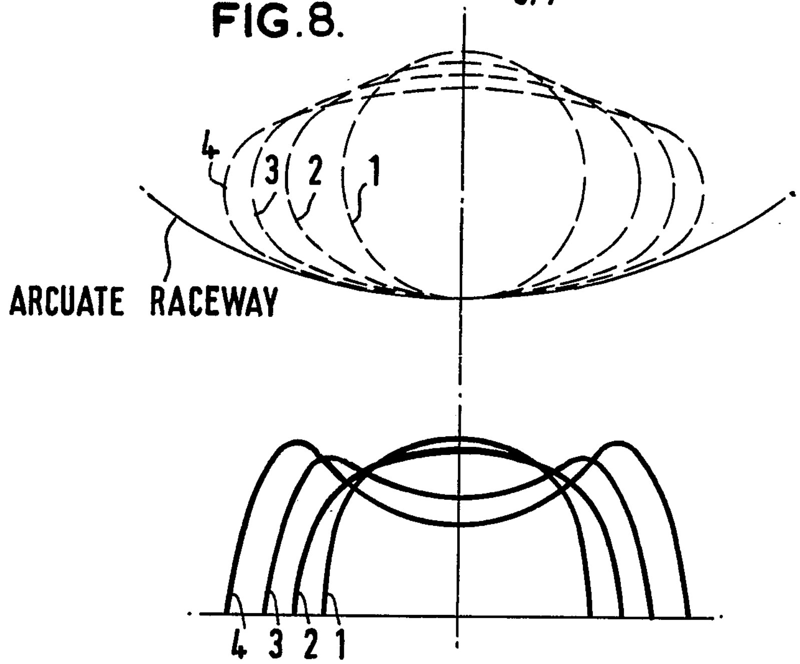


FIG. 9.

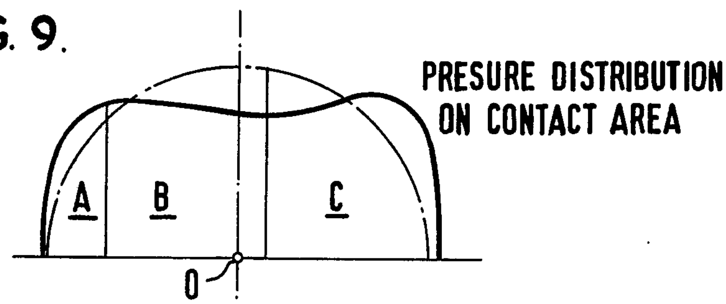


FIG. 10

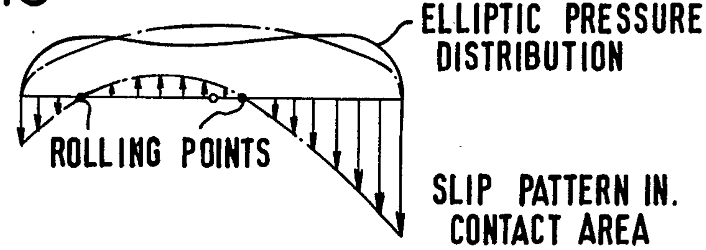


FIG.11.

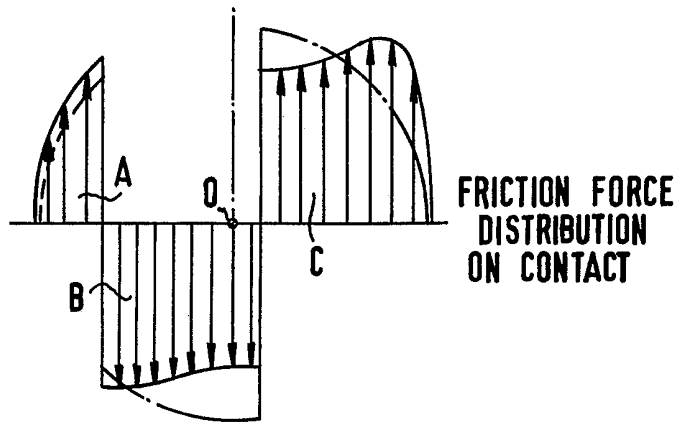


FIG 12.

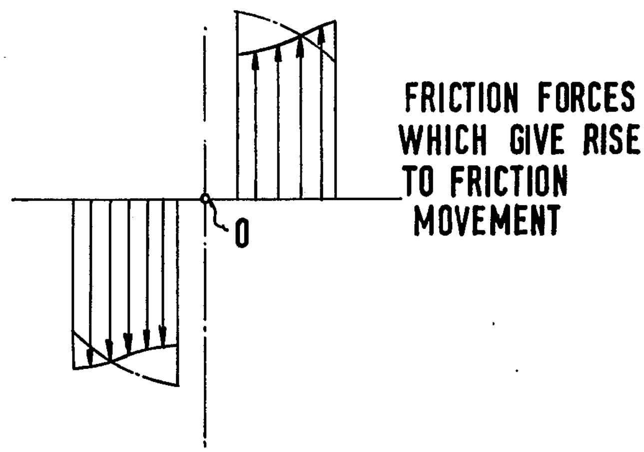


FIG.13.

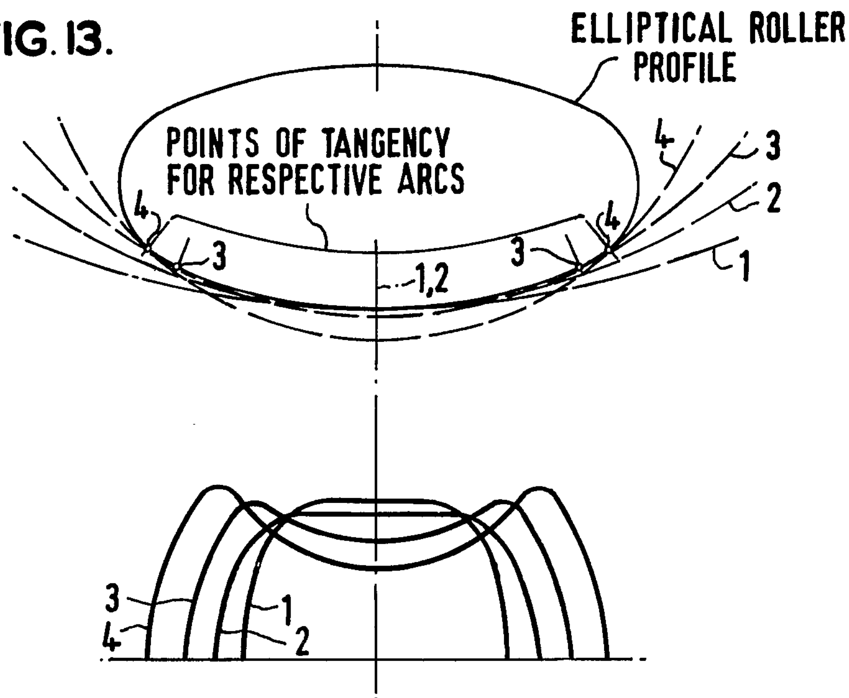


FIG. 13a

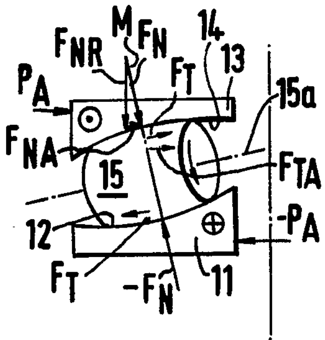


FIG. 14a

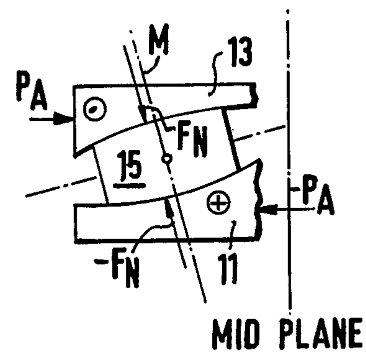


FIG. 15a.

