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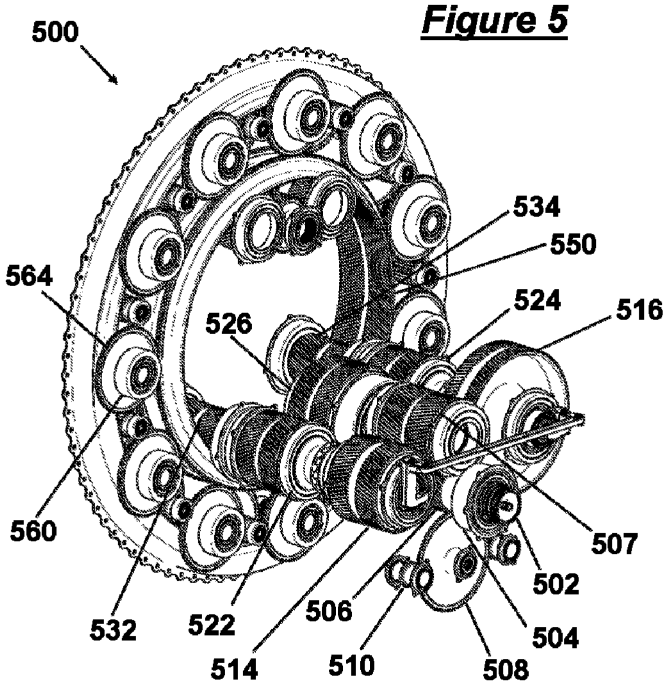
(54) **Title**: TORQUE BALANCING GEARBOX

Figure 5



(57) **Abstract**: A gearbox comprising a gearset is able to transmit high output torques at high numerical reduction ratios at a power-to-weight ratio higher than previously attainable with existing designs. A distributor gear is disposed relative to a spur gear in order to produce automatic torque balancing. The distributor gear can be advantageously configured as an input floatring gear, for which support in all directions is provided by gear tooth mesh forces rather than bearings. Automatic torque balancing is achieved by configuring the distributor gear with first and second rows of helical teeth on the external circumference, and with a set of double helical teeth disposed on the internal circumference of the distributor gear. This allows for the placement of up to 50 or more planet gears in a countershaft arrangement with one end having helical teeth and the other end having spur teeth.

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TORQUE BALANCING GEARBOX

[0001] This application claims priority to US provisional application ser. no. 61/034462, filed March 6, 2008, the disclosure of which is incorporated herein by reference in its entirety.

Field of the Invention

[0002] The field of the invention relates to the transmission of power by gears.

Background

[0003] When gas turbine or turboshaft engines are employed to drive a plant, machinery, or a vehicle, a high numerical reduction ratio is frequently required because of the high output speed (often measured in rotations per minute or rpm) of the turbine. Power transmission of several thousands of horsepower is encountered in many applications. In the case of a stationary plant, or in marine applications, mechanical reliability can be readily achieved if the weight of the gearbox is not critically important. However, with propeller drives for aircraft or rotor drives for helicopters, weight and efficiency of the gearbox are critically important. This requirement led to the widespread adoption of planetary or epicyclic gearboxes in flight applications. Planetary gearboxes achieve their weight advantage over simple gear trains of the same ratio by virtue of increasing the number of mesh points, and hence load-carrying gear engagements, in a given circumferential length of gearing.

[0004] With increasing scale and power transmission capacity, the weight of a gearbox increases approximately as a cube function of linear size because the steel elements of the gears span the entire radial distance from the center of rotation to the periphery of the largest gear, usually a ring gear. The tangential force resisting a torque is inversely proportional to the distance from the center of rotation, thus it is clear that while gear tooth loading from tangential force decreases with radius, weight increases disproportionately.

[0005] A conventional prior art planetary gear assembly **100** is shown in **Figure 1**. Four planet gears **120**, **122**, **124**, **126** orbit and mesh with a central sun gear **130**. The planet gears **120**, **122**, **124**, **126** are constrained within and mesh with a ring gear **110**.

[0006] A prior art planetary reduction gearbox system **200** is shown in **Figure 2**, adapted from Dudley's Gear Handbook, McGraw-Hill, 1991. **Figure 2** shows a rotating planetary carrier **254** and a fixed annulus, ring gear **240** having teeth **242**. An input shaft **210** rotates as shown by arrow **212** and is connected to a sun gear **214**. The sun gear meshes with a plurality of planet gears **220**, **230** having teeth **222**. The planet gears **220**, **230** are on shafts **224**, **234** fixed to a planet carrier **250**. The planet carrier rotates as shown by arrow **254** and is fixed to an output shaft **252**. A double planetary system (not shown) could be formed by combining the gearsets of Figures 1 and 2, where the first stage planetary is as gearset **200**, but the rotating planet carrier **250** is attached to a sun gear **130** of the second planetary assembly **100**. The second planetary assembly **100** usually has a rotating carrier attached to a final output shaft which could be a rotor shaft in a typical helicopter transmission. In the case of a helicopter planetary, the reduction ratio of each planetary reduction is usually limited to 3.875 because that is all that can be achieved with a maximum of six planets revolving around the sun gear.

[0007] Such prior art planetary gearbox systems tend to become extremely heavy and thus impractical for aircraft applications when scaled to sizes commensurate with large transport aircraft. Thus, a need remains for a highly efficient, light weight gearbox system for aircraft applications.

Summary Of The Invention

[0008] The present inventive material provides systems, methods, and apparatus in which a gearbox comprising a gearset is able to transmit high output torques at high numerical reduction ratios at a power-to-weight ratio higher than previously attainable with existing designs. These high power-to-weight ratios are achieved through the use of a large array of load sharing gears.

[0009] In preferred embodiments, a distributor gear is disposed relative to a spur gear in order to produce automatic torque balancing. The distributor gear can be advantageously configured as an input floatring gear, for which support in all directions is provided by gear tooth mesh forces rather than bearings. It is further contemplated that the distributor gear could be configured with first and second rows of helical teeth oriented to opposite sides of the gear and disposed on the external circumference of said gear, with a set of double helical teeth disposed on the internal circumference of the distributor gear.

[0010] Such a preferred configuration could advantageously allow for the placement of at least 6, 10, 20, 30 or 50 planet gears that mate with the first row of helical teeth as well as at least 6, 10, 20, 30 or 50 planet gears that mate with the second row of helical teeth. It is contemplated that these planet gears could further be placed in a countershaft arrangement whereby some face forward and some face aft, thereby distributing load more evenly and allowing for the geometric fit of more planet gears.

[0011] It is further contemplated that each planet gear could be configured with one end having helical teeth and the other end having spur teeth. In preferred embodiments, the end with helical teeth would mate with the distributor gear, while the end with spur teeth would mate with a spur gear that is also an output internal ring gear. Further, the helical and spur gears could be advantageously sized and dimensioned to produce a reduction of at least 4:1, 10:1, 20:1, 50:1, or even 80:1. In especially preferred embodiments, a rotorcraft is equipped with the gearset as described above.

