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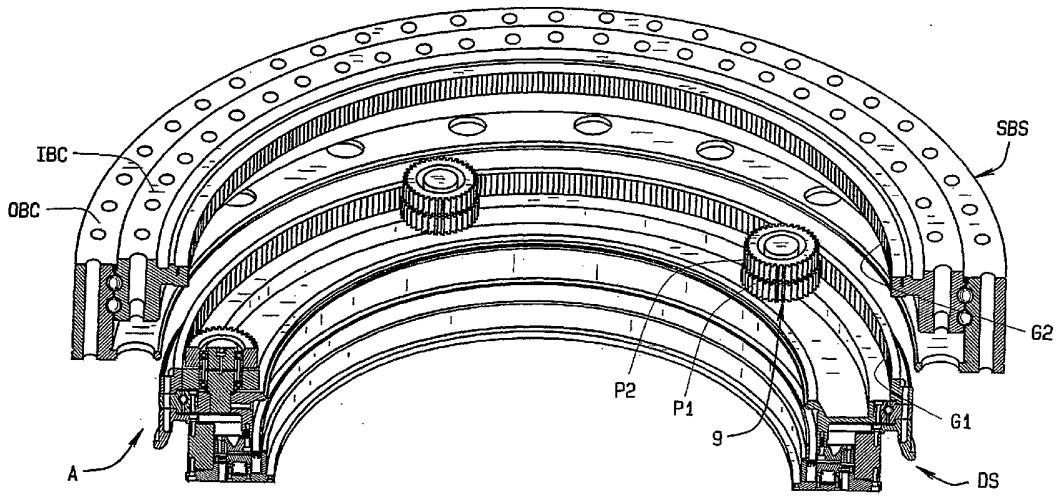
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(54) Title: BEARING ACTUATOR MODULE



(57) Abstract: A bearing actuator module comprises a support bearing assembly having a mounting ring (1) adapted to be mounted to an application support structure; a rotatable gear (G2) received within the mounting ring for rotation relative to said mounting ring. A drive subassembly (DS) which rotates said rotatable gear (G2) is removably received in the support bearing assembly. The drive subassembly comprises a stationary gear (G1), a plurality of pinion assemblies, each of which comprises a first pinion P1 which meshes with the gear G1 and a second pinion which meshes with the gear G2. The pinion assemblies are mounted in a carrier (10), which is rotated by a motor (M). A brake assembly (B) is provided to prevent rotation of the rotatable gear (G2) when the motor is not activated. Additionally, a feed back or position sensor is provided which indicates the relative rotational position of the carrier (and hence of the rotatable gear (G2)).

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BEARING ACTUATOR MODULE

CROSS-REFERENCE TO RELATED APPLICATIONS

This application claims priority to United States Provisional Application No. 60/569,084 filed May 7, 2004 entitled BEARING ACTUATOR MODULE and which is incorporated herein by reference.

TECHNICAL FIELD

This invention relates to a self-contained compact servo actuator device incorporating a torque motor and electronic position feedback device with an integrated high capacity support bearing/planetary gear reducer.

BACKGROUND ART

Prior designs of such devices include bearings with an integral ring gear and non-integral servomotor/ pinion concept. Such designs are not compact or integrated. They are custom designed by OEM and field adjustment is required at installation. They also have high backlash which requires a motor mounting surface and a pinion mesh adjustment device. Additionally, in such designs, gear lubricant is exposed to rotor interior mechanisms.

Another design includes an integral ring gear and servo hydraulic pinion. This design requires a hydraulic power supply. It is neither compact nor integrated. Rather, it is custom design required by OEM, and hence field adjustment is required at installation. It also suffers from high backlash and requires a motor mounting surface and a pinion mesh adjustment device. Additionally, in such designs, gear lubricant is exposed to rotor interior mechanisms.

A third design includes direct drive torque motors. This design requires a massive motor and motor drives and motor currents/heat to produce output torques comparable to other methods.

Yet another design includes a linear servo electric. This design produces good output torque with reduced backlash. No ring gears are required. However, this design is bulky and usable as a limited angle actuator only. Further, the output angle is non-linear to input stroke position.

Lastly, the linear hydraulic method has good torque and backlash but requires a hydraulic power unit and its output angle is non-linear to input stroke position. No ring gears are required.

SUMMARY OF THE INVENTION

Briefly stated, a bearing actuator module made in accordance with the present invention eliminates the necessity for an OEM manufacturer to design a custom actuator/motor/support bearing mechanism out of many discrete parts. An actuator is defined here as a device that is used to rotate a load, usually for the purpose of repositioning it, and usually in less than 360° of rotation. A motor is defined here as a device that rotates a load through multiple rotations, usually much greater than 360° and usually at a constant speed or slightly varying speed.

The module of the present invention can be used either as an actuator or as a motor. It has a compact, modular design and operates at a high torque/ low speed capability. The bearing actuator module includes an integral planetary in-line gear reducer to provide speed reduction and torque amplification. This reduces motor size and current requirements. The reducer design is flexible and permits either low (such as 10:1) or high (such as 300:1) reduction ratios. Higher and lower ratios are possible as well. The exact ratios available to the application depend on the application requirements and geometry restrictions.

The bearing actuator module has a large open bore design, and thus, a sun gear is not required. The hollow center of the assembly permits passing through of control cables for the convenience of external components, e.g. power and control cables or plumbing to and from external devices, or even from the device itself. For larger designs (i.e.,

greater than 1 meter in OD), the bore hole diameter is generous. For example, the module can have an ID/OD ratio of greater than about 0.5 (very hollow) and a width to OD ratio of about 0.2 or less.

According to one aspect of the invention, the bearing actuator module includes a support bearing and a separate, removable drive module. The drive module is sealed relative to the support bearing to prevent grease and other contaminants from the support bearing from entering the drive module. Additionally, because the drive module is removable, the drive module can be swapped out and replaced without the need to de-mount the support bearing.

According to a further aspect of the invention, the module is hermetically sealed. The module includes a top plate and an inner plate. The inner plate encloses the inner diameter of the support bearing and drive module. In combination, the inner and top plates and will prevent grease and other lubricants from exiting the module. They will also prevent contaminants from entering the module.

In a further aspect of the invention, the module is provided with an air channel that surrounds the stator of the motor. This air channel allows for the heat produced by the motor to be dissipated to atmosphere, rather than being retained in the module.

The bearing actuator module is provided with an integral feedback device that permits operation of the assembly as a closed loop servo-mechanism.

The design of the gear geometry permits very low backlash or zero backlash, thereby reducing wear.

The module can include an integral multi-pole hollow shaft torque motor to provide the input torque needed to drive the planetary reducer and output.

The module can include an integral brake. The brake is a holding brake to prevent rotation of the load accept for when the motor has been activated. The brake also prevents rotation during power loss. The brake is designed such that whenever the motor is powered up, the brake is

magnetically retracted out of engagement by a DC coil to allow for rotation of a load.

The actuator bearing module can be used, for example, for very large pitch and yaw bearing actuator positions such as on large wind turbine rotor blade root bearing positions and wind turbine drive (nacelle) housing yaw support bearing position, although other applications such as for jib cranes, or machining axis are possible as well. The actuator bearing module can also have many other uses.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a perspective cross-sectional view of an illustrative embodiment of a bearing actuator module of the present invention;

FIG. 2 is an elevational cross-sectional view of the bearing actuator module;

FIG. 3 is an exploded view of the module;

FIG. 4 is an enlarged cross-sectional view of the module;

FIG. 5 is a further enlarged cross-sectional view of the module taken along circle 5 of FIG. 4 and showing the drive motor and brake components of the module;

FIGS. 6 and 7 are exploded, cross-sectional views of the module taken from opposing angles and shown in differing scales;

FIG. 8 is an enlarged plan view of the feed back device and brake components for the module;

FIG. 9 is a free body diagram of the module; and

FIG. 10 is a cross-sectional view similar to FIG. 4, but of an alternative embodiment of the module.

Corresponding reference numerals will be used throughout the several figures of the drawings.

BEST MODES FOR CARRYING OUT THE INVENTION

The following detailed description illustrates the invention by way of example and not by way of limitation. This description will clearly enable

one skilled in the art to make and use the invention, and describes several embodiments, adaptations, variations, alternatives and uses of the invention, including what we presently believe is the best mode of carrying out the invention. Additionally, it is to be understood that the invention is not limited in its application to the details of construction and the arrangements of components set forth in the following description or as illustrated in the drawings. The invention is capable of other embodiments and of being practiced or being carried out in various ways. Also, it is to be understood that the phraseology and terminology used herein is for the purpose of description and should not be regarded as limiting.