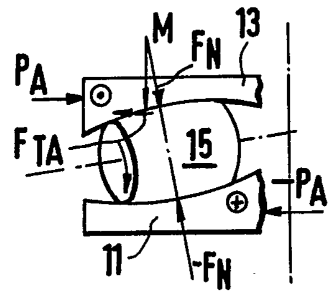


FIG. 13b

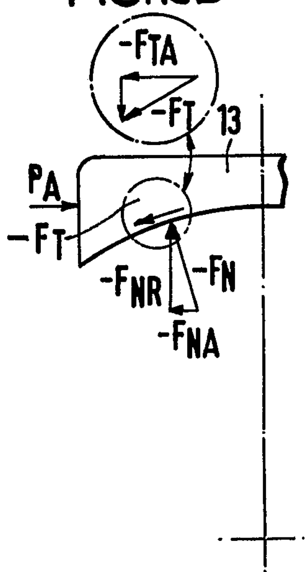


FIG. 14b.

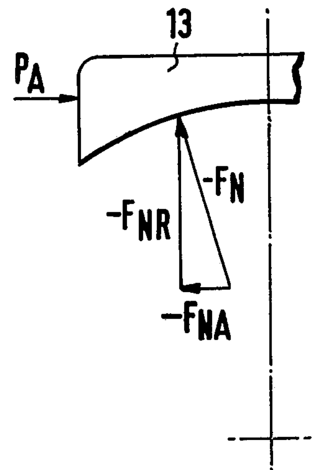


FIG. 15b

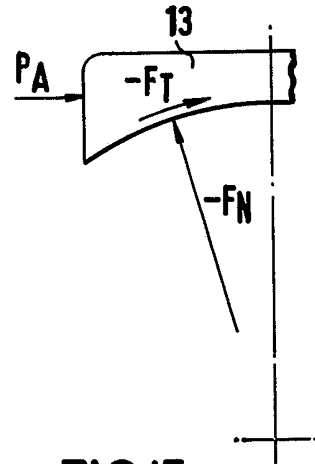


FIG. 13c.

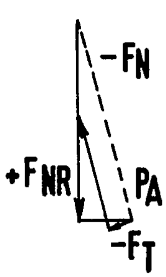


FIG. 14c.

FIG. 15c

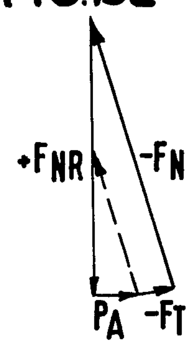


FIG. 13d.

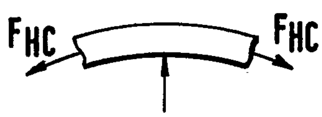


FIG. 14d.

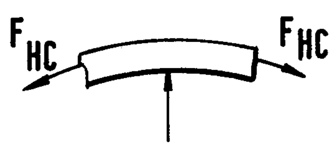


FIG. 15d.

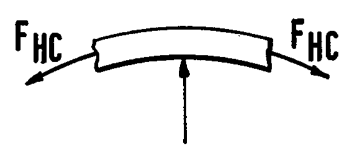


FIG. 16a.

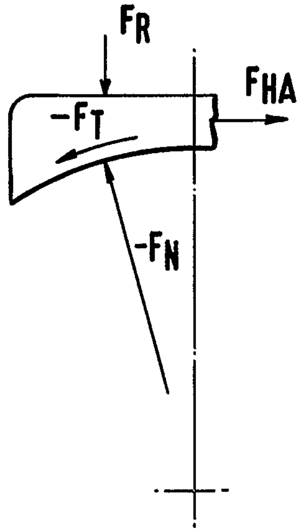


FIG. 17a.

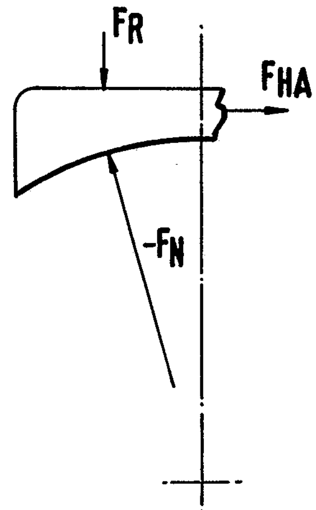


FIG. 18a.

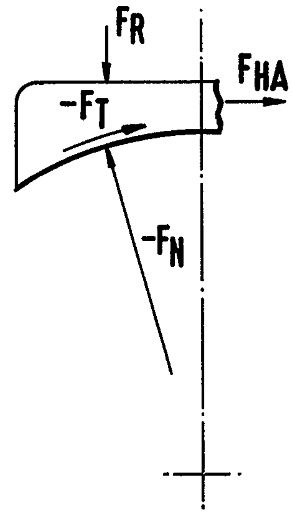


FIG. 16b.

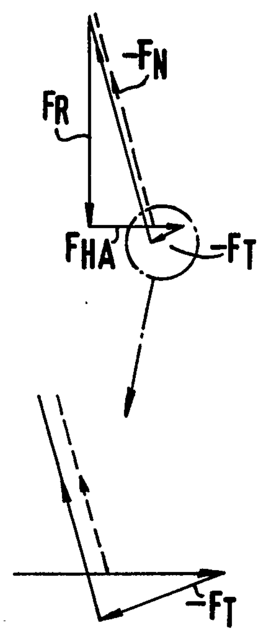


FIG. 17b

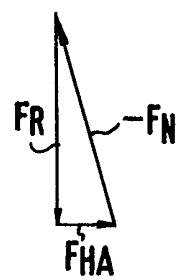


FIG. 18b.

FIG.19a.

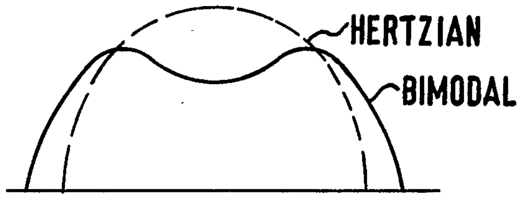


FIG.20a.

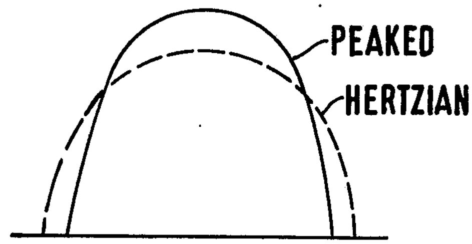


FIG.19b.