[0012] Viewed from another aspect, the present inventive subject matter allows for a gearbox to yield a power-to-weight ratio of at least twice that of a corresponding prior art double planetary reduction system. This reduction in power-to-weight ratio is achieved by having a number of planet gears disposed about a distributor gear, where the number of planet gears is at least 12, 20, 40, or even 60. It is contemplated that the planet gears could be sized to collectively transfer at least 5000, 20,000, or even 50,000 horsepower.

[0013] The planet gears can advantageously be manufactured with a helical tooth pattern, and could be disposed on opposite sides of the ring gear in a countershaft arrangement. In more preferred embodiments, the planet gears are disposed to transmit power to an output ring gear having internal spur teeth. Further, in especially preferred embodiments, the planet gears, distributor gear, and output ring gear are sized and dimensioned to yield a power-to-weight ratio of at least three times that of the corresponding double planetary reduction system.

[0014] Various objects, features, aspects and advantages of the inventive subject matter will become more apparent from the following detailed description of preferred embodiments, along with the accompanying drawing figures in which like numerals represent like components.

Brief Description Of The Drawing

[0015] **Figure 1** is a schematic of a prior art planetary gearset.

[0016] **Figure 2** is a perspective illustration of a prior art planetary gearset with a rotating planet carrier.

[0017] **Figure 3** a schematic of a preferred tiltrotor aircraft having a gearbox and rotor.

[0018] **Figure 4** is a schematic of a preferred reduction gearset layout.

[0019] **Figure 5** is a perspective view of an alternate preferred drivetrain.

[0020] **Figure 6** is an alternate perspective view of the gearbox of Figure 5.

[0021] **Figure 7** is a perspective view of a torsionally compliant quill shaft.

[0022] **Figure 8** is a perspective view of a distributor gear, advantageously configured with helical teeth in such a way as to be an input floatring gear.

Detailed Description of The Invention

[0023] **Figure 3** is a schematic top view illustration of a preferred tiltrotor aircraft **300**. The aircraft comprises a fuselage **302**, wing **304**, first rotor system **310**, and second rotor system **320**. The second rotor system **320** is shown in a vertical orientation, consistent with helicopter-mode flight. The first rotor system **310** is shown in a horizontal orientation, consistent with airplane-mode cruise flight. In practice, the first rotor system **310** and second rotor system **320** are likely to have a substantially similar orientation at any given time in flight.

[0024] A rotor system **310** comprises a hub **314** coupled to a tilting nacelle **312**, which tilts with respect to the wing **304**. A rotor blade **316** is coupled to the rotor hub **314**. An engine **350** is preferably disposed within the tilting nacelle **312** and is coupled to a shifting gearbox **370**. The shifting gearbox is coupled to a numerical reduction ratio reduction gearbox **180**. The reduction gearbox **380** is coupled to and drives the rotor hub **314**. A miter gearbox **360** is also coupled to the shifting gearbox **370** as well as a cross-wing driveshaft **362**. The cross-wing driveshaft **362** is preferably disposed within the wing **304** and distal ends of the cross-wing driveshaft **362** are preferably coupled by a mid-wing gearbox **364**. The cross-wing driveshaft **362** serves to transmit power from an engine **350** in a first tilting nacelle **312** to a rotor system **320** on the opposite side of the aircraft **300**.

[0025] An especially preferred shifting gearbox **370** is described in US provisional patent application ser. no. 61/034457, “Rotorcraft Engine and Rotor Speed Synchronization”, filed March 6, 2008. An especially preferred tiltrotor rotor system **310** is described in U.S. Patent 6641365 to Karem. This and all other extrinsic materials discussed herein are incorporated by reference in their entirety. Where a definition or use of a term in an incorporated reference is inconsistent or contrary to the definition of that term provided herein, the definition of that term provided herein applies and the definition of that term in the reference does not apply.

[0026] **Figure 4** is a schematic depiction of a preferred tiltrotor drivetrain **400**. An engine **350** is coupled to an engine input shaft **402**. A sprag clutch **404** is torsionally coupled between the engine input shaft **402** and an engine pinion gear **406**. The engine pinion gear **406** meshes with both a high output speed shift gear **414** and low output speed shift gear **416**. Through the use of a first dog clutch **418** and second dog clutch **420**, only one of the high output speed shift gear **414** and low output speed shift gear **416** transmit power to the first torque split gear **422** or second torque split gear **424**. The two torque split gears **422**, **424** are connected through a torque split idler gear **426**, which assists in sharing input torque.

[0027] An accessory gear **408** meshes with the input selection gear **406** and powers accessories such as an oil pump or generator **412** through a generator gear **410**. Thus, a generator **412** is torsionally coupled in the drivetrain **400**, and is preferably sized to produce at least 10, 50, 100, 250, or even 500 kilowatts of electrical power.

[0028] Power is shared between and transmitted by first and second torque split gears **422**, **424** to first and second distributor gear pinions **432**, **434** by means of first and second quill shafts **428**, **430**. A quill shaft **428** is a torsionally compliant member coupled between a torque split gear **422** and a distributor gear pinion **432**. The opposite ends of the quill shaft **428** are advantageously splined for connection with a gear with a different number of spline teeth on either end of the quill shaft **428**. In preferred embodiments, one end of the quill shaft **428** has one more tooth than the other, which allows for a precise Vernier-effect calibrated installation of torque split gear **422** and distributor gear pinion **432** and ensures that the desired precision of mesh of the distributor gear pinion **432** with distributor ring gear can be achieved. Thus, a quill shaft **428** is intermediately torsionally coupled in the drivetrain **400** between a torque split gear **422** and a distributor gear pinion **432**. The stiffness of the quill shaft **428** can be tuned to achieve desired gearbox dynamic characteristics.

[0029] The first and second distributor gear pinions **432**, **434** transmit torque to a distributor gear **450**. The distributor gear **450** is an input floatring gear. As used herein, a “floatring gear” refers to a ring gear that is not directly supported by bearings or directly carried on a shaft. Instead, the “floatring gear” is supported by tooth forces at gear mesh interfaces, and thus floats. To achieve effective floating support of a distributor gear **450**, the mesh faces and interfacing gears must be configured in specific manners.

[0030] The distributor gear **450** achieves floating support through advantageous configuration of an internal face **452**, a left- or right handed first external helical interface **456**, and an opposite-handed second external helical interface **458**. It is contemplated that the first and second external helical interfaces can be spaced further apart on centerline than twice the face width of either external helical interface. A plurality of planet gears **460**, **462** orbit the distributor gear **450**. A planet gear **460** is configured with a single helical interface **464** to mesh with the distributor gear **450** on one end, and with a spur gear **466** on the distal end. Some planet gears may be placed in an outward configuration **462**, while other planet gears may be placed in an inward configuration. An outward configuration of a planet gear **462** is one in which the larger diameter end of the planet gear is positioned away from the power source **350**. The combination of planet gears **460**, **462** in both outward and inward configurations between a common distributor gear **450** and output ring gear **470** advantageously allows for more planet gears and hence greater torque transmission with minimal weight increase, thereby increasing the power-to-weight ratio of the drivetrain **400**.