An illustrative example of a bearing actuator module A of the present invention is shown in FIGS. 1-9 in varying levels of detail. The module A is comprised of two main parts to satisfy the requirements of a servo gear reducer motor/ support bearing/ safety brake. The module A is comprised of a support bearing subassembly SBS and a drive subassembly DS. (FIGS. 3, 6 and 7). The support bearing subassembly is mounted to a support structure and supports a load (such as turbine rotor blades). The drive subassembly DS is removable from the support bearing subassembly SBS, allowing the drive subassembly to be removed from the actuator for servicing (or replacement) without the need to remove the complete actuator from support structure to which it is mounted.

The support bearing subassembly SBS has an outer mounting bolt circle OBC used to mount the Module A to the application support structure. The outer bolt circle OBC includes a plurality of bolt holes through which bolts or other fasteners can extend to mount the outer bolt circle OBC of the SBS to the support structure. The inner surface of the outer bolt circle defines an outer race 1 of the support bearing structure SBS. An inner mounting bolt circle IBC is received within the outer bolt circle. The outer surface of the inner mounting bolt circle defines an inner race 3 of the support bearing subassembly SBS. A plurality of spherical roller elements or balls BB between the inner and outer races allow the

inner and outer races to rotate relative to each other. The application loads are supported directly through this bearing subassembly and into the application support structure. Hence, the support bearing subassembly SBS must be sized to properly carry these loads.

The support bearing subassembly SBS (as shown in FIGS. 4 and 7), as noted above, contains an outer support bearing race 1 and a support bearing inner race 3. A support plate 2 is mounted to the bottom surface of the outer bolt circle OBC, and supports the drive subassembly DS in the actuator module A as will become apparent below. The inner bolt circle has a toothed radial inner surface defining an internal ring gear G2. Lastly, the support bearing assembly includes a top cover 4 which covers the components of the inner bolt circle and the drive subassembly.

The drive subassembly DS is sized to support and carry only the application torque loads. The drive subassembly DS is removable from the support bearing subassembly SBS for servicing and replacement, as noted above, without disturbing the support bearing subassembly. (This assumes of course that the application support structure permits enough room to remove the subassembly.) The drive subassembly has a hollow shaft torque motor M (FIGS. 4 and 5), a safety brake mechanism B, a positional sensor or feedback device FD, a planet carrier 10, planet gear subassemblies 9 (FIGS. 6 and 7), a stationary internal gear 8 (G1), a planet carrier support bearing 9d and an inside cover 25.

Rotary shaft seals 5 located at the outer diameter of the module A (between the inner and outer races 1 and 3) and at the inner diameter of the top plate 4. The cover plate 4 and associated bores are used to protect the support bearing and gears from lubricant loss or ingress of contaminants. The seals 5 need only support slow rpm movement. A third high speed interior shaft seal 11 is positioned axially below the planet gear sub-assembly 9 to help separate the gearbox from the motor/feedback section to keep the motor and feedback device clean and functioning properly. A sealed carrier support bearing 7 is used to support unbalanced planet gear tooth loads as well as non-rotating thrust loads

from the engaged safety brake springs 16 during power down. Electrical connections to the module A are made for the motor M, a feedback device FD and a DC safety brake coil 21.

Figure 4 illustrates a large diameter Module A designed for use with a wind turbine blade pitch actuator bearing application. Numerous other arrangements could be used to mount the bearings, support the motor, etc. For example, an alternative arrangement for mounting the drive support DS in the support bearing subassembly SBS is shown in Figure 10. In FIG. 10, the Module A' uses tapered crossed rollers inside the rotating output ring.

The items comprising an illustrative embodiment of Module A will now be described in more detail. The support bearing subassembly (SBS) includes the outer bearing race 1, support plate 2, and inner race 3. The support plate 2 has holes 2a to access mounting stud nuts in the inner bolt circle IBC which fasten the application load (i.e., turbine blades) to the inner race 3. The holes are tapped to receive threaded hole closures which seal the support plate from the environment. Alternatively, the holes may be straight bores which will then receive push-fit non-threaded hole closures. The support plate 2 mounts to the outer race 1 and supports the drive sub-assembly DS. In the embodiment of FIGS. 1-8, the loads applied to the inner race 3 are carried directly by the outer bearing race 1, and through the support plate 2. The support plate 2 acts substantially as a compression spacer only. Hence, the support plate 2 does not support the application bearing's axial, radial and over turning moment loads. The module A', of FIG. 10 requires that the support plate 2' carry the application bearing loads and is therefore shown to be of a greater cross sectional thickness.

The inner bearing race 3 has an internal bearing gear G2 with number of teeth Ntg2. The gear G2 is shown to be integral with the race 3, but could be formed separately if desired. Spherical roller elements or balls BB are positioned between the inner race 1 and the outer race 3 to allow the outer race to rotate relative to the inner race. As noted above, an

outer seal 5 is positioned near the top surface of the actuator between the inner and outer races 1 and 3.

The top cover plate 4 sits on a gasket 4b and is fastened through the gasket to the upper surface of the inner race 3, seating against a pilot diameter 4c on the inner race 3 to maintain radial alignment of the seal 5. The cover plate 4 is placed over the upper surface of the inner race 3 and extends radially inwardly from the inner race 3. The cover plate 4 includes an axially inwardly directed flange 4a spaced slightly radially outwardly from the radial inner edge of the cover plate. Upper inner seals 5 are positioned adjacent the radial inner surface of the cover plate flange 4a. The relative rate of motion between the inner and outer races is small, less than 5 rpm, hence, the inner and outer upper seals 5 need only be sufficient to withstand a small rpm.

Turning to FIGS. 4 and 5, the Drive Subassembly DS includes a Motor Mounting Plate 6. A sealed Carrier Support Bearing 7 is fixed to the motor mounting plate 6. The bearing 7 separates the motor/feed back portion of the drive sub-assembly DS from the gearbox of the actuator module A and prevents contaminants from the gearbox from entering the motor/feed-back portion of the drive sub-assembly. Additionally, the bearing 7 prevents a loss of lubricant from the gearbox of the actuator module. A Stationary Internal Gear G1 is fixed to the motor mounting plate 6 via the support bearing 7. As seen in FIG. 4, the outer race of the bearing 7 includes a flange. Aligned fastener receiving bores in the motor mounting plate 6, the flange of the support bearing 7 and the Gear G1 enable the connection of the three components together. Gear G1 has Ntg1 teeth, which is greater than the number of teeth on the integral bearing gear G2 .

The drive subassembly DS is provided with a planet subassembly 9 (FIGS. 4, 5 and 7). The planet subassembly 9 includes a planet pinion gear 9a comprised of a lower pinion P1 and an upper pinion P2. The upper and lower pinions (P2 and P1) are interconnected to move together. They can be formed as an integral unit, or can be individual pinions which

are connected together to rotate together. The lower pinion P1 has more teeth than upper pinion P2. The pinions P1 and P2 have Ntp1 teeth and Ntp2 teeth, respectively, with Ntp1 being greater than Ntp2. Pinion P1 meshes with gear G1 and pinion P2 meshes with gear G2. Both gear sets (i.e., gear set G1/P1 and gear set G2/P2) have the same center distance. A planet gear bearing spacer 9b with radial retaining pin holes is positioned internally of the plant pinion gear 9a. The planet gear bearing spacer 9b is fastened into the planet pinion gear 9a with the planet gear bearing spacer retaining pin 9c. The upper and lower planet gear support bearings 9d are provided with one seal for internal relubrication. A planet shaft 9e is journaled through the bearings 9d and is also provided with lubrication holes. The planet shaft bearing spacer 9f with lubrication holes is provided to maintain the relative position of the pin 9c about the shaft 9e. The spacers 9b and 9f are dimensioned such that an air gap is formed between the two spacers. A planet shaft standoff height spacer 9g surrounds the shaft 9e and is positioned below the lower support bearing 9d. The spacer 9g can be integral with, or separate from, the shaft 9e. The spacer 9g could also be formed integrally with the lower planet gear bearing 9d. The spacer 9b could also be formed integrally with the planet pinion gear 9a, in which case the retaining pin 9c would not be required.

The planet subassembly 9 is supported by a planet carrier 10. The planet carrier 10 is received within the carrier support bearing 7 and is secured to an inner race of the bearing 7. The planet carrier 10 includes a plurality of threaded openings 10b which receive a threaded end of the planet shaft 9e. A retaining or set screw is received in a planet carrier planet shaft retaining screw hole 10a to maintain the planet carrier shaft 9e in the threaded opening 10b.