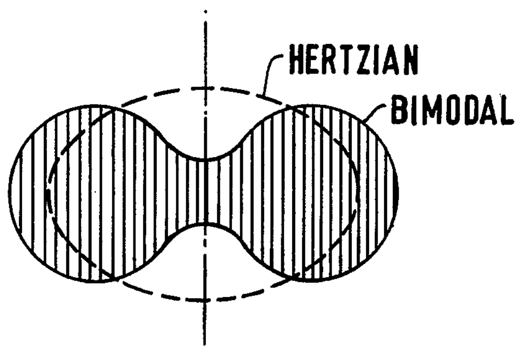


FIG.20b

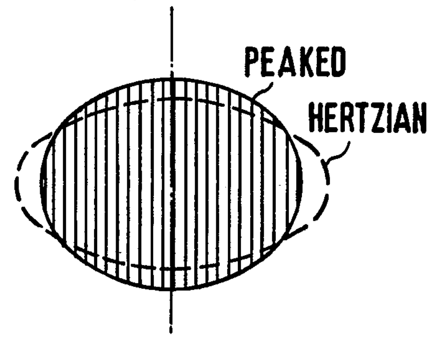


FIG.19c.

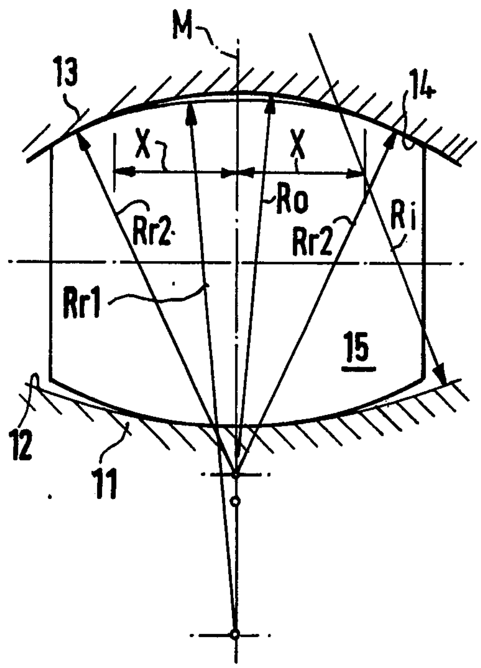
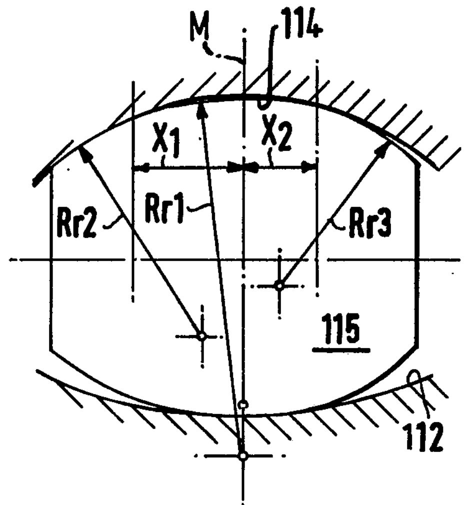


FIG.20c.



SPECIFICATION

Spherical roller bearing

The present invention relates to spherical roller bearings and particularly to structures for effecting roller skew control.

A skew angle is the angle of deviation of the rotational axis of the roller from a plane passing through the roller and in which lies the rotational axis of the bearing. The skew angle can be positive, negative or zero.

The skew angle of a roller is positive when the resultant parallel to the axis about which an arc of a circle is rotated to form an inner raceway of the friction force components acting on the roller at its contact points with a raceway is so directed that it increases the resultant along or parallel to that axis of the forces acting on the roller to the surface of the roller at the same contact points.

The concept of controlling roller skew is not new per se. This concept and the theory involved are fully discussed in Kellstrom et al U.S. Patent No. 3,990,753 for ROLLER BEARINGS COMPRISING ROLLERS WITH POSITIVE SKEW ANGLE issued November 9, 1976, incorporated herein by reference and referred to herein as "Kellstrom Patent". In the Kellstrom Patent the cause of roller skewing in a spherical roller bearing is shown to be due to residual pivotal moments arising in the contacts with the raceways. Roller skew of small magnitude in a direction defined as positive, was shown to have beneficial effects on the bearing friction, heat generation and ultimately on fatigue life. The Kellstrom Patent describes various techniques for controlling the pivotal moments to achieve positive skew. These techniques all involved raceway changes. For example, one technique involved adjusting the friction coefficient through surface roughness control. Another involved changes in the raceway conformities and a third skew control technique involved provision of relief areas in one or both raceways. In all of these techniques the roller profile was conventional. Even though it has been found that the techniques in the Kellstrom Patent are all feasible and may be effectively utilized to achieve the desired positive skew control, it may be difficult as a practical matter to control them all sufficiently accurately in manufacture to assure the desired slightly positive skew attitude of the loaded rollers in all bearings over a wide range of load, speed and lubrication conditions as explained in more detail below. The present invention provides an alternate and somewhat more versatile means of achieving positive skew control by selectively varying the roller profile in a predetermined manner while maintaining a constant radius of curvature for the inner and outer raceways.

In the raceway profiling or recessing technique of the Kellstrom Patent it has been found that if the geometric relationships between the rollers and the raceways change during operation of the bearing under load, the pressure distribution is altered and the theoretically designed static condition to produce the desired small positive skewing of the rollers or even negative skew. Both of these conditions are obviously undesirable. The relative geometric changes can occur for example, in a double row self-aligning spherical roller bearing where the inner and outer rings move relative to one another to accommodate for changes in operating conditions and thus change the force relationships. Further the technique of the Kellstrom Patent of selectively varying the coefficients of friction of the raceways has limitations because to the presently relatively limited choice of bearing materials. Moreover, controlling friction coefficients by surface roughness is a difficult manufacturing technique. Furthermore, surface roughness changes during the life of the bearing by reason of wear which of course changes the skew control. Also, friction is strongly influenced by the degree of elastohydrodynamic film, thereby impairing the effectiveness of surface roughness as a means of skew control.

In accordance with the present invention, the roller profile is selectively varied and the geometric relationships between the roller and the raceways remain constant even during asdjusting movement of the raceways relative to one another in a self-aligning bearing. It has also been found that the principle of the present invention compensates for adjustments in the bearing to take up normal internal clearances and also geometry changes which occur when the bearing is subjected to varying load conditions. In summary, the present invention provides a skew control technique wherein the geometric relationships between the roller and raceways remain the same either during misalignment or under load. Accordingly, the desired moment relationship to provide the small positive skew remains essentially unchanged.

The principle underlying the specially profiled roller of the present invention is that the pressure field in the two raceway contacts can be tailored to suit the needs of roller skew control even though the raceway profile remains arcuate and thus it can still satisfy the requirement that the contact geometry is not influenced by bearing axial misalignment. This can be best illustrated by the following discussion directed to the pressure distribution and force analysis in a conventional spherical roller bearing.