[0031] In a corresponding double planetary system of the of the prior art, which achieves the same reduction ratio, one would need much thicker planetary gears to accomplish an equivalent torque transmission without breaking gear teeth. This is because in such a prior art system speed reduction is accomplished in two smaller steps, which at least doubles the gear weight relative to a gearbox that uses a floatring gear. Thus it can be seen that in preferred embodiments, a gearbox having a number of planet gears disposed about a distributor gear in a manner that yields a power-to-weight ratio of at least twice that of a corresponding double planetary reduction system that uses the same materials and has the same load capability.

[0032] Any suitable number of planet gears **460**, **462** can be utilized, but preferably at least 6, 8, 10, 12, 15, 20, 22, or more such gears. Unless the context dictates the contrary, all ranges set forth herein should be interpreted as being inclusive of their endpoints. Similarly, all lists of values should be considered as inclusive of intermediate values unless the context indicates the contrary. The planet gears are preferably sized to collectively transfer at least 3,000, 5,000, 10,000, 20,000, 30,000, 40,000, 50,000 horsepower or even more. A currently preferred version operates at 9,400 horsepower.

[0033] In preferred embodiments, the spur gear portion of the planet gears **460**, **462** mesh with and transmit power to an output ring gear **470** which has an internal gear mesh. The output ring gear **470** is advantageously coupled to a rotor hub **314** by means of a flex coupling **472** that performs a strain isolation function. A first blade **316** and second blade **318** are advantageously coupled to the rotor hub **314**.

[0034] All suitable reduction ratios are considered, but in especially preferred embodiments, gears are sized and configured to achieve a speed reduction from at least 4:1 to 100:1 or more. Such large reductions are contemplated to be especially useful for helicopters and other rotorcraft, with a currently preferred embodiment having a ratio of 4.8 to 1 for the distributor stage reduction, and a ratio of 55.2:1 for the gearbox in helicopter mode and 96.5 to 1 in airplane-mode cruise flight. As used herein, the term "gearset" means the gears inside a gearbox. The final reduction ratio of a gearset is largely influenced by the ratio of diameters of the distributor gear **450** which is preferably a helical gear, the planet gears **460**, **462**, and the output ring gear **470**, which is preferably a spur gear. The helical and spur gears could be advantageously sized and dimensioned to produce a reduction of at least 4:1, 10:1, 20:1, or even 50:1.

[0035] Preferred embodiments achieve these reduction ratios from a maximum engine input speed of 4,000 to 30,000 revolutions per minute. A currently preferred configuration has an output ring gear **470** with 374 internal teeth, but all suitable numbers of teeth are contemplated, including 100, 300, 500, 700, 900, and even 1,100 teeth. A currently preferred output ring gear **470** has a diameter of approximately 5 feet, but all suitable diameters are contemplated including 2, 4, 6, 8, 10, 16, and even 20 feet. In especially preferred embodiments, the distributor gear pinions **432**, **434** and the distributor gear **450** are advantageously disposed within the inside diameter of the output ring gear **470**. In currently preferred configurations, a pinion gear **460** has 20 teeth on the spur end **466** and 73 teeth on the single helical end **464**. However, all suitable numbers of teeth are contemplated, including 10, 25, 50, and 75 on the spur end, and 20, 50, 100, and 200 teeth on the single helical end **464**.

[0036] A person skilled in the art will recognize that the drivetrain **400** achieves effective load sharing and torque splitting through the advantageous combination of several mechanisms. One such mechanism is the use of a plurality of torque split gears **422**, **424** connected through a torque split idler gear **426**, which allows all torque split gears **422**, **424** to transmit power even though not all of the torque split gears **422**, **424** might be powered at a given time. Another mechanism is the use of quill shafts **428**, **430** with tailored torsional compliance to allow for precision mesh and load sharing between torque split gears **422**, **424**, first and second distributor gear pinions **432**, **434**, and the distributor gear **450**. A further mechanism arises from the configuration of the distributor gear **450** as a floatring gear and the tooth profile shape of the gear teeth on the planet gears **462**. The load sharing and torque splitting features incorporated in the gearbox nearly equally share loads and reduce weight and increase the power-to-weight ratio by evening the loads on gear teeth in the drivetrain **400**. As used herein, to "nearly equally share" a load means that components share a load within 15% of each other.

[0037] Viewed from another perspective, the present inventive material can enable alternate preferred embodiments, such as the drivetrain **500** shown from opposite angles in **Figure 5** and **Figure 6**.

[0038] In this preferred embodiment, a turbine engine provides the input to the gearbox through an engine pinion gear **502**. Power is then transferred to a drive gear **506**. In the case of left and right handed assemblies a reversing idler gear **507** can be added. In especially preferred embodiments there can be one or more drive gears for driving in more than one mode. For example a helicopter mode gear at a high output speed, low-ratio ratio gear **514** and an airplane cruise low output speed, high-ratio gear **516**. In other preferred embodiments there can be more than one engine input, with the idler and drive gears described above replicated for each additional engine installation. Further, an accessory gear **508** and generator gear **510** can be included in the drivetrain.

[0039] Distributor pinion gears **532**, **534** are coupled in the drivetrain through a torque split gears **522**, **524** and a torque split idler gear **526**. The distributor pinion gears **532**, **534** then engage a large diameter floatring distributor gear **550**. The purpose of this gear **550** is to create a large diameter about which small, lightweight driven planet gears **560**, **562** can be distributed to share the output torque load. This distributor gear **550** has two single rows helical tooth pattern on the external circumference **556**, **558**, meaning that any planet gear **560**, **562** driven off these tooth patterns on the external circumference **556**, **558** only engages with one row of teeth. Thus, the driven end **564** of planet gear **560** engages only with the single row **558** of distributor gear **550**. By contrast, the distributor gear **550** has a double row helical tooth pattern on the internal circumference **552**, **554**, meaning that the driving distributor pinion gears **532**, **534** each engage with both a forward helical tooth pattern **554** and aft helical tooth pattern **552**. This outer diameter drives a large array of alternating forward planet gears **560** and forward planet gears **562**.

[0040] Advantageously, a quill shaft **528**, as shown in **Figure 7** can be intermediately torsionally coupled between a torque split gear **522** and a distributor pinion gear **532**. The stiffness of such shaft **528** can be tailored to allow precision assembly and meshing between distributor pinion gear **532** and distributor gear **550**. It is contemplated that the quill shaft **528** can be configured to have a first spline **710** on a forward end of the shaft **528** and a second spline **720** on an aft end. The number of teeth on the splines **710**, **720** is advantageously different, preferably by a single tooth to allow for precision installation.