A shoulder 10c is formed at the radial inner surface of the planet carrier 10. The planet carrier seal 11 is positioned about the shoulder, and, as seen, is axially below the inner seal 5. The planet carrier seal 11 will be subject to higher rates of rotation (i.e., greater than 30 rpm) and thus must be able to withstand the greater rotation.

The planet subassembly 9, the planet carrier 10, the ring gear G1 and the planet carrier seal 11 make up a portion of the gear box section of the module A. The rest of the gear box is comprised of the inner bolt circle IBC containing the ring gear G2. The seals 5 and 11,,the integral seals inside the carrier support bearing 7, and the plugs for the holes 2a in the support plate 2 substantially prevent lubricants from escaping the gear box section, to thereby substantially prevent the lubricants from splattering or otherwise entering the motor/brake/feed back portion of the drive subassembly DS.

The motor/brake/feed back portion can include a rotating rotor brake plate 12 with teeth 12a (FIG. 5). The teeth 12a have angled tooth engagement faces for over running torque protection. The tooth engagement faces forms an acute angle F with the vertical. The acute angle F can be, for example, between 5° and 30°..

The motor M comprises a torque motor rotor 13 and a torque motor stator 14 with attached motor leads 14a. A gap or slot S (FIG. 4) exists between the motor M and the motor mounting plate 6. This slot S, which surrounds the motor M, forms an air gap to allow for cooling of the motor. As can be appreciated, the heat from the stator 14 is dissipated to the atmosphere, reducing heat buildup within the actuator module A. Without the capability of dissipating the motor heat, the heat that would build up in the actuator could affect the lubricants and the electronic components. Heat build up also requires a de-rating of the stator magnetization current limit and results in a de-rating of the motor torque output to prevent the degradation of the stator windings insulation.

A bottom plate 15 is positioned below the motor M and is secured to the motor stator 14. The bottom plate 15 has clearance for all electric leads (i.e., leads 14a, 21a and 23). Aligned screw holes 14b and 15a (FIG. 5), in the stator 14 and plate 15, respectively, allow for attachment of the plate 15 to the stator 14.

Spring plungers 16 are provided to actuate a safety brake 18 for over running torque protection. The brake plate 18 is a non-rotating plate

provided with teeth 18a. The teeth 18a provide angled tooth engagement faces. The spring plunger or elements 16 press against a flange in the plate 15 to urge the brake plate teeth 18a into engagement with the rotor brake plate teeth 12a. The spring plungers 16 can be a compressible material or a coil spring, or any other desired spring element.

The spring plungers 16 can, for example, comprise a plurality of compressible members. The plungers 16 are spaced about the bottom plate 15 and are received in openings 15b in the bottom plate 15.

The brake 18 includes a radial flange 18b provided with a plurality of radial bores 18c. Cam followers 17 are received in the bores 18c for anti-friction axial engagement and anti-rotation of the brake plate 18. The cam followers 17 include a head 17a and a shaft 17b. The cam follower shaft 17b is received in the bore 18c; and the cam follower head 17a rides in a groove or pocket 15c formed on an axial rib 15d of the bottom plate 15. The cam followers 17 and pockets 15c allow for angular misalignment between the between the brake plate 18 and the bottom plate 15 and for vertical movement of the brake plate 18 relative to the bottom plate 15, while preventing rotational movement of the brake plate 18 relative to the bottom plate 15. The cam followers can also transmit torque from the bottom plate 15 to the brake plate 18.

The brake 18 is activated by a brake actuator. The brake actuator is comprised of an outer pole plate 19, an inner pole plate 20, and a brake coil 21 with attached leads 21a. When the brake coil 21 is activated, the pole plates 19 and 20 will be magnetized and attract the brake plate 18, to pull the brake plate 18 out of engagement with the rotor brake plate 12.

As noted above, the module includes a positional sensor or feedback device FD. The feedback device includes a feedback mounting spacer 22 which spaces a feedback pickup 23 above the bottom plate 15. The pickup 23 is mounted by a bracket 23a and includes leads 23b. A feedback scale 24 surrounds the pickup 23 to be read by the pickup 23. The pickup 23 reads the scale 24 to provide an output indicative of the angular position of the motor rotor 13.

An inside cover 25 covers the inner surface of the module A. The inner seals 5 and 11 seal against the cover 25. The seal 11 and the sealed bearing 7 thus together isolate the gearbox from the motor/brake/feedback portion of the actuator. The covers 25 and 4 mate together, as seen in FIG. 4, to, in effect, seal the module, thereby protecting the gear box portion and the motor/brake/feed back portion of the module A against the entry of contaminants. As seen in FIG. 4, the seal 5 is located at the connection between the top and inside covers 4 and 25. The top cover 4 includes a flange 4a which extends parallel to the inside cover 25 to define a channel which receives the seal 5. The construction of the covers 4 and 25 along with the interconnection of the covers 4 and 25 protect the seal 5 from compressive forces, for example, from someone standing on the cover 4 while servicing the module A. The cover 4 is supported by the cover 25 and preferably is designed to support considerable weight. Hence, a person working on the actuator can stand on the cover 4 without affecting the sealing of the actuator by the covers 4 and 25.

Principle of Operation

As described in Machinery's Handbook, 22nd Edition, pg. 843, Fig. 13, for a planetary gear set with a dual internal ring gear arrangement, a driver carrier, a fixed ring gear, and a follower ring gear. With the larger ring gear fixed instead, we get;

$$\text{SUR} = \text{F} = 1-\left(\frac{y*B}{C*x}\right)=1-\left(\frac{Ntg1*Ntp2}{Ntp1*Ntg2}\right)$$

This is because the number of gear teeth, Nt, is proportional to the pitch diameter, P.D., where SUR is the SPEED UP RATIO. As used below, P.D.g1 is the pitch diameter of gear G1; P.D.g2 is the pitch diameter of gear G1; P.D.p1 is the pitch diameter of pinion P1; and P.D.p2 is the pitch diameter of pinion P2.

The GEAR REDUCTION RATIO (GRR) is the reciprocal of the SPEED UP RATIO and is defined as follows:

$$GRR = 1/F = \frac{1}{1-\left(\frac{Ntg1 * Ntp2}{Ntp1 * Ntg2}\right)}$$

This is derived independently, below. As noted above, Ntg1, Ntg2, Ntp1, and Ntp2 are the number of teeth on gear G1, gear G2, pinion P1 and pinion P2, respectively.

From the Free Body Diagram of FIG., 9, we observe that the stationary internal gear G1 acts to retard the planet rotation. The tooth count differences between pinions P1 and P2 cause the output ring gear to advance at a differential rate. The vector equations are;

1. Summation of Tangential Forces *F* on Pinion 9 = 0, or *Ft1+Ft2 = F*

2. Summation Moments on Pinion 9 = 0, or $\frac{Ft1 * P.D.p1}{2} = -\frac{Ft2 * P.D.p2}{2}$

3. Summation Moments on Carrier 10 = 0 according to the equation *Tin/planet – F* * Center distance = 0, where T*in/planet* is the motor input torque per planet gear 9 and the center distance is equal to the distance from the center of rotation of the module to the center of the planet shaft 9e.

4. Center distance = $\frac{P.D.g1 - P.D.p1}{2} = \frac{P.D.g2 - P.D.p2}{2}$

Substituting equation (4) into equation (3) gives,

5. *F = Tin/planet* * [2 / (P.D.g1 – P.D.p1)]

Substituting equations (1) and (2) into equation (5) gives;

6. $Ft2 = \frac{F * P.D.p1}{P.D.p1 - P.D.p2} = F * \left(\frac{1}{1-\frac{P.D.p2}{P.D.p1}}\right)$, and

7. *Ft1 = F – Ft2*

Once again, because the number of gear teeth, Nt, is proportional to the pitch diameter, P.D., equations (5) and (6) can be rewritten as follows:

8. $$F = Tin/planet * \left(\frac{2}{1 - \frac{Ntp1}{Ntg1}} \right)$$

9. $$Ft2 = F * \left(\frac{1}{1 - \frac{Ntp2}{Ntp1}} \right)$$

Thus looking at equation (9), if *F* is arbitrarily defined to be in a positive direction, then whenever Ntp2 is less than the Ntp1, *Ft2* will be greater than *F* and also will be in a positive direction. Thus from equation (7), *Ft1* must be in a negative direction or the "pinion 1/gear 1" tooth interaction will cause a retarding force onto the planet carrier.