When an arcuately profiled roller is loaded against an arcuately profiled raceway of lesser curvature (larger radius of curvature), the contact normal or perpendicular pressure distribution is elliptical in profile, the maximum pressure occurring on the line of centers of curvature, the pressure distribution also being symmetrical about this line, (see Fig. 1).

In a significant class of practical spherical roller bearings, (those with rollers symmetrical about a plane perpendicular to the axis of rotational symmetry of the roller) design considerations prevent the tangents to the rollers at the point of initial contact between the rollers and raceways from passing through the bearing apex (point of conic rolling). When the bearing is under load, and elastic distortion of the surfaces of the contacts has taken place to establish a "footprint" of finite width, the deformed contact profile can only intersect a line from the apex at two points, H_1 and H_2, (see Fig. 2). When the bearing is in motion these two points represent regions in the contact "footprint" where the one surface is truly rolling on the other. All other regions of the contact above the line are slipping with respect to each other in one direction, and the regions below the line are slipping in the opposite direction, (see Fig. 3). Slip, of necessity induces friction forces on the surface in the directions of slip, (see Fig. 4). The actual position of the apex line and its intersection points H_1 and H_2 must be such that the resultant tangential friction force is zero. (This disregards secondary considerations of cage friction and lubricant squeeze film friction losses that modify this equilibrium statement slightly). Graphically, the areas A and C together must equal area B in Fig. 4.

For convenience, the magnitudes of the slip friction forces in Fig. 4 are shown proportional to the normal or perpendicular contact forces as they would be under Coulomb friction conditions. Other friction/normal force relationships can be assumed without invalidating the overall theses.

Although the friction forces are in translational equilibrium, it is obvious that the asymmetry of H_1 and H_2 can produce a lack of moment balance about the center of contact "O". The magnitude of the resultant friction spin moment about "O" can be assessed graphically by subtracting balanced areas as shown in Fig. 5, resulting in a net clockwise moment for the case illustrated. A similar analysis of the contact of the same roller with the other raceway typically leads to a net spin moment in the opposite direction. The actual behaviour of the roller is thus a response to the difference between the spin monents arising at the two raceway contacts.

The foregoing arguments have considered the axis of the roller to be co-planar with the axes of the raceways. Once unequal friction moments arise, the response of the roller is to skew in the direction of the dominant contact friction moment. A further slip velocity is then developed at each contact in the direction of the apex line. These additional sliding velocities then modify the direction of the slip friction and, due to the change in orientation of the planes of principal curvature of the rollers, also modify the pressure distribution at the contact and thus the magnitude of the forces shown in Fig. 4 and thus alter the net friction moment in each contact. Whether or not an equilibrium skew angle exists where the friction moments are equal and opposite is dependent entirely on the geometry, loading and lubrication conditions prevailing at the two roller contacts.

Analysis has shown that conventional bearings of this type either have roller equilibrium at negative skew angles or are unstable in the negative direction with the roller skew attitude limited by the cage pockets.

Another class of self-aligning roller bearing utilizes rollers which are not symmetrical about a plane perpendicular to the axis of rotational symmetry of the roller, so that tangents to the rollers at the nominal point of contact pass through the bearing apex, (see Fig. 6) and consequently the Coulomb sliding friction forces depicted in Fig. 4 are in both translational and moment balance, and thus no skewing moments arise at such contacts. However, the geometry of such bearing prevents the normal forces F_1, F_2 from the two roller contacts being collinear and thus a third contact force F_3, $F_{3'}$ is required to provide roller equilibrium. This reaction force F_3, $F_{3'}$ is provided by a flange attached to one raceway or the other. In either configuration contact with the flange produces a sliding velocity component which gives rise to tangential friction force on the roller end which in turn induces a skewing moment on the roller, again in the negative direction. Whether this negative skewing force can establish an equilibrium negative roller attitude depends not only on the raceway contact but also on the geometry of the roller end and the flange. The absence of a resultant spin moment in the asymmetrical contacts described above is only true for the assumed simple Coulomb friction conditions. More realistic friction force relationships have a viscous term which is velocity sensitive and when these friction relationships are considered, analysis shows that an asymmetric field of the type shown in Fig. 4 is again developed and skewing moments arise. Thus even the asymmetric roller class of bearing suffers from unwanted negative skew tendencies.

The above discussions outline the cause of roller skew and show that in current bearings the resultant friction moment from all the roller contacts (either two or three) causes an undesirable negative roller skew. However, in each case, one contact produces a net skew moment in the desired positive direction, and if this positive skew moment can be enhanced with respect to the negative moments, equilibrium can be achieved at a preselected small positive roller skew attitude.

The friction moment in a contact may be modified by varying either the magnitude of the slip forces themselves or by changing their distribution with respect to the center of contact "O". The former is primarily achieved by deliberately increasing the effective coefficient of friction at one contact; and the latter primarily by redistributing the normal contact forces which give rise to the friction forces. It is this latter approach which is the subject of this invention.

As described earlier, arcuate profiles in contact give rise to elliptical normal force distributions. This system constraint can be removed by making one of the contacting surfaces have a varying profile curvature. Perhaps the simplest example of such a profile is an ellipse although the feasibility is not restricted to a single class of curves. The only practical constraints are continuity of slopes (no lines of infinite curvature) to avoid local stress concentrations; and manufacturability considerations.

From a stress redistribution point of view, the varying arcuate profile may be applied to either of the raceways or to the roller. Overall bearing performance will dictate the most appropriate choice. Consider the typical spherical roller bearing configuration shown in Fig. 7. In this case the outer raceway contact develops the positively directed skew moment and the inner raceway the negative skew moment (see the Kellstrom Patent). Positive skew of the rollers could be achieved either by increasing the outer raceway moment or decreasing the inner raceway moment or by a combination of the two actions. In this bearing, self alignment properties dictate that the outer raceway be spherical. Any profile modifications to this raceway would disturb the sphericity and also be subject to a position error of up to 3° with respect to the roller, as the outer ring is misaligned in service.

Profile adjustment can be made to the inner raceway (see the Kellstrom Patent) such that the width of the pressure profile and hence the negative skewing moment is reduced. However, the maximum contact pressure is increased. Also, as loading directions vary, the axial position of the roller on the inner raceway varies, thus leading to position errors of the inner race profile with respect to the roller.

If the profile change is made to the roller, in the manner described herein, then the desired skew moment equilibrium can be achieved by beneficial adjustment of the normal force distributions in both contacts, and consequently in the resultant frictional slip fields.

By way of example, consider the normal stress distributions resulting from the loaded contact of rollers of differing elliptical profiles with an arcuate raceway of a specified radius of curvature (Fig. 8). Curve No. 1 is the special case of an ellipse with equal axes, a circle, which gives rise to the familiar elliptic pressure distribution. Curve No. 2 is that of an ellipse whose maximum radius of curvature is the same as that of the contacting raceway. This curve gives a flat centered pressure distribution. Curves No. 3 and No. 4 illustrate increasing bimodality of the pressure distribution as the ellipticity is increased. Referring back to the sequence of Figs. 1, 3, 4 and 5, the construction is repeated in Figs. 9 to 12 so that one can see that the widening 'footprint' associated with the greater roller ellipticity will give rise to a greater skew moment about the contact center line. However, if the two raceways were of a common curvature, both raceway friction moments would be enhanced to a similar degree and so this condition alone is not sufficient to achieve the skew control objective.