[0041] The helical tooth form on the distributor gear **550** and planet gears **560**, **562** results in an axial load being imparted to the planet gears **560**, **562**. The alternating forward and aft driven helical gears are placed such that the helical tooth form will drive them each toward the distributor gear **550**, thus balancing the net force on the gear system and allowing the distributor gear **550** to float and thus act as an input floatring gear. Backlash in the gear system is taken up by applying an axial load to an individual driven planet gear so that its helix engages that of the external teeth of the distributor gear **550**. It is this adjustment of backlash in each driven gear that allows the system to evenly load share across an array of driven planet gears. By evenly load sharing across a large array of driven planet gears **560**, **562**, the net torque carried by each is reduced.

[0042] Conducting the largest change in gear ratio in the final step of reduction confines the large torque increase associated with this decrease in speed, to the large load sharing array of driven planet gears, thereby reducing the weight of the system. In the preferred embodiment, the axial load imparted to the array of driven planet gears **560**, **562** is reacted through the use of bearings **563** attached to each end of the driven planet gears such that the loads are reacted through bearings to a fixed housing. The array of driven planet gears **560**, **562** is positioned during assembly so they will equally share the load. This enables the array of driven planet gears **560**, **562** to dynamically control their axial displacement and balance their engagements to the distributor gear **550**. In large torque capacity gearbox applications such as this, it is this large distributed torque splitting capability that reduces overall system weight and increases reliability.

[0043] Located on the same shaft as each helical driven gear **560**, is a drive pinion gear **566**. An exemplary drive pinion **566** engages an output internally toothed ring gear **570** which in turn drives the rotor system. The toothed ring gear **570** is preferably a spur gear, meaning it has teeth in spur pattern. In the preferred embodiment, all gears have their rotating axis in the same direction as that of the output gear and the engine pinion. In other preferred embodiments the engine pinion can be orthogonal to the output gear and other gears in the drive system. This would accommodate, for instance, a vertical helicopter drive system with a horizontally mounted engine input. Thus, the preferred embodiment can be viewed as a gearset **500** having a helical distributor gear **550** that is a floatring gear disposed relative to a spur gear **570** to produce automatic torque balancing by means of planet gears **560**, **562** equipped with both helical teeth **564** and spur teeth **566**.

[0044] It is thus contemplated to provide a gearset having a helical gear disposed relative to a spur gear to produce automatic torque balancing. This can be advantageously accomplished where the helical gear is an input floatring gear, and preferably where the helical gear has first and second rows of helical teeth oriented to opposite sides of the gear. An exemplary input floatring distributor gear **550** is shown in **Figure 8**. First and second single rows of helical teeth **556**, **558** are oriented to opposite sides of the gear **550**, and disposed on the external circumference of said gear **550**. A double row of helical teeth, comprising left-handed teeth **552** and right-handed teeth **554** is disposed on the internal circumference of the gear **550**. A plurality of aft planet gears, exemplified by gear **562**, can mate with the first row of helical teeth **556**. Similarly, a plurality of forward planet gears, exemplified by gear **560**, can mate with the second row of helical teeth **558**. All suitable numbers of such planet gears are contemplated, including at least 6, 10, 16, 20, 30, and 40 planet gears mating with either the first or second row of helical teeth.

[0045] As used herein, the term "double row" of teeth is distinguished from "two rows" of teeth. A "double row" of teeth is used to indicate that both of the double rows simultaneously meshes with a single mating gear. The notation "two rows" of teeth is used where each of the two rows meshes with a different gear.

[0046] A planet gear **560** preferably has both helical teeth **564** to mate with the distributor gear **550**, and spur teeth **566** to mate with an output ring gear **570**. The planet gears **560**, **562** are preferably disposed in pairs on opposite sides of the helical gears **556**, **558**. Ideally, a gearbox **500** would have a number of planet gears **560**,**562** disposed about the input ring gear

550 in a manner the yields a power-to-weight ratio of at least twice or three times that of a corresponding conventional double planetary reduction system that uses the same materials and has the same load capability.

[0047] The spur teeth **566** of a planet gear **560** advantageously mate with an output internal ring gear **570**. The helical and spur gears can have any suitable relative sizes, dimensions, and positions, but preferably cooperate to produce any suitable reduction.

[0048] All suitable materials are contemplated for the various gears. For example, the gears are manufactured from carburized and nitrided aerospace steels that come from vacuum arc re-melt steel bar. Additional gear processing include shot peening and finish grinding them to AGMA Quality 13 standards. Similarly, all suitable tooth designs are contemplated for the helical and spur gear teeth. Currently, the most preferred helical and spur tooth design starts with a basic design that conform to the AGMA standard design system but have modified tooth profiles with the appropriate tip and flank modification as well as longitudinal tooth modifications to maximize load carrying capability and life as well as operate efficiently and quietly.

[0049] Methods are especially contemplated for increasing a power-to-weight ratio in a rotorcraft, such methods comprising: utilizing a gearbox that transmits torque from a floating input ring though a plurality of assembled interconnect gears to a floating output gear; and using teeth patterns on opposite sides of the input ring such that operation of the gearset transmits less than 0.01%, 0.001%, and even 0001% of an operating load axially or radially to a wall of the gearbox. This can be advantageously accomplished where the input gear has helical teeth oriented in opposite directions on opposite sides of the input ring, and independently where the assembled interconnect gears have both helical teeth and spur teeth.

[0050] Such methods are contemplated to be especially useful where the input ring is sized to transmit at least 3,000, 5,000, 10,000, 20,000, 30,000, 40,000 50,000 horsepower or even more. It is especially advantageous when carrying such large loads that the methods can be implemented while eliminating a need for a bearing to support the output gear.

[0051] During assembly, a gearbox according the inventive concepts herein can be fabricated by: providing a helical ring gear and an internal spur ring gear; axially positioning each of a plurality of distributor planet gears (countershaft gears) relative to the helical and spur gears; rotating the input gear such that different ones of the countershaft gears will move to slightly different axial positions due to their respective manufacturing tolerance differences, and whereby individual radial and axial loads from the countershaft gears are balanced to produce substantially no collective axial or radial load on the output gear; and clamping the countershaft gears in place.

[0052] In one aspect of preferred methods of assembly the countershaft gears are paired, with members of each pair disposed on opposite sides of the helical gear. Another aspect of preferred methods of assembly includes installing at least 6 pairs of the countershaft gears. Yet another aspect of preferred methods of assembly includes causing each idler counter shaft gear to equally share a distributed load.

[0053] Thus, specific embodiments and applications of a torque balancing gearbox have been disclosed. It should be apparent, however, to those skilled in the art that many more modifications besides those already described are possible without departing from the inventive concepts herein. The inventive subject matter, therefore, is not to be restricted except in the spirit of the appended claims. Moreover, in interpreting both the specification and the claims, all terms should be interpreted in the broadest possible manner consistent with the context. In particular, the terms "comprises" and "comprising" should be interpreted as referring to elements, components, or steps in a non-exclusive manner, indicating that the referenced elements, components, or steps may be present, or utilized, or combined with other elements, components, or steps that are not expressly referenced. Where the specification claims refers to at least one of something selected from the group consisting of A, B, C …. and N, the text should be interpreted as requiring only one element from the group, not A plus N, or B plus N, etc.