The net force F on the pinion bearings is:

10. $F = Ft2 + Ft1$, but *F* is positive and *Ft2* is positive for Ntp2 <Ntp1, then *Ft1* is negative. This can be written in scalar terms as:

11. $F = Ft2 + (-Ft1) = Ft2 - Ft1$

Thus, *F* is a small value as it is the difference between the tooth reaction forces *Ft1* and *Ft2*. *F* represents the radial force component on the two Planet Shaft Bearings 9d due to the tangential tooth reaction forces. The main point is that this effect helps keep the planet bearing loads much lower than if the forces were not somewhat self-canceling. The radial gear tooth force is calculated in the standard way and is added to give a total force for the planet bearing. (Not shown.)

Because the torque motor rotor 13 is rigidly fastened to the planet carrier 10, the motor rotor RPM input speed is equal to the planet carrier speed, RPMcarrier. The various rotational speeds of the device are derived as follows:

12. RPMplanetbearing = (RPMcarrier – RPMg1)*(Ntg1/ Ntp1), but RPMg1=0. Thus, RPMplatentbearing = RPMcarrier * (Ntg1/Ntp1), where RPMplanetbearing is the rate of rotation of the planet bearing or the planet pinions P1 and P2 about the planet shaft 9e.

13. RPM planet to input = RPMcarrier – RPMplanetbearing. Substituting equation (12), the planet RPM relative to the carrier = RPMcarrier*(Ntg1/Ntp1). Thus the speed ratio of the planet relative to the carrier is,

14. SPEED RATIO planet to input = RPMplanet to input/RPMcarrier or,

= 1 – (Ntg1/Ntp1).

Now the output gear speed relative to the input becomes,

15. RPMg2 to input = RPMcarrier – RPMplanetbearing * (Ntp2/Ntg2)

or, = RPMcarrier–RPMcarrier * (Ntg1/Ntp1)*(Ntp2/Ntg2),

or, $= \text{RPMcarrier} * \left(1 - \frac{Ntg1 * Ntp2}{Ntp1 * Ntg2}\right)$,

and the overall speed up ratio SUR is,

16. SUR = RPMg2 to input / RPMcarrier

or, $= 1 - \frac{Ntg1 * Ntp2}{Ntp1 * Ntg2}$

Thus the Gear Reduction Ratio is,

17. GRR = 1/ SUR or, $= \frac{1}{1 - \frac{Ntg1 * Ntp2}{Ntp1 * Ntg2}}$

This agrees with the Machinery's Handbook equation from above.

Operation

When the power to the Module A (or A') has been interrupted, the safety brake plate 18, with it's tapered teeth 18a, is forced into engagement with the mating tapered teeth 12a of the rotor brake plate 12 by the spring plungers 16. The force required is given by;

Fbrake = [(F.S. * Tout)/(G.R.R * MeanBrakeToothRadius)] * tan(F), and

Numberspringplungers = Fbrake / Fspringmin

Where:

Fbrake = total required spring starting force in kN

F.S.= factor of safety desired in the holding torque

Tout = desired holding torque in kNm

GRR = Gear Reduction Ratio from above

Mean Brake Tooth Radius = radius in meters from the centerline axis of the Module A drive to the center of the brake tooth contact (i.e., the center of the plate teeth 12a and 18a).

F =acute angle measured from main centerline axis to tooth flank face, (equates to the pressure angle if an involute gear tooth form is used), in degrees.

Fspringmin = starting force of spring plunger in kN.

(Fspringmax = ending force of spring plunger in kN when the plunger is fully depressed)

Should the wind load to the rotor blades produce a holding torque which exceeds (F.S. * Tout), the brake plate 18 and rotor brake plate 12 will ratchet harmlessly tooth by tooth until the excessive torque overload is removed, at which point, the teeth 12a and 18a of the brake plates 12 and 18, respectively, will engage. The F.S. and Tout limits should be chosen to reduce the tendency for slip to occur except under exceptional circumstances. However, the brake coil 21, the outer pole plate 19 and the inner pole plate 20 must be designed to provide sufficient magnetic attraction to overcome the Fbrake spring force. The smaller the coil, the larger the current required, subject to ampacity limitations of the winding.

Once the torque motor M and brake coil 21 are energized, the brake plate 18 retracts from engagement and the torque motor rotor 13 and the planet carrier 10 are free to rotate. That is, the motor rotor 13 is connected to the carrier 10 via the rotor brake plate 12. Thus, when the brake coil 21 is not energized, the spring plungers 16 will force the brake plate 18 into engagement with the rotor brake plate 12 to prevent rotation of the carrier plate, as described above. However, when the coil 21 is energized, the magnetic force of the coil overcomes the spring force of the spring plungers 16 to pull the brake plate 18 out of engagement with the

rotor brake plate 12, thereby allowing for relative rotation of the carrier 10. Because the attraction force of the brake plate 18 to the inner and outer pole plates 19 and 20 increases faster (inverse square law rate) than the spring rate force (linear rate) of the spring plungers 16, the motor control will be designed to reduce the coil current to a small fraction of the initial current to limit the steady state heat output of the brake coil 21 and required energy consumption.

During either a planned shut down or during an emergency shut down, the control will use either battery backup or super-capacitor backup to remain active for sufficient time to send a small reverse current through the brake coil 21. This will overcome any residual flux due to the materials coercivity (tendency for a material to remain magnetized once magnetization current is removed) and will allow the spring plungers 16, which have been completely compressed to their highest spring force, or Fspringmax, to re-engage the brake plate 18 with the rotor brake plate 12. The cam followers 17 carry the torque loads to resist any rotation of the brake plate 18 as it engages the rotor brake plate 12 and also manage the friction in the axial guide slots 15c to prevent bind up of the brake plate 18.

The servo control receives a control set-point from a control computer based upon real time wind loads and the pitch control algorithm, and commands a positional move based on the positional feedback received from the module feedback device FD. When a move is commanded, the brake coil 21 is activated to release the planet carrier 10, as noted above. Additionally, the motor M is activated and the planet carrier 10 begins to rotate. For this example, it will be assumed the carrier 10 is rotated in a counter-clockwise direction. The lower pinion P1 of the planet gear 9 engages the stationary internal gear G1 to force a certain angular rate of clockwise rotation onto the planet pinion gear P2 relative to the input rpm of the motor rotor, called the SPEED RATIO. However due to differences in tooth count, the inner bearing race with integral internal bearing gear G2 is driven by the upper pinion P2, of the planet gear 9 at a clockwise rate which does not quite make up for the slightly greater

counter-clockwise rotation rate of the planet carrier 10, thus imparting to the inner bearing race 3 a differential counter-clockwise speed based upon the tooth count differences in the gear reduction ratio formula (Equation 6) above. Thus two high speed pinion/gear meshes of slightly unequal and opposite direction result in a slow output ring gear rotation.

In an emergency stop or a planned shutdown, the motor is activated to turn the turbine blades approximately 90° to their normal operating position. The brake assembly B will then hold the turbine blades in this position. To accomplish this, energy is stored in batteries or super capacitors, so that the control system has sufficient power for its cpu, for the motor to operate, and for the feedback devices to generate signals indicative of the blade's rotational position. As discussed above, should the control system release the brake when the blade is still moving, and still has inertia, the angled tooth faces allow for the teeth to cam out of engagement until the spring force is greater than the inertia force.

FIG. 10 shows an alternative embodiment of the module in which the forces are directed radially as well as axially. In this embodiment, the base plate 2' will bear substantially more load than the base plate 2 of FIGS. 1-8. Hence, the base plate 2' is more massive than base plate 2. Other than the use of different bearings, as seen in FIG. 10, operation of the module A' works identically to the operation of the module A. In FIG. 9, the pinion assembly 9' is shown to include tapered roller bearings 9d', rather than ball bearings.

As various changes could be made in the above constructions without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense. For example, although the gear G1 is the stationary gear and the gear G2 is the rotatable gear, the module could be designed such that the inner gear is rotatable and the outer gear is stationary. The brake assembly B actuator is positioned beneath the stator brake plate 18, and hence pulls the stator brake plate away from the rotor brake plate against the force of

the spring element 16 when energized. However, the brake assembly actuator could be positioned to act on the rotor brake plate 12, rather than the stator brake plate 18. The position of the pickup device 23 and scale 24 of the feedback device FD could be switched such that the pickup device would be operatively connected to the carrier 10, such that the pickup 23 would rotate with the carrier rather than the scale 24. These examples are merely illustrative.