In Fig. 13 the effect of varying the radius of curvature of a raceway loaded against an elliptical roller profile is shown to influence the contact pressure distribution. Thus, if the inner raceway is made with a somewhat greater radius of curvature than the outer raceway radius, the desired difference in contact pressure distributions can be obtained and consequently roller skew control achieved. A particular benefit of this approach as highlighted in Fig. 13 is that the maximum stress at the inner raceway (always the most severely stressed and thus fatigue sensitive) is not sharply peaked, but can remain well distributed, and the tendency towards a bimodal stress field is confined to the outer raceway where stresses are inherently lower due to the conforming nature of that contact.

In the above example, the changes in pressure profiles alone have been considered. Of course, the translational equilibrium of the slip forces must also be taken into account. The relocations of the rolling points H_1 and H_2 are relatively minor within the range of practical profile changes, but even so, their displacements tend to augment the desired moment adjustment at the respective contacts.

In the example, where the roller contacts the interior surface of a sphere, the roller profile radius of curvature decreases towards the roller end, or in other words, the contact region of the roller profile is located near the intersection of the curve with the minor axis of the ellipse. The illustrated pressure profiles of Figs. 8 and 13 are symmetrical; the center of curvature of the raceway being assumed to lie on the minor axis of the ellipse. This is not a prerequisite.

In a bearing configuration wherein the rollers contact the exterior surface of sphere, the rollers being hourglass shaped, the pressure profile adjustments discussed above can be achieved by utilizing a roller profile in which the radius of curvature increases towards the roller ends. Such a profile exists adjacent the intersection of an ellipse with its major axis.

In one aspect the invention provides a

spherical roller bearing having inner and outer raceways and a plurality of rollers arranged to roll on both raceways, each raceway being formed as the rotation of an arc of a circle about an axis, the external surface of each roller being formed as the rotation of another arc about an axis, which said another arc is composed of at least two different curves, all such that in use of the bearing there is a resultant frictional moment on each roller causing it to be skewed to a skew angle which is positive as herein defined.

Each roller may have a datum plane extending through it and perpendicular to its axis of rotational symmetry, the radius of curvature of the said arc of the inner raceway may be greater than the radius of curvature of a curve of the said another arc of the external surface of the roller intersecting the datum plane, and the latter radius of curvature may be greater than the radius of curvature of the said arc of the outer raceway, so that each roller, in use of the bearing, may produce unimodal and bimodal pressure distributions with the inner and outer raceways respectively.

Each roller may have a datum plane extending through it and perpendicular to its axis of rotational symmetry, and the said another arc of the external surface of roller may include a curve on each side of the datum plane having a radius of curvature smaller than the radius of curvature of a curve of the said another arc intersecting the datum plane.

Each roller may have a datum plane extending through and perpendicular to its axis of rotational symmetry mid-way along the length of the roller.

Two adjacent curves of different radii of curvature of the said another arc of the external surface of each roller may intersect such that a tangent can be drawn common to both curves.

Each roller may have a datum plane extending through it and perpendicular to its axis of rotational symmetry, and each roller may be symmetrical about its datum plane.

In another aspect, the invention provides in a bearing having inner and outer members with inner and outer raceways spaced apart to define a space therebetween, a complement of rolling elements rotatable about their axes in the space between the raceways, the improvement wherein at least one rolling element has a profile with a predetermined, varying curvature from a transverse reference datum extending through the axis of the rolling element, said inner and outer raceways having curvatures of constant radii of curvature, said rolling element profile producing a pressure distribution under load resulting in a residual frictional moment producing forces counteracting the externally applied load, said rolling elements having a skew angle when the axis of a rolling element and the axis of the bearing are not coplanar and said skew angle being positive when the friction force components arising in each contact between said rolling element and raceways and acting on said rolling element in the axial direction, are so directed that they are co-directional to the axial component of the normal contact force acting on said rolling element in the same contact.

Said inner raceway profile may have a radius of curvature greater than said rolling element profile adjacent said datum and said outer raceway profile may have a radius of curvature less than said rolling element profile adjacent said datum to provide a profile relation establishing unimodal and bimodal pressure distributions with said inner and outer raceways respectively.

Said rolling element profile curvature may include radii of curvature on opposite sides of said datum which are smaller than the radius of curvature of said rolling element profile adjacent said datum. Said reference datum may be located at the longitudinal median of the rolling element. Adjacent different radii of curvature of said rolling element profile curvature may merge together at common tangents. Said smaller radii of curvature of said rolling element profile curvature may be of the same magnitude or said smaller radii of curvature of said rolling element may be of different magnitudes.

Said rolling element profile curvature may include radii of curvature on opposite sides of said datum which are greater than the radius of curvature of said rolling element profile adjacent said datum.

In a further aspect, the invention provides in a bearing having inner and outer members with inner and outer members with inner and outer raceways spaced apart to define a space therebetween, a plurality of rolling elements rotatable about their axes in the space between the raceways, the improvement wherein at least one rolling element has a profile with a predetermined, variable curvature from a transverse reference datum extending through the axis of the rolling element, said inner and outer raceways having curvatures of constant radii, said rolling element profile producing a pressure distribution under load resulting in a residual frictional moment producing forces counteracting the externally applied load.

In the accompanying drawings:

Figures 1, 3, 4 and 5 illustrate the general sliding pattern and friction forces in an angularly oriented roller under load;

Figure 2 is a schematic showing of an angularly oriented, loaded roller of a spherical roller bearing showing the points of pure rolling;

Figure 6 is a view similar to Fig. 2 showing the flange reaction forces and the raceway forces on the roller;

Figure 7 is a sectional view through a typical double-row spherical roller bearing;

Figure 8 is a schematic showing the normal stress distributions resulting from the loaded contact of rollers of differing ellipticity with an arcuate raceway of a specified radius of curvature;

Figures 9 to 12 inclusive are views similar to Figs. 1, 3, 4 and 5 showing the greater skew moment about the contact center line as the roller ellipticity is increased;

Figure 13 is a schematic view showing the effect of varying the radius of curvature of a raceway loaded against an elliptical roller profile particularly as it influences the contact pressure distribution;

Figures 13a, 14a and 15a are fragmentary views schematically illustrating certain forces which are developed in axially-loaded bearings operating with their rollers disposed at positive, zero and negative skew angles respectively;

Figures 13b 14b and 15b are simplified free body diagrams illustrating the relationship of the forces developed in the bearings illustrated in Figs. 13*a*, 14*a* and 15*a*, respectively;

Figures13c, 14c and 15c are simplified vector diagrams showing force equilibria for a fragment of the outer ring under the loading of a single roller, comparing the effects of operation of the bearings of Figs. 13*a*, 14*a* and 15*a* at positive, zero and negative roller skew angles, respectively;

Figures 13d, 14d and 15d are schematic diagrams illustrating the relative magnitudes of the hoop stresses in the outer rings of the bearings illustrated in Figs. 13*a*, 14*a* and 15*a*, respectively, which furnish reaction to the forces F_{NR} in the diagrams 13*c*, 14*c* and 15*c*;

Figure16a, 17a and 18a are simplified free body diagrams similar to Figs. 13*b*, 14*b* and 15*b* but illustrating the force relationships developed in the bearings under pure radial loads;

Figures 16b, 17b and 18b are simplified vector diagrams comparing the effects of operation of the bearings of Figs. 16*a*, 17*a* and 18*a* at positive, zero and negative skew angles, respectively;

Figures 19a and 19b illustrate graphically a bimodal pressure distribution in the outer ring contact of the roller bearing configuration chosen to illustrate an application of the present invention;

Figures 20a and 20b illustrate graphically a peaked unimodal pressure distribution in the inner ring contact of the roller bearing configuration chosen to illustrate an application of the present invention; and

Figures 21 and 22 are views illustrating certain geometrical relations which are present in bearings embodying the present invention.