CLAIMS

What is claimed is:

1. A gearset having a distributor gear disposed relative to a spur gear in a manner that produces automatic torque balancing.

2. The gearset of claim 1, wherein the distributor gear is an input floatring gear.

3. The gearset of claim 1, wherein the distributor gear has first and second rows of helical teeth oriented to opposite sides of the gear.

4. The gearset of claim 3, wherein the first and second rows of helical teeth are disposed on the external circumference of the distributor gear.

5. The gearset of claim 3, wherein the distributor gear has a set of double helical teeth disposed on the internal circumference of the distributor gear.

6. The gearset of claim 3, further comprising at least 6 planet gears that mate with the first row of helical teeth.

7. The gearset of claim 3, further comprising at least 10 planet gears that mate with the second row of helical teeth.

8. The gearset of claim 6, wherein each planet gear has an end having helical teeth, and a distal end having spur teeth.

9. The gearset of claim 1, wherein the spur gear is an output internal ring gear.

10. The gearset of claim 1, wherein the distributor and spur gears are sized and dimensioned to produce a rotational speed reduction of at least 4:1

11. The gearset of claim 1, wherein the distributor and spur gears are sized and dimensioned to produce a rotational speed reduction of at least 20:1.

12. A rotorcraft having a gearset according to any of claims 3.

13. A gearbox having a number of planet gears disposed about a distributor gear in a manner that yields a power-to-weight ratio of at least twice that of a corresponding double planetary reduction system that uses the same materials and has the same load capability.

14. The gearbox of claim 13, wherein the planet gears are sized to collectively transfer at least 5,000 horsepower.

15. The gearbox of claim 13, wherein the planet gears are sized to collectively transfer up to 50,000 horsepower.

16. The gearbox of claim 13, wherein the number of planet gears is at least 12.

17. The gearbox of claim 13, wherein the number of planet gears is at least 20.

18. The gearbox of claim 16, wherein the planet gears have a helical teeth pattern, and are disposed on opposite sides of the ring gear.

19. The gearbox of claim 16, wherein the planet gears are disposed to transmit power to an output ring gear having internal spur teeth.

20. The gearbox of claim 19, wherein the planet gears, distributor gear, and output ring gear are sized and dimensioned to yield a power-to-weight ratio of at least three times that of the corresponding double planetary reduction system.

AMENDED CLAIMS
received by the International Bureau on 17 August 2009 (17.08.2009)

What is claimed is:

1. A gearset having a distributor gear disposed relative to a spur gear in a manner that produces automatic torque balancing, the distributor gear floating between a plurality of pinion gears and a plurality of planet gears.

2. The gearset of claim 1, wherein each of the plurality of planet gears has helical and spur gear interfaces.

3. The gearset of claim 1, wherein the distributor gear has adjacent first and second rows of helical teeth oriented to opposite sides.

4. The gearset of claim 3, wherein the first and second rows of helical teeth are disposed on the external circumference of the distributor gear.

5. The gearset of claim 3, wherein the distributor gear has a set of double helical teeth disposed on the internal circumference of the distributor gear.

6. The gearset of claim 3, wherein each of the plurality of planet gears number at least 6, and mate with the first row of helical teeth.

7. The gearset of claim 6, wherein each of the plurality of planet gears number at least 10, and mate with the second row of helical teeth.

8. The gearset of claim 6, wherein each of the plurality of planet gears has an end having helical teeth, and a distal end having spur teeth.

9. The gearset of claim 1, wherein the spur gear is an output internal ring gear.

10. The gearset of claim 1, wherein the distributor and spur gears are sized and dimensioned to produce a fixed rotational speed reduction of at least 4:1

11. The gearset of claim 1, wherein the distributor and spur gears are sized and dimensioned to produce a fixed rotational speed reduction of at least 20:1.

12. A rotorcraft having a gearset according to any of claims 3 -11.

13. A gearbox having a number of planet gears disposed about a floating distributor ring gear in a manner that drives power to a floating output ring gear, and yields a power-to-weight ratio of at least twice that of a corresponding double planetary reduction system that uses the same materials and has the same load capability.

14. The gearbox of claim 13, wherein the planet gears are sized to collectively transfer at least 20,000 horsepower.

15. The gearbox of claim 13, wherein the planet gears are sized to collectively transfer up to 50,000 horsepower.

16. The gearbox of claim 13, wherein the number of planet gears is at least 12.

17. The gearbox of claim 13, wherein the number of planet gears is at least 20.

18. The gearbox of claim 16, wherein each of the planet gears has a helical teeth pattern, and are disposed adjacently on opposite sides of the ring gear.

19. The gearbox of claim 18, wherein each of the planet gears are disposed to transmit power to an output ring gear having internal spur teeth.

20. The gearbox of claim 19, wherein the planet gears, distributor ring gear, and output ring gear are sized and dimensioned to yield a power-to-weight ratio of at least three times that of the corresponding double planetary reduction system.

Figure 1 (Prior Art)