Claims

1. A bearing actuator module comprising:

a mounting ring (1) adapted to be mounted to an application support structure;

a rotatable gear (G2) received within said mounting ring for rotation relative to said mounting ring; and

a drive subassembly (DS) comprising:

a stationary gear (G1);

a carrier (10) mounted in said module to be rotatable relative to said inner gear (G1);

a plurality of pinion assemblies (9) mounted on said carrier (10); each pinion subassembly comprising a first pinion (P1) positioned to mesh with said stationary gear (G1) and a second pinion (P2) positioned to mesh with said rotatable gear (G2); said first and second pinions being connected to rotate together;

a motor (M) operatively supported by said stationary gear (G1); said motor comprising a rotor (13) and a stator (14); said stator being operatively connected to said stationary gear (G1) and said rotor (13) being operatively connected to said carrier (10);

whereby, activation of said motor causes rotor (13) and hence said carrier (10) to rotate in a first direction; the rotation of said carrier (10) moving said pinion assemblies relative to said stationary gear (G1); the meshing of said stationary gear (G1) with said first pinion (P1) causing said pinions (P1 and P2) to rotate about an axis, and the rotation of inner gear (G1) rotates the first and second pinions (P1 and P2), and rotation of said pinion (P2) induces rotation of said rotatable gear (G2).

2. The bearing actuator module of claim 1 wherein said first pinion (P1) has Ntp1 teeth and said second pinion (P2) has Ntp2 teeth, and the stationary gear (G1) has Ntg1 teeth and the rotatable gear (G2)

has Ntg2 teeth; wherein in Ntp1 and Ntp2 are not equal and Ntg1 and Ntg2 are not equal.

3. The bearing actuator module of claim 2 wherein Ntp1 is greater than Ntp2, and wherein Ntg1 is greater than Ntg2.

4. The bearing actuator module of claim 1 including a motor mounting plate (6); said motor (M) and said stationary gear (G1) being mounted to said motor mounting plate (6).

5. The bearing actuator module of claim 4 where said motor (M) is mounted to a first side of said motor mounting plate (6) and said stationary gear (G1) is mounted to an opposite side of said motor mounting plate (6).

6. The bearing actuator module of claim 4 comprising a carrier support bearing (7) mounted to said motor support plate (6) by a first race of said bearing (7); said carrier (10) being connected to a second race of said bearing (7).

7. The bearing actuator module of claim 1 wherein each said pinion assembly includes a pinion shaft (9e) which is secured to said carrier (10); said first and second pinions (P1 and P2) being rotatably mounted about said pinion shaft (9e).

8. The bearing actuator module of claim 7 wherein each said pinion assembly include a pinion bearing assembly positioned between an inner surface of said first and second pinions (P1 and P2) and an outer surface of said pinion shaft (9e).

9. The bearing actuator of claim 8 wherein said pinion bearing assembly comprises upper and lower bearings (9d).

10. The bearing actuator module of claim 1 comprising a brake assembly; said brake assembly comprising a rotor brake plate (12) operatively supported by said motor rotor (13) and a stator brake plate (18) operatively supported by said motor stator (14); said rotor and stator brake plates (12 and 18) having facing surfaces provided with teeth (12a, 18a), said rotor and stator brake plates being normally biased into a meshing position by a spring member 16; said brake assembly further

including an actuator operable to overcome the force of the spring member 16.

11. The bearing actuator module of claim 10 wherein said brake assembly actuator comprises at least one pole plate (19, 20) and an energizable coil (21), said coil (21) being selectively energizable between an energized state wherein said coil magnetizes said at least one pole plate to magnetically attract one of said stator and rotor brake plates to disengage said stator and rotor support plates.

12. The bearing actuator module of claim 11 including a controller; said controller being in electrical communication with both said motor (M) and said brake actuator to selectively and actuate and deactivate said motor (M) and brake assembly actuator; whereby, said controller activates said brake assembly actuator prior to activating said motor (M)

13. The bearing actuator module of claim 1 including a feedback device (FD); said feedback device comprising a scale (24) and a pickup (23); one of said scale and pickup being operatively connected to said carrier (10) and the other of said scale and pickup being maintained generally stationary; said feedback device generating a signal indicative of the rotational position of said carrier based on the relative position of said pickup (23) relative to said scale (24).

14. The bearing actuator module of claim 1 wherein the motor stator is positioned at an axial outer position such that said stator will dissipate heat to the atmosphere.

15. A bearing actuator module comprising:

a mounting ring (1) adapted to be mounted to an application support structure;

a rotatable gear (G2) received within said mounting ring for rotation relative to said mounting ring; and

a drive subassembly (DS) comprising:

a stationary gear (G1);

a carrier (10) mounted in said module to be rotatable relative to said inner gear (G1);

a plurality of pinion assemblies (9) mounted on said carrier (10); each pinion subassembly comprising a first pinion (P1) positioned to mesh with said stationary gear (G1) and a second pinion (P2) positioned to mesh with said rotatable gear (G2); said first and second pinions being connected to rotate together;

a motor (M) operatively supported by said stationary gear (G1); said motor comprising a rotor (13) and a stator (14); said stator being operatively connected to said stationary gear (G1) and said rotor (13) being operatively connected to said carrier (10); whereby activation of said motor, through the meshing engagement of the gears (G1 and G2) and the pinions (P1 and P2) will induce rotational motion in said rotatable gear (G2);

a brake assembly comprising a stator brake plate, a rotor brake plate, and an actuator; said stator brake plate being operatively connected to said stator; said rotor brake plate being operatively connected to said rotor; said rotor and stator brake plates comprising facing toothed faces; said stator and rotor brake plates being selectively movable between a first position in which said brake plates are in meshing engagement and rotation of said rotatable gear is prevented and a second position in which said brake plates are disengaged from each other to permit rotation of said rotatable gear (G2); said brake plates being normally biased to one of said first and second positions by a spring member; said actuator being operable move said brake plates between their first and second positions; and

a controller in electrical communication with said brake assembly actuator and said motor to selectively activate and deactivate said brake assembly actuator and motor; whereby said controller activates said brake assembly actuator to move said

brake plates to their second position prior to activation of said motor.

16. The bearing actuator module of claim 15 further including a positional sensor which generates a signal indicative of the rotational position of said carrier (10); said controller controlling said brake assembly actuator and said motor in response to signals received from said feedback device.

17. A modular bearing actuator comprising:

a support bearing assembly having a mounting ring adapted to be mounted to a support structure and ring gear mounted to said mounting ring to be rotatable relative to the mounting ring; said ring gear being adapted to have a load mounted thereto; and

a drive subassembly; said drive subassembly being removably received in said support bearing assembly; said drive assembly comprising a motor section having a motor and gear box driven by said motor; said gear box meshing with said support bearing assembly ring gear to rotatably drive said ring gear.

18. The modular bearing actuator of claim 17 and further comprising a top plate extending from said support bearing assembly ring gear over said drive sub-assembly and an inner plate extending axially from said top plate along an inner surface of said drive sub-assembly; whereby, said top plate and inner plate seal said actuator against the loss of lubricant and against the ingress of contaminants.

19. The modular bearing actuator of claim 18 and further comprising an inner seal between said gear box and said motor section and an outer seal between said gear box and said motor section.

20. The modular bearing actuator of claim 19 wherein said outer seal comprises a sealed bearing.

21. A bearing actuator module comprising:

a mounting ring (1) adapted to be mounted to an application support structure;

a rotatable gear (G2) received within said mounting ring for rotation relative to said mounting ring;

a drive subassembly (DS) comprising a gear box and a motor; said motor comprising a stator and a rotor; said gear box being drive by said motor and engaging said rotatable gear to drive said rotatable gear; and

a brake assembly comprising a stator brake plate, a rotor brake plate, and an actuator; said stator brake plate being operatively connected to said stator; said rotor brake plate being operatively connected to said rotor; said rotor and stator brake plates comprising facing toothed faces; said stator and rotor brake plates being selectively movable between a first position in which said brake plates are in meshing engagement and rotation of said rotatable gear is prevented and a second position in which said brake plates are disengaged from each other to permit rotation of said rotatable gear (G2); said brake plates being normally biased to position one of said first and second positions by a spring member; said actuator being operable to move said brake plates between their first and second positions.