Referring now to the drawings, Fig. 7 illustrates a double row self aligning spherical roller bearing 10 embodying the present invention. The bearing 10 comprises an inner ring 11 having two raceways 12, an outer ring 13 having a raceway 14 confronting the raceways 12 and a series of rollers 15 rotatably mounted in annular space 16 between the inner and outer raceways. The rolling elements 15 are disposed in a pair of axially-spaced rows between the rings 11 and 13. The rolling elements 15 are usually separated and offset circumferentially in the annular space 16 by means of cages 17.

The bearing 10 has self-aligning capabilities. For this purpose, the outer ring is designed so that it can pivot about an axis transverse to the rotational axis of the inner ring 11. To this end, the outer raceway 14 is designed with a profile having a constant radius of curvature and the inner ring 11 is designed so that the raceways 12 are symmetrical with respect to a plane drawn perpendicular to the rotational axis of the bearing 10. The inner raceways 12 incline upwardly toward the centre plane of the bearing from opposite axial ends thereof. As is conventional is some double row spherical roller bearings, a guide ring 18 mounted on the inner ring 11 between the two rows of rollers 15 to separate the rollers 15 endwise from one another.

According to the present invention, the bearing 10 is designed to operate with its rollers 15 at positive skew angles.

As discussed in the aforementioned Kellstrom Patent, and as will become more fully apparent hereinafter, operation of the bearing with the rollers disposed at positive skew angles within a predetermined range minimizes overall friction within the bearing and increases the service life of the bearing.

In the Kellstrom Patent, the outer raceway is provided with a shallow relief adjacent the middle of the roller, and the inner raceway is provided with a pair of reliefs, one adjacent each end of the roller. These raceway modifications cause a bimodal pressure distribution (see Fig. 19*a*) to occur between the outer raceway and the roller and a peaked unimodal pressure distribution (see Fig. 20*a*) to occur between the inner raceway and the roller. These pressure distributions create friction force moments which pivot the roller into a positive skew angle. As noted above, in some applications such as double row self-aligning spherical roller bearings, the force relationships may change under different operating conditions thus altering the pressure distribution in some instances to an extent to negatively skew the rollers. Some forms of roller contouring in accordance with the present invention produce essenti~~ll~~y the same bimodal-unimodal pressure distribution discussed above. However, the relative pressure distribution remains essentially the same to maintain positive skew even under varying operating conditions.

Before describing specific embodiments of the present invention, it may be helpful to analyse more specifically negative and positive skew.

When a bearing operates with its rollers skewed at a positive angle, friction forces caused by the relative sliding motion between the surfaces of the raceways and the surface of the roller develop in the load in the load zone between the roller and the inner and outer raceways. As an aid to understanding this phenomenon, reference is made to Fig. 13*a* wherein the outer ring may be visualized as moving out of the plane of the paper towards the reader; the inner ring 11 may be visualized as moving away from the reader into the plane of the paper; and the roller 15 may be visualized as rotating about its axis 15*a* in the direction indicated by the arrow. Thus, the roller 15 may be regarded as rolling in a backward direction into the plane of the paper away from the reader at its contact with the outer raceway 14 if the outer ring 13 is stationary. Because of the skew of the roller 15, however, the direction of rolling motion of the roller 15 is not perpendicular to the plane of the paper. Rather, the rolling motion has a direction component which diverges from a plane M drawn through the inner and outer rings at the median of the roller 15. Since the outer ring 13 can only sustain motion about the bearing axis, its motion at its contact with the roller 15 is precisely perpendicular to the plane of the paper. As a result, the outer raceway 14 must slip over the surface of the roller 15 in a direction toward the plane M. This motion creates friction force F_T which is shown acting on the roller 15 and which can be resolved into a component F_{TA} directed axially of the bearing 10. By similar reasoning, a sliding friction force of the same magnitude but opposite in direction, $-F_T$, is developed on the bottom of the roller 15 at its contact with the inner raceway 12.

When an external axial load P_A is applied to the outer ring 13, there is an equal but opposite reaction load $-P_A$ on the inner ring 11. These loads cause a normal force F_N to be applied to the roller 15 by the outer raceway 14. As best seen in Fig. 13*a*, the normal force F_N has a radial component F_{NR} and an axial component F_{NA}. When the axial component F_{NA} of the normal force F_N is co-directional with the axial component F_{TA} of the friction force F_T acting on the roller 15, the roller is defined as being at a positive skew angle. Of course, a similar analysis holds for the forces at the inner raceway 12.

In the bearing illustrated in Fig. 15*a*, the roller 15 is disposed at a negative skew angle. As seen therein, the outer ring 13 slips relative to the roller 15 and causes the sliding friction force F_{TA} to be directed opposite the corresponding force F_{TA} in the bearing of Fig. 13*a* with its roller at a positive skew angle.

As an aid in understanding the advantages realized in a bearing operating with its rollers at a positive skew angle (as compared with zero and negative skew angles) reference is made to Figs. 13*b*, 14*b* and 15*b* which are free body diagrams of the various forces acting on the outer ring. For purposes of illustration, the various forces are referenced to the centreplane M of the roller even though, as will become apparent hereinafter, the forces do not strictly act at such location in the bearing of the present invention, due to the biomodal pressure distribution which exists at the outer raceway.

Considering first the bearing illustrated in Fig. 14*a* operating with its roller at a zero skew angle, it may be seen in Fig. 14*b* that the axial external load P_A gives rise to a normal force $-F_N$ which has an axial component $-F_{NA}$ acting to balance the force P_A. The normal force $-F_N$ also has a radial component $-F_{NR}$. This force balance is illustrated in Fig. 14*c*.

Considering now the bearing of the present invention illustrated in Fig. 13*a* operating with its roller at a positive skew angle, it may be seen in Fig. 13*b* that the axial external load P_A gives rise to the normal force $-F_N$ which acts on the outer ring. The normal force $-F_N$ has axial and radial components $-F_{NA}$ and $-F_{NR}$, respectively. In addition, the friction force $-F_T$, discussed earlier, acts on the outer ring in the direction opposite the external load component P_A.