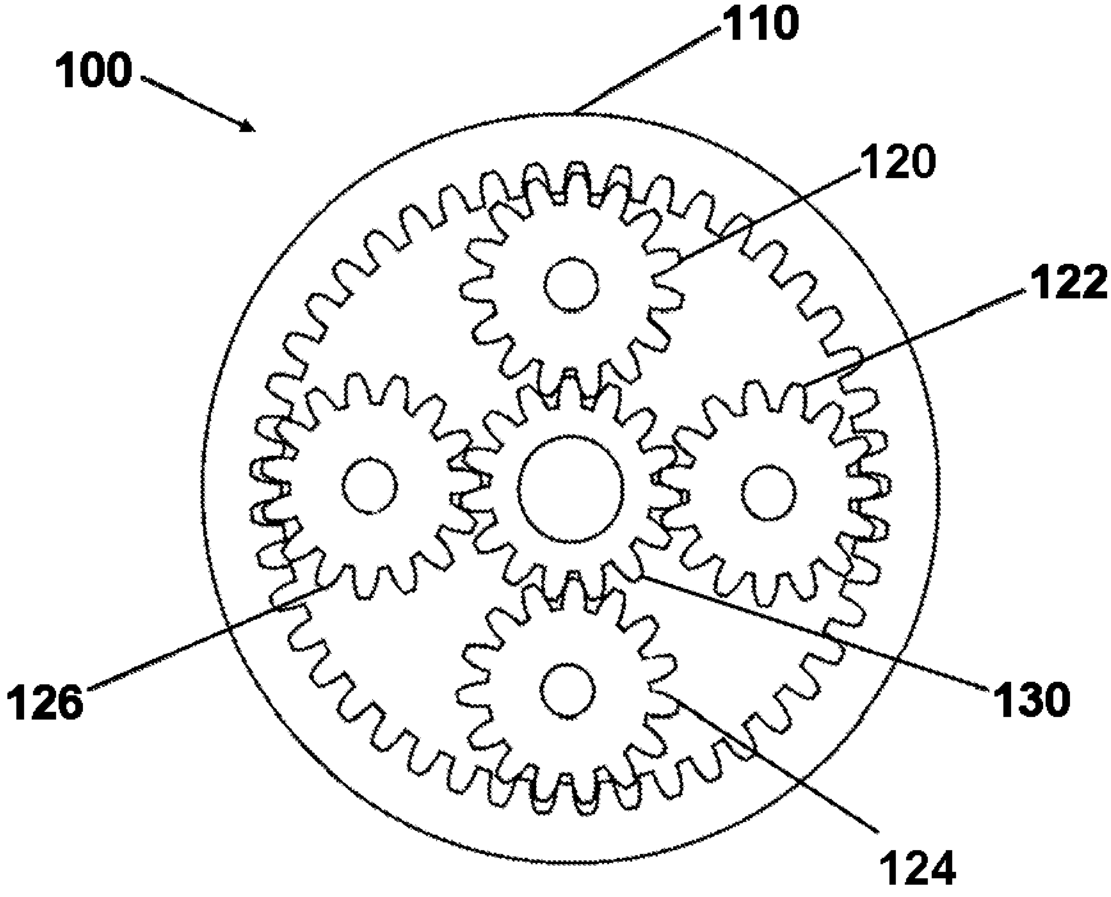


Figure 2 (Prior Art)

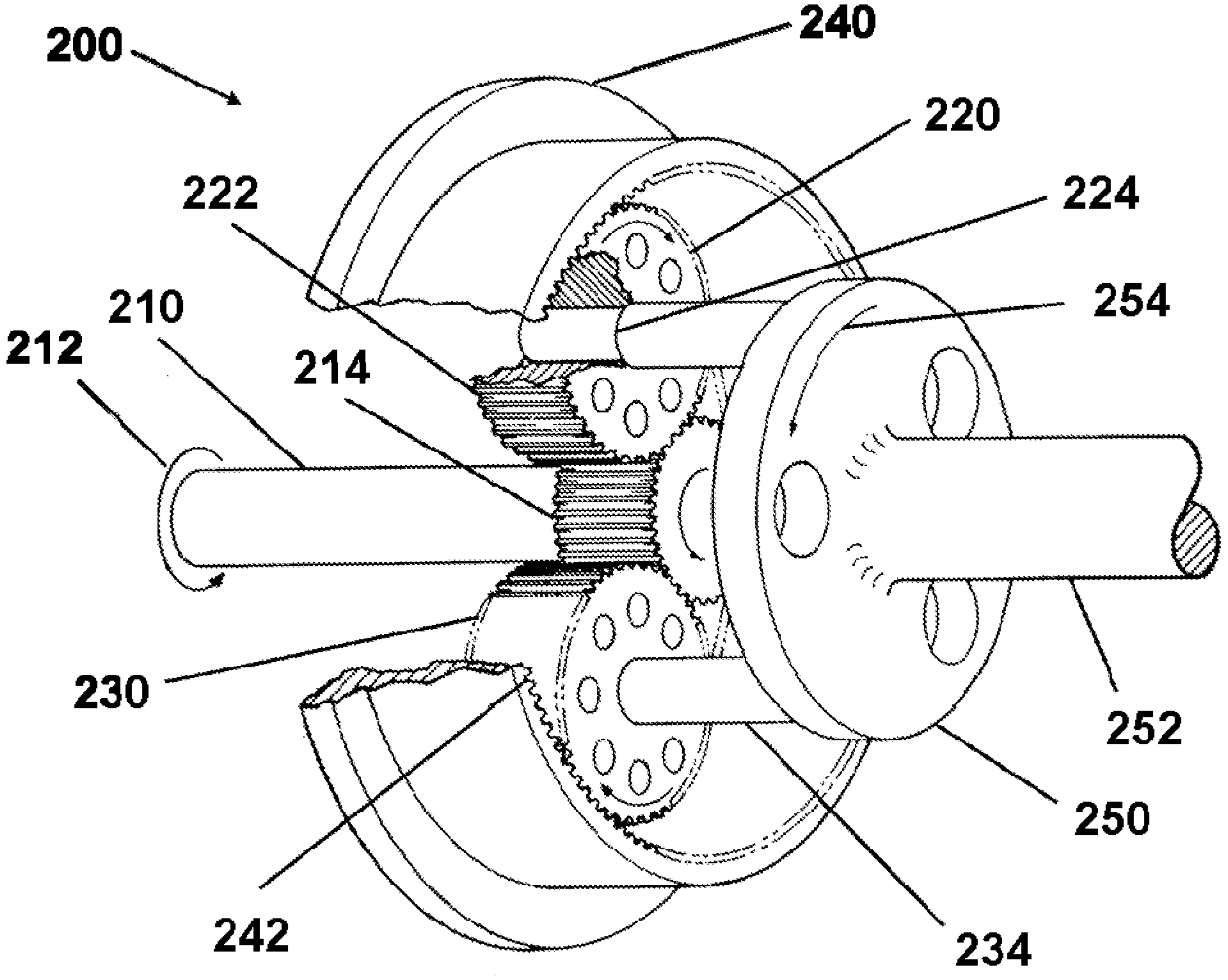
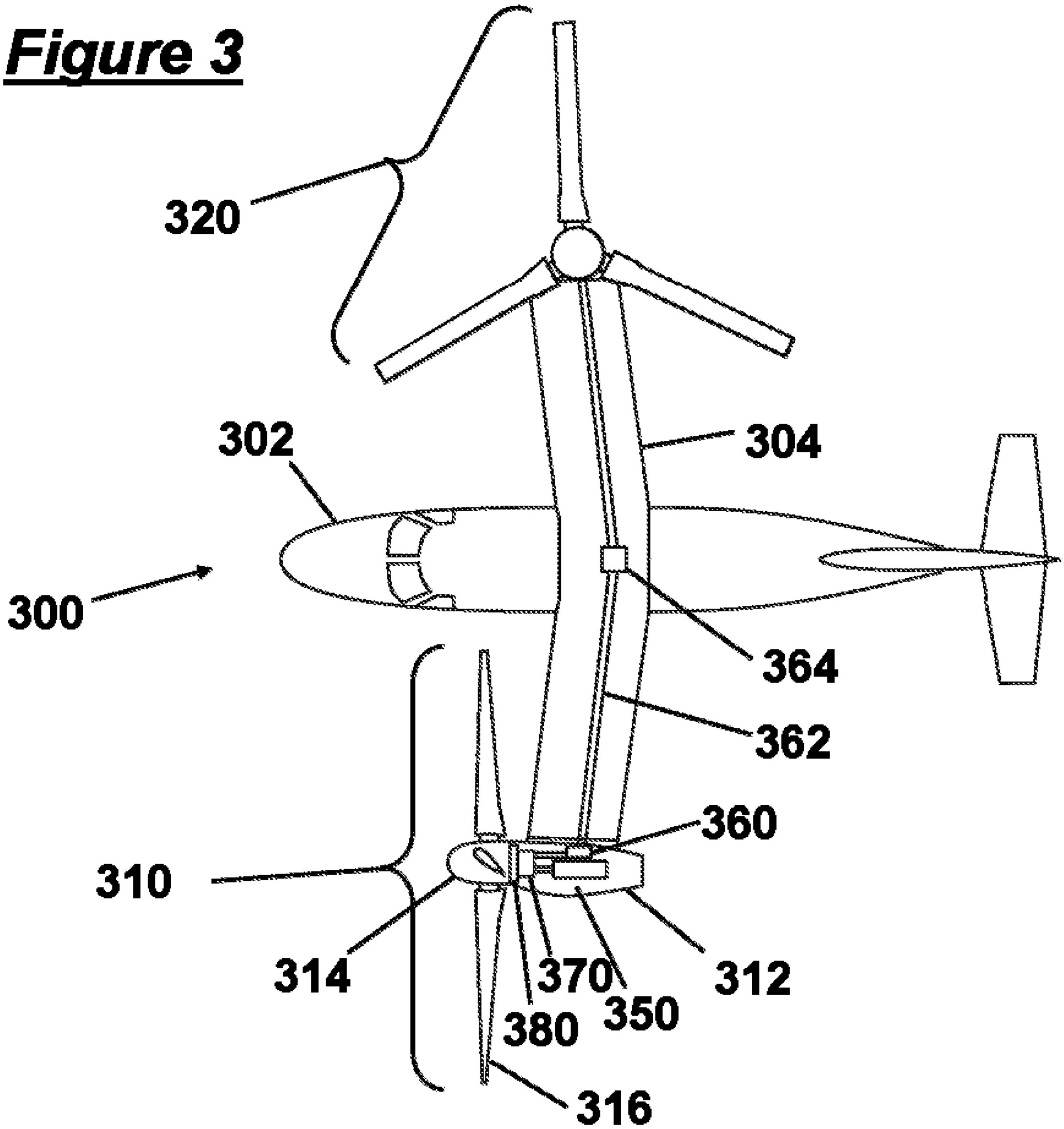