22. The modular bearing actuator of claim 21 wherein said brake plate teeth have sloped side surfaces.

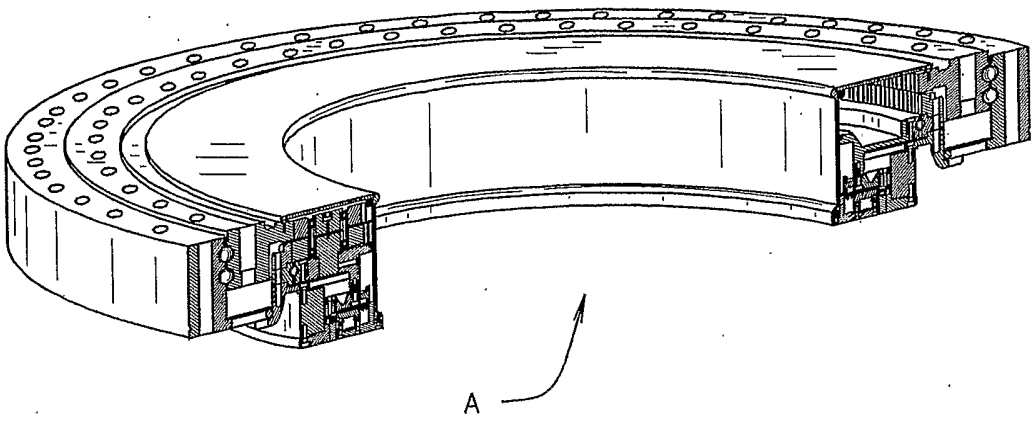

A

FIG. 1

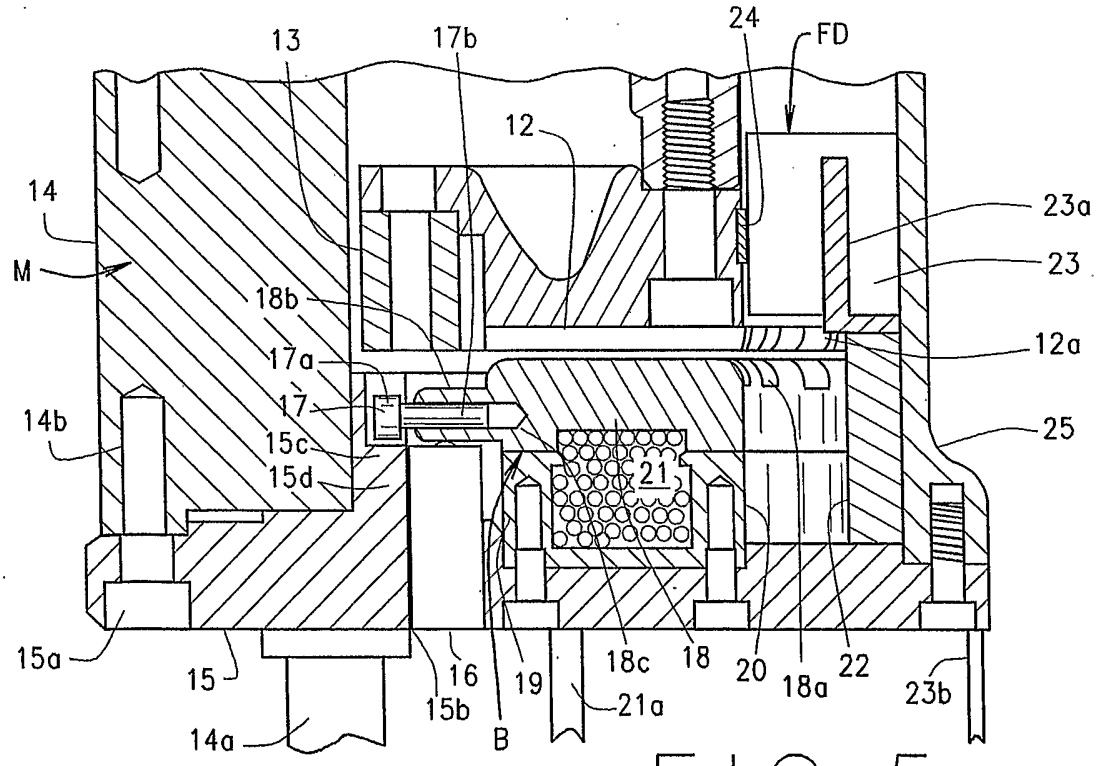

13
17b
24
FD
12
14
M
23a
23
18b
12a
17a
17
14b
25
15c
15d
21
15a
15
16
19
18c
18
20
18a
22
23b
15b
B
21a
14a