The effect of the friction force $-F_T$ on the force balance in the bearing is best seen in Fig. 13*c* wherein the friction force $-F_T$ is vectorially subtracted from the external load vector P_A. The line of action of normal force vector $-F_N$ is drawn parallel to the normal force vector $-F_N$ (indicated in broken lines) and to the tip of the friction force $-F_T$. The resulting normal force $-F_N$ (indicated in full lines) is considerably smaller than the force which is applied to the outer ring by a roller at a zero skew angle as can be seen by comparing Figs. 13*c* and 14*c*. The lower normal force has the effect of increasing the fatigue life of the bearing. Outer ring hoop stresses are also reduced as shown in Figs. 13*d* and 14*d*.

The bearing which operates with its roller at a negative skew angle has higher hoop stresses and a shorter fatigue life than the bearing operating with its rollers skewed at positive or zero skew angles. This may be seen from a comparison of Figs. 13*c*, 14*c* and 15*c*. As illustrated in Fig. 15*b*, the friction force $-F_T$ applied to the outer ring by the roller acts in the same direction as the axial external load P_A. As a result, the friction force $-F_T$ is added vectorially to the axial load component P_A in Fig. 15*c*. The resulting normal force vector $-F_N$ (indicated in full lines) is therefore considerably greater than normal force vector $-F_N$ corresponding to zero skew (indicated in broken lines). Hence, the fatigue life of the bearing is reduced accordingly as shown in Figs. 14*d* and 15*d*.

The above analysis is applied to bearings under pure axial thrust load. When a bearing (constructed within conventional small bearing contact angles) operates with its rollers at positive skew angles and is subjected to pure radial, insignificantly greater normal forces are produced in the bearing, as compared with a similar bearing operating with its rollers at a zero skew angle. In a similar bearing operating with its rollers at negative skew angles, an insignificantly smaller normal contact force is developed as compared with a bearing operating with its rollers at zero skew. This may be seen by reference to Figs. 16*b*, 17*b* and 18*b*, which illustrate forces in a radially loaded bearing corresponding to the roller skew angles illustrated in Figs. 13*a*, 14*a* and 15*a*, respectively.

Unlike axially-loaded bearings, bearings under pure radial loads so not have significant hoop stress in their outer rings. Rather, outer rings of such bearings are internally stressed in the axial direction between each of the outer ring overlying each row of rollers. This relationship of external loads to internal forces and stresses for positive, zero and negative roller skew conditions is illustrated in the free body diagrams of Figs. 16*a*, 17*a* and 18*a*.

As best seen in Figs. 16*b*, 17*b* and 18*b* pure radial loads cause the axial stresses F_{HA} between the ring half sections to vary, depending on roller skew angle. For instance, as illustrated in Fig. 17*b* (zero skew angle) the axial stress is of a predetermined magnitude indicated by the vector F_{HA}. The axial stress F_{HA} is greater in Fig. 16*b* due to the friction force $-F_T$ caused by the positive skew angle of the roller. In contrast, the axial stress F_{HA} is smaller in Fig. 18*b* due to the friction force $-F_T$ caused by the negative skew angle of the roller. By comparing Figs. 16*b*, 17*b* and 18*b*, it may be seen that the magnitude of the normal force $-F_N$ under positive skew conditions is slightly greater than that under zero skew conditions, and the normal force $-F_N$ under negative skew conditions, is slightly less than that under zero skew conditions. Thus, for pure radial loads, bearings designed to operate with rollers at a positive skew angle do not realize the significant advantages realized when operating under pure axial loads.

When the external load on a bearing is a combination of radial and axial load components operation of the bearing with its rollers at positive angle reduces normal contact forces which arise predominately from the axial load component. However, because of the small contact angle (angle of inclination of the roller axis to the bearing axis) of most conventional double row spherical roller bearings, this advantage exists as long as the ratio of the axial load component to radial load component is greater than about 1:5.

Having now discussed the broad general concept of the present invention, attention will now be directed to a specific bearing incorporating preferred embodiments of contoured rollers. In accordance with the present invention, the external surface of each roller is formed as the rotation of an arc about an axis, which arc is composed of at least two different curves, and each of the inner and outer raceways is formed as the rotation of an arc of a circle about an axis.

Referring now to Fig. 19*c*, a specific embodiment of the present invention is illustrated which is a greatly enlarged and simplified view of the geometrical relations present in the bearing. The outer raceway 14 has a constant radius of curvature R_o, and the inner raceway 12, has a constant radius of curvature R_i. The radius of curvature R_i is greater than the radius of curvature R_o. Each roller 15 has a profile with a varying radius of curvature which is, in the present instance, increasing with increasing distance from a reference datum plane M which extends through the roller and perpendicular to the axis of the roller 15.

In the illustrated embodiment, the variable curvature of the profile of the roller 15 comprises radii of at least two different magnitudes R_{r1} and R_{r2}. One radius of curvature R_{r2} applies to the portions of the roller profile furthest from or outboard of the reference datum plane M and adjacent the ends of the roller. The radius of curvature R_{r2} merges with the radius R_{r1} at a distance X on each side of the roller median plane M. The radius of curvature R_{r1} is less than the radius of curvature R_i of the inner raceway but is greater than the radius of curvature R_o of the outer raceway. The radius of curvature R_{r2} is less than the radius of cuvature of the outer raceway R_o. In other words, the radius of curvature of the inner raceway profile R_i exceeds the radius of curvature of the roller everywhere along its length (including the zone at the roller median plane M) and the radius of curvature R_o of the outer raceway profile is less than the radius of curvature R_{ri} of the roller profile at the median plane M but greater than the radius of curvature R_{r2} of the roller at the ends of the roller. This may be expressed in the following geometrical relationships:

I. $R_i > R_{r1} > R_o$

II. $R_{r2} < R_o$

These conditions are satisfied by a roller having a profile provided by the arc of an ellipse having its minor axis at the roller median plane M.

The different roller profile radii have common tangents at their juncture with one another at distance X on each side of plane M. In other words, the roller profile is characterized by the absence of any edges (lines of zero radius of curvature) or blended edges (areas of small radii of curvature on the order of about R_{r1}) anywhere along the roller profile.

The roller illustrated in Fig. 19*c* is an example of a profile having a non-increasing radius of curvature, since the radius of curvature R_{r2} of the profile adjacent the ends of the roller 15 is smaller than the radius of curvature R_{r1} at the roller reference datum plane M. In this embodiment, the roller reference datum plane M is located at the lengthwise median of the roller 15, and the roller 15 is symmetrical with respect to the reference datum plane. Thus, the roller 15 may be defined as being symmetrically crowned.

There may be applications in which an asymmetrically profiled roller is preferable to the symmetrically profiled roller described above. An example of the roller and raceway profiles in a bearing having an asymmetrically-profiled roller is illustrated in Fig. 20*c*. The roller 115 has a profile with a variable curvature comprising three different radii of curvature: R_{r1}, R_{r2} and R_{r3}. The reference datum plane M is located closer to the right end of the roller 115 than to the left end. The radius R_{r1} merges with the radius R_{r2} at a distance X_1 to the left of the reference datum plane M, and the radius R_{r1} merges with the radius R_{r3} at a distance X_2 to the right of the reference datum plane M. The distance X_1 is farther from the median plane M than the distance X_2. As illustrated, the radius R_{r1} is greater than the radius R_{r2} which in turn is greater than radius R_{r3}. Of course, the aforementioned relationships of the profiles of the inner and outer raceways 112 and 114 also hold for the asymmetrically profiled roller bearing.