Figure 3



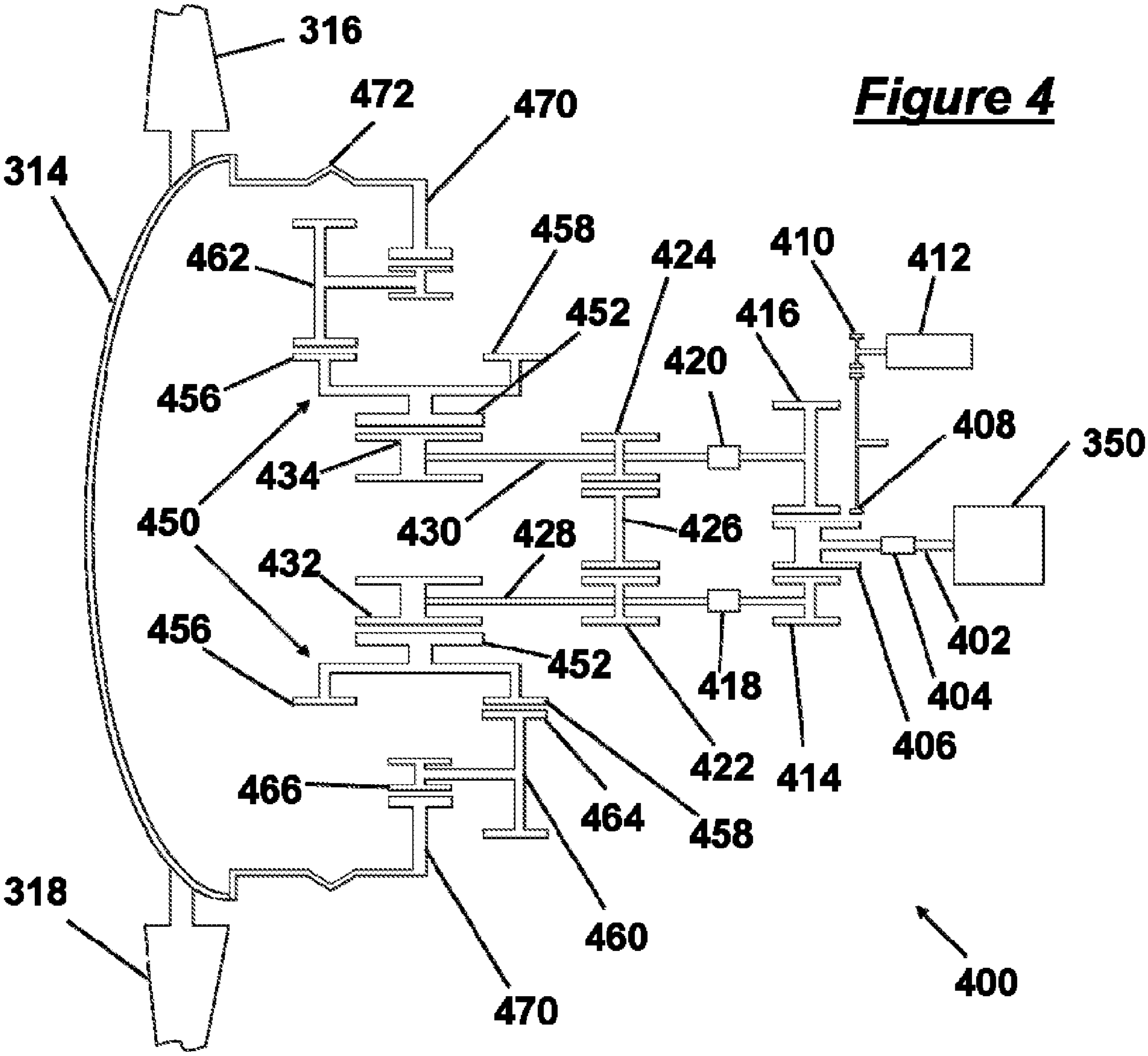

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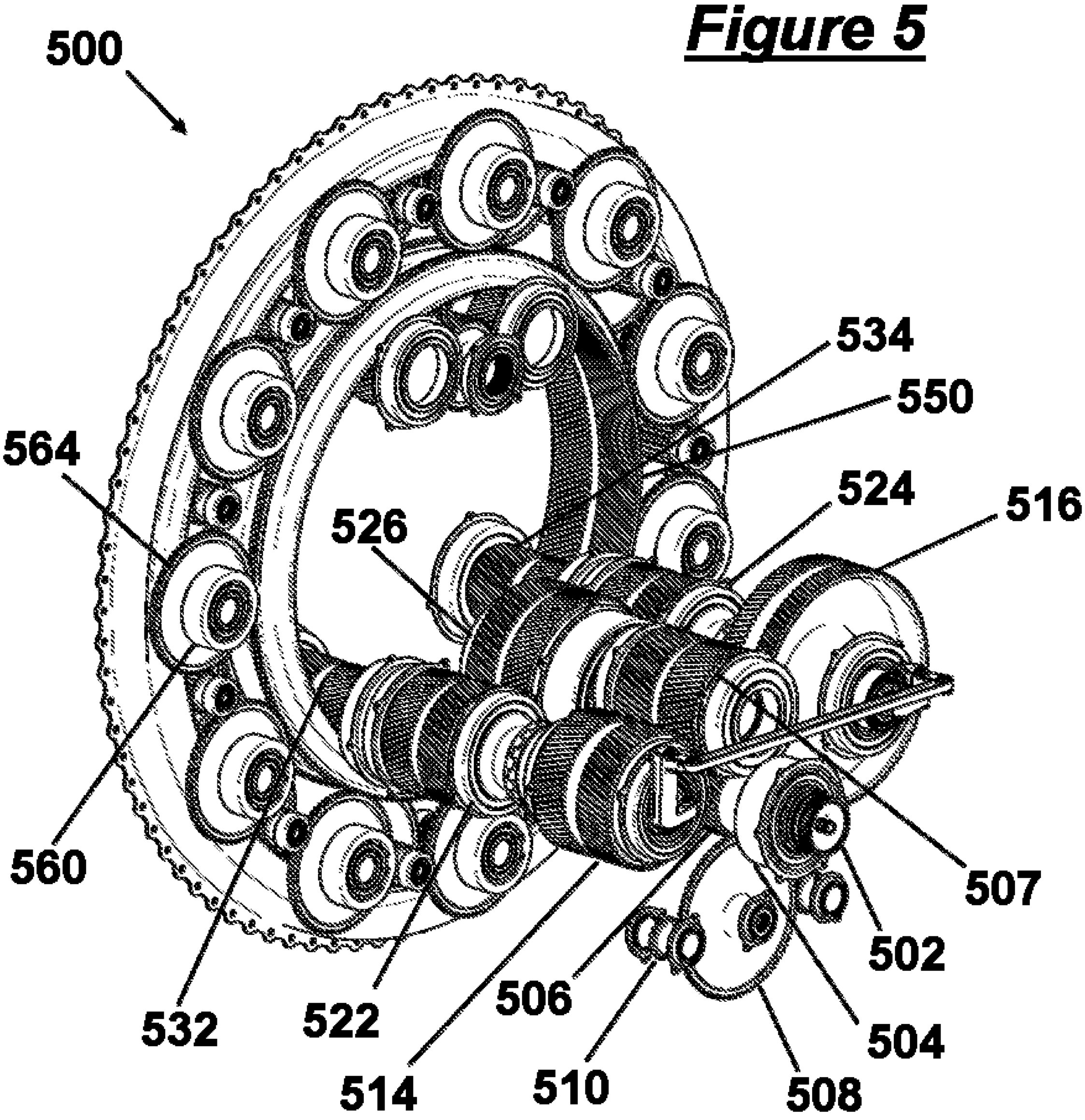