FIG. 5

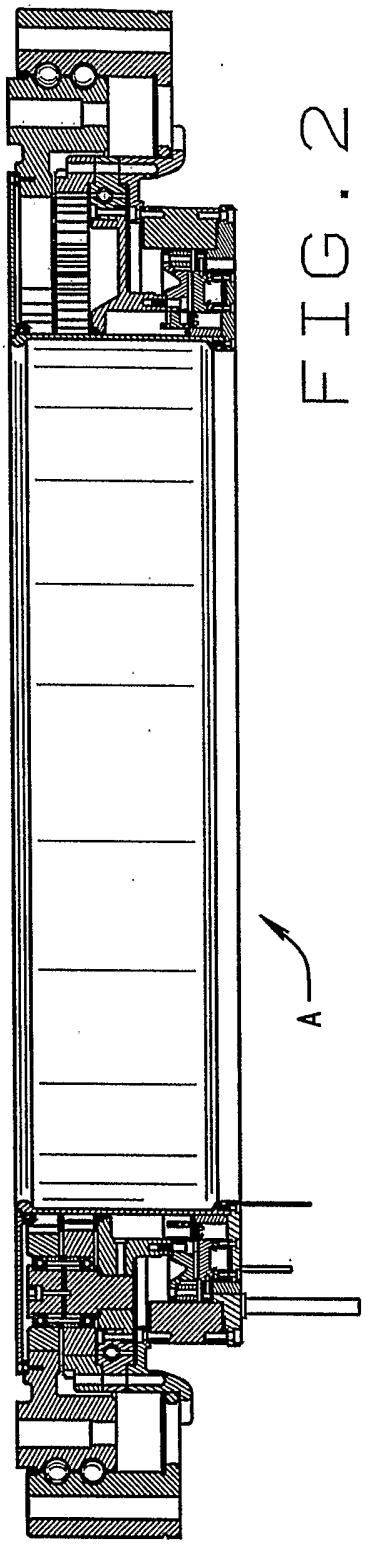


FIG. 2

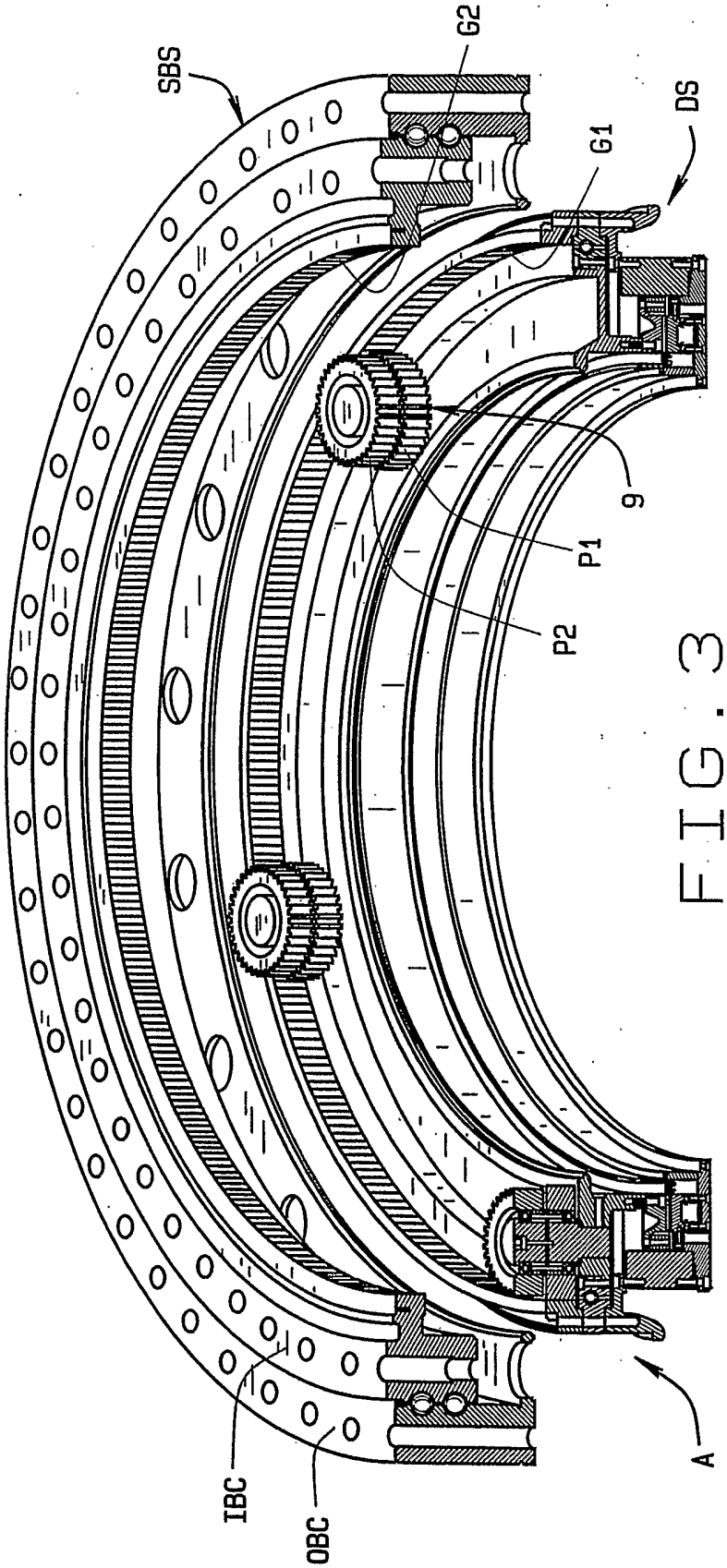


FIG. 3

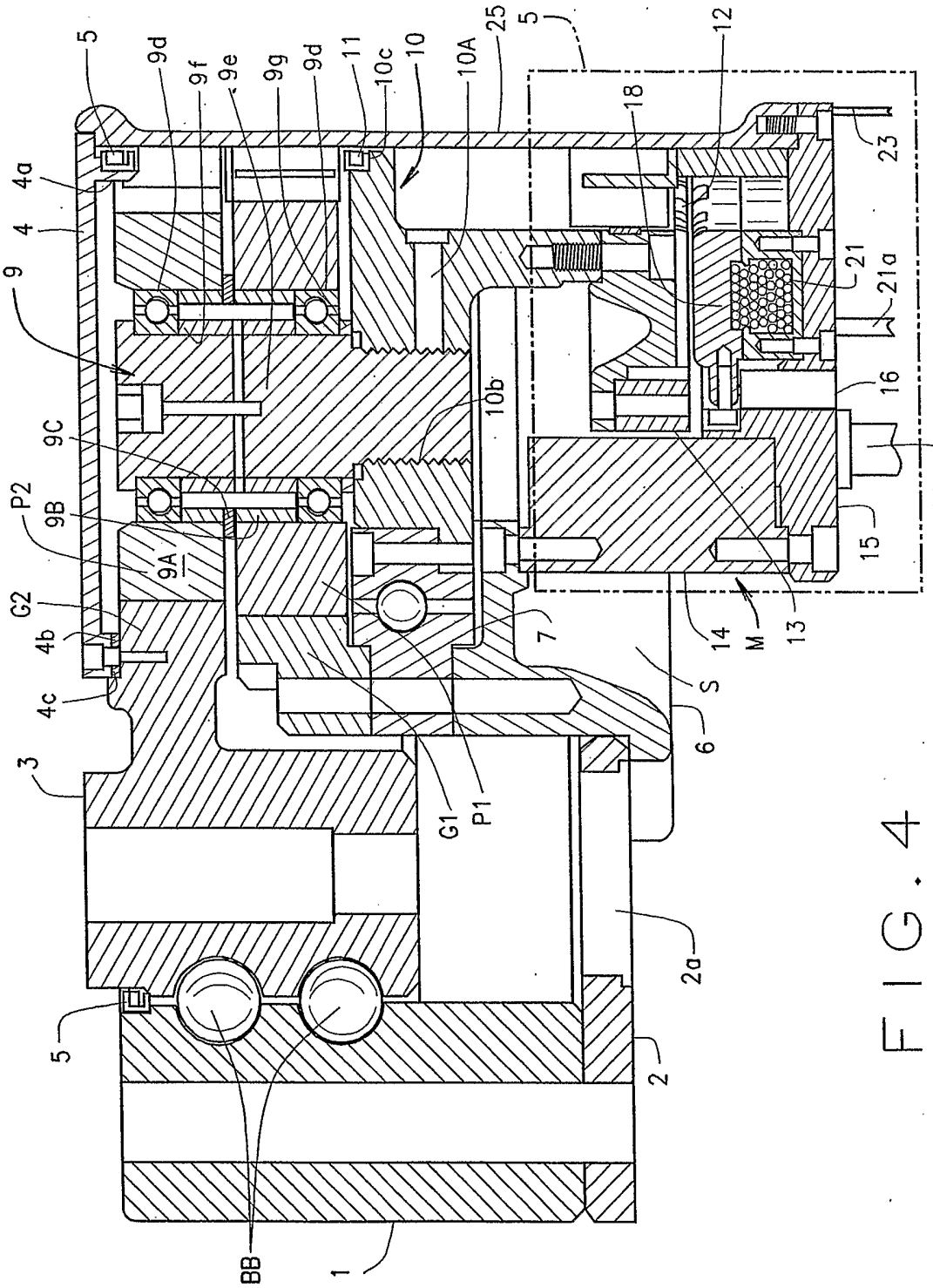

5
9d
9f
9e
9g
9d
11
10c
10
10A
25
5
12
18
23
4a
4
9
21
21a
16
10b
9C
P2
9B
9A
14a
15
G2
4b
4c
3
7
14
M
13
S
6
G1
P1
2a
5
2
BB
1