In accordance with the geometrical relationsships I and II set forth above, in both embodiments of the present invention the roller contacts the inner raceway at the roller median or reference datum plane M (X = 0) under load with a conformity ratio:

$$\frac{R_{r1}(X=0)}{R_1} < 1$$

which, for many spherical roller bearings, is about 0.98. It is known that as the load at the roller-inner raceway contact increases, pressure is distributed in a predetermined manner along the roller profile. For instance, between rollers and raceways of constant curvature, the Hertzian pressure distribution is an elliptical function of X which may be expressed by the following equation:

$$\text{III} \quad \left(\frac{P(X)}{P(0)}\right)^2 + \left(\frac{X}{le/2}\right)^2 = 1$$

wherein P(X) is the pressure at a distance X from the contact centre; P(0) is the pressure at the contact centre; X is the distance from the contact centre; and 1e/2 is the distance from the contact centre to the end of the roller. In the present embodiment of the invention, since the radius of curvature of the roller decreases with increasing X, the roller diameter is reduced at an increasing rate as X increases away from the roller median. Thus there exists a sharper drop in pressure as X increases than exists in the roller-raceway contact expressed in equation III. As a result, the pressure distribution at the inner raceway is more peaked than Hertzian as shown in Figs. 20*a* and 20*b*.

According to the geometrical relationships I and II, the roller does not contact the outer raceway at X = 0 under light load. This is because at

$$X = 0, \quad \frac{R_{r1}}{R_o} > 1;$$

however, since the reciprocal

$$\frac{R_o}{R_{r1}} < 1,$$

an inverse crown effect exists. At distances X^1 and X^2 however, contact does occur at light load. At these locations the following conformities exist:

$$\frac{R(X_1)}{R_o} < 1;$$

$$\frac{R(X_2)}{R_o} < 1.$$

The conformities may have predetermined values of about 0.98, as is customary for many spherical roller bearings. Thus, as the load increases, the contact areas at distances X_1 and X_2 from the reference or datum plane increase and merge together around X = 0 resulting in a pressure distribution which is defined as bimodal because of the pair of pressure levels occuring on opposite sides of the contact centre as shown in Figs. 19*a* and 19*b*.

In view of the foregoing, in the bearings of the present invention, the relation of the roller profile with the outer raceway remains the same even though the outer ring axis is not coaxial with the bearing axis. Thus, the bearing operate with their rollers at positive skew angles even when the inner and outer rings do not rotate about a common axis. Moreover, since the roller profiles have varying curvatures, the inner ring can be manufactured readily with a constant radius of curvature by conventional methods.

While preferred embodiments of the present invention have been described in detail with regard to spherical roller bearings havings symmetrically profiled rollers, various modifications, alterations and changes may be made without departing from the scope of the present invention as defined in the appended claims. For instance, the invention may be utilized in conjunction with other types of spherical roller bearings, wherein the inner ring raceway is spherically convex and linear recirculating bearings.

CLAIMS

1. A spherical roller bearing having inner and outer raceways and a plurality of rollers arranged to roll on both raceways, each raceway being formed as the rotation of an arc of a circle about an axis, the external surface of each roller being formed as the rotation of another arc about an axis, which said another arc is composed of at least two different curves, all such that in use of the bearing there is a resultant frictional moment on each roller causing it to be skewed to a skew angle which is positive as herein defined.

2. A bearing as claimed in claim 1, wherein each roller has a datum plane extending through it and perpendicular to its axis of rotational symmetry, the radius of curvature of the said arc of the inner raceway is greater than the radius of curvature of a curve of the said another arc of the external surface of the roller intersecting the datum plane, and the latter radius of curvature is greater than the radius of curvature of the said arc of the outer raceway, so that each roller, in use of the bearing, produces unimodal and bimodal pressure distribution with the inner and outer raceways respectively.

3. A bearing as claimed in claim 1 or 2, wherein each roller has a datum plane extending through it and perpendicular to its axis of rotational symmetry, and the said another arc of the external surface of roller includes a curve on each side of the datum plane having a radius of curvature smaller than the radius of curvature of a curve of the said another arc intersecting the datum plane.

4. A bearing as claimed in claim 1, 2 or 3, wherein each roller has a datum plane extending through it and perpendicular to its axis of rotational symmetry mid-way along the length of the roller.

5. A bearing as claimed in any preceding claim, wherein two adjacent curves of different radii of curvature of the said another arc of the external surface of each roller intersect such that a tangent can be drawn common to both curves.

6. A bearing as claimed in any preceding claim, wherein each roller has a datum plane extending through it and perpendicular to its axis of rotational symmetry, each roller being symmetrical about its datum plane.

7. A spherical roller bearing substantially as herein described with reference to and as shown in Fig. 21 or with reference to and as shown in Fig. 22 of the accompanying drawings.

8. In a bearing having inner and outer members with inner and outer raceways spaced apart to define a space therebetween, a complement of rolling elements rotatable about their axes in the space between the raceways, the improvement wherein at least one rolling element has a profile with a predetermined, varying curvature from a transverse reference datum extending through the axis of the rolling element, said inner and outer raceways having curvatures of constant radii of curvature, said rolling element profile producing pressure distribution under load resulting in a residual frictional moment producing forces counteracting the externally applied load, said rolling elements having a skew angle when the axis of a rolling element and the axis of the bearing are not coplanar and said skew angle being positive when the friction force components arising in each contact between said rolling element and raceways and acting on said rolling element in the axial direction, are so directed that they are codirectional to the axial component of the normal contact force acting on said rolling element in the same contact.

9. A bearing according to claim 8, wherein said inner raceway profile has a radius of curvature greater than said rolling element profile adjacent said datum and said outer raceway profile has a radius of curvature less than said rolling element profile adjacent said datum to provide a profile relation establishing unimodal and bimodal pressure distributions with said inner and outer raceways respectively.

10. A bearing according to claim 8, wherein said rolling element profile curvature includes radii of curvature on opposite sides of said datum which are smaller than the radius of curvature of said rolling element profile adjacent said datum.

11. A bearing according to claim 10, wherein said reference datum is located at the longitudinal median of the rolling element.

12. A bearing according to claim 10, wherein adjacent different radii of curvature of said rolling element profile curvature merge together at common tangents.

13. A bearing according to claim 10,

wherein said smaller radii of curvature of said rolling element profile curvature are of the same magnitude.

14. A bearing according to claim 10, wherein said smaller radii of curvature of said rolling element are of different magnitudes.

15. A bearing according to claim 8, wherein said rolling element profile curvature includes radii of curvature of opposite sides of said datum which are greater than the radius of curvature of said rolling element profile adjacent said datum.

16. In a bearing having inner and outer members with inner and outer raceways spaced apart to define a space therebetween, a plurality of rolling elements rotatable about their axes in the space between the raceways, the improvement wherein at least one rolling element has a profile with a predetermined, variable curvature from a transverse reference datum extending through the axis of the rolling element, said inner and outer raceways having curvatures of constant radii, said rolling element profile producing a pressure distribution under load resulting in a residual frictional moment producing forces counteracting the externally applied load.

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