FIG. 4

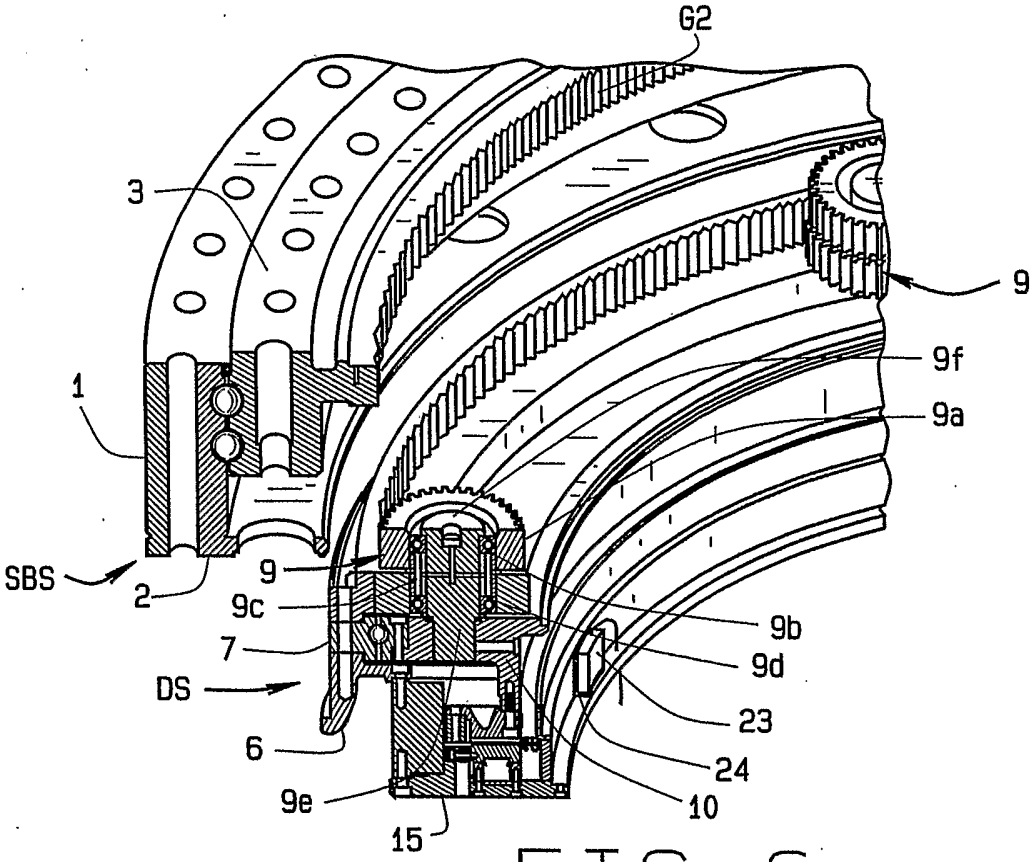

G2
3
9
9f
1
9a
SBS
2
9
9c
9b
7
9d
DS
23
6
24
9e
10
15

FIG. 6

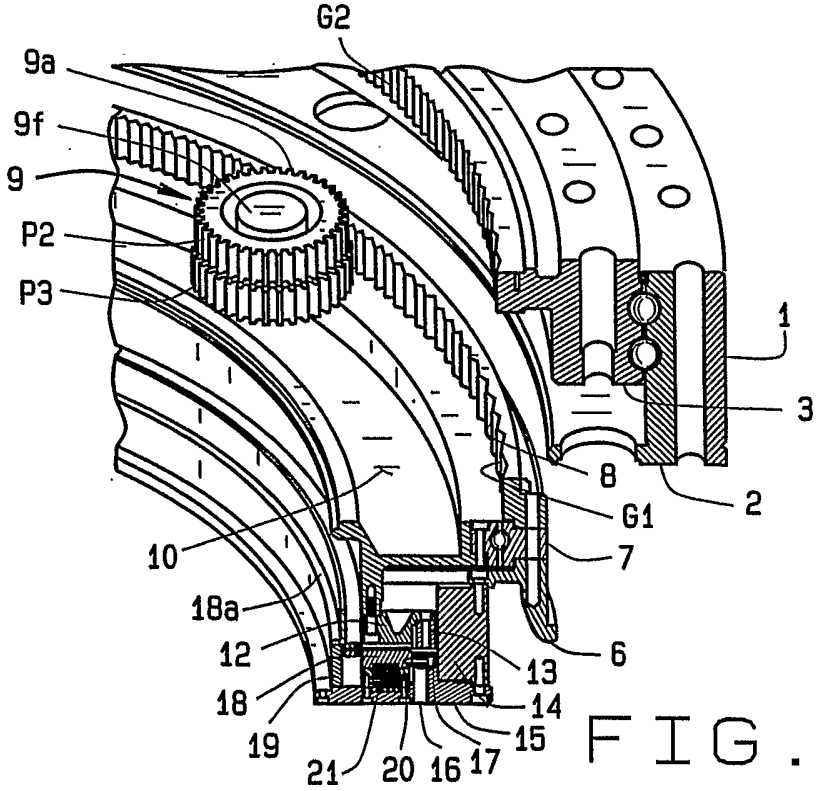

G2
9a
9f
9
P2
P3
1
3
8
2
G1
10
7
18a
6
12
13
18
14
19
15
21
20
16
17

FIG. 7

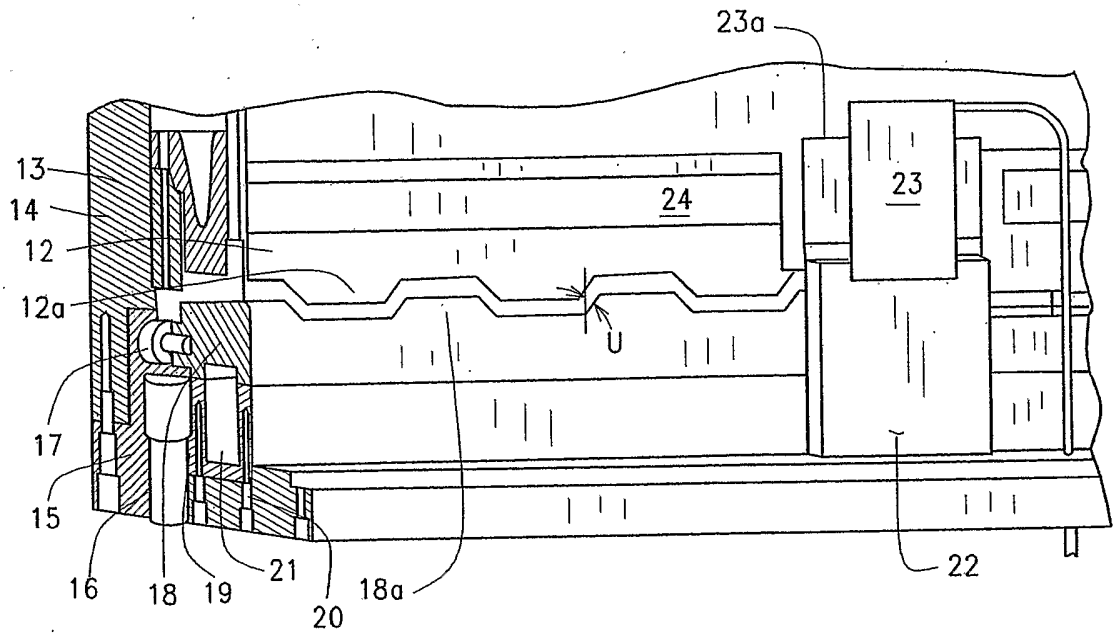

23a
13
14
12
12a
17
15
16
18
19
21
20
18a
24
U
23
22

FIG. 8

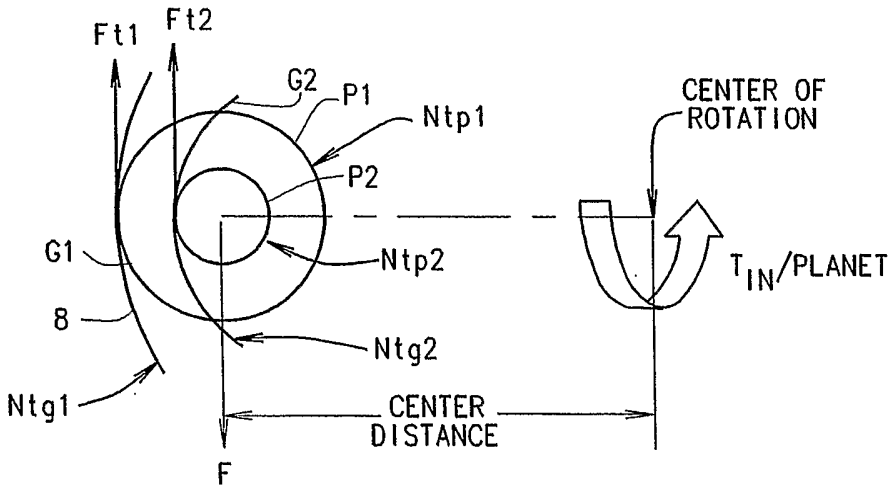

Ft1
Ft2
G2
P1
Ntp1
P2
G1
Ntp2
8
Ntg2
Ntg1
CENTER DISTANCE
F
CENTER OF ROTATION
T_{IN}/PLANET

FIG. 9

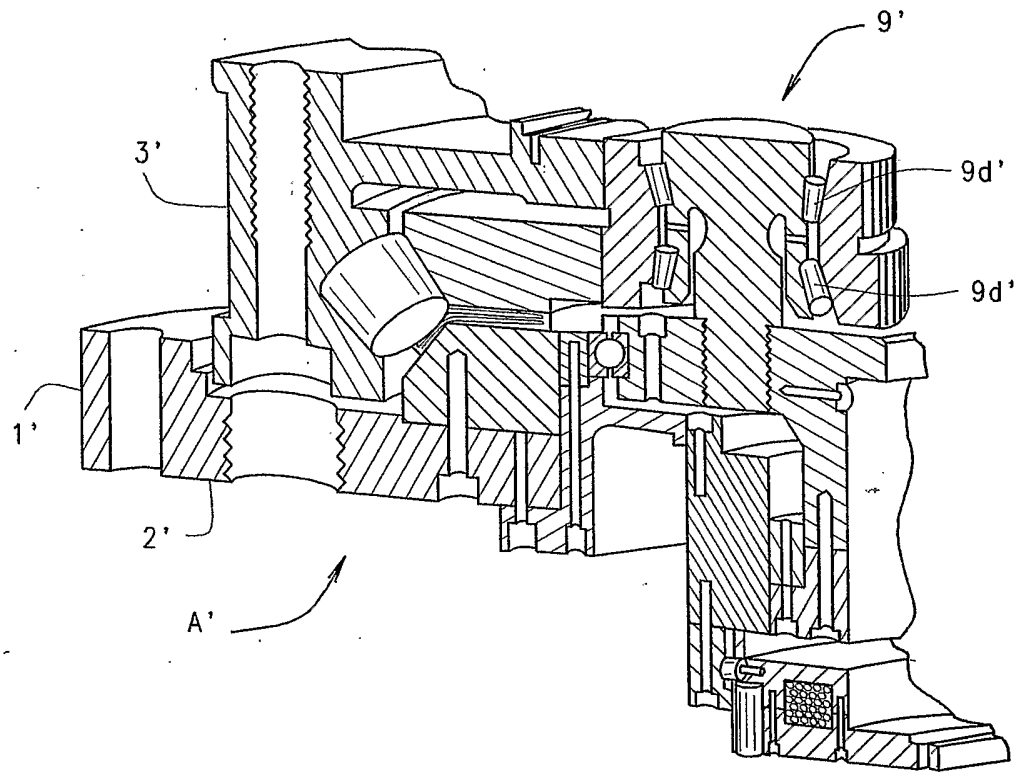


FIG. 10