Figure 4

Figure 5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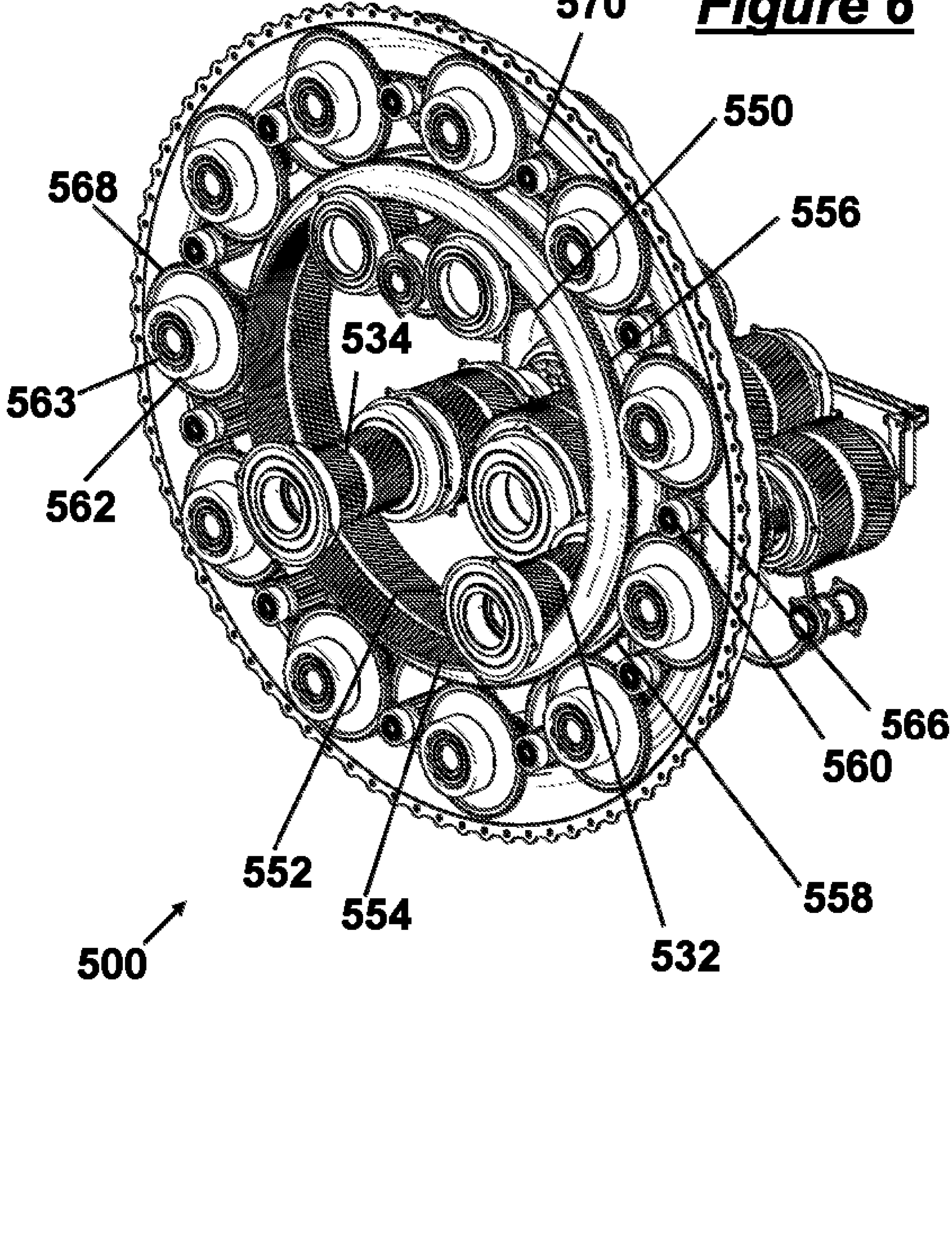


Figure 6

Figure 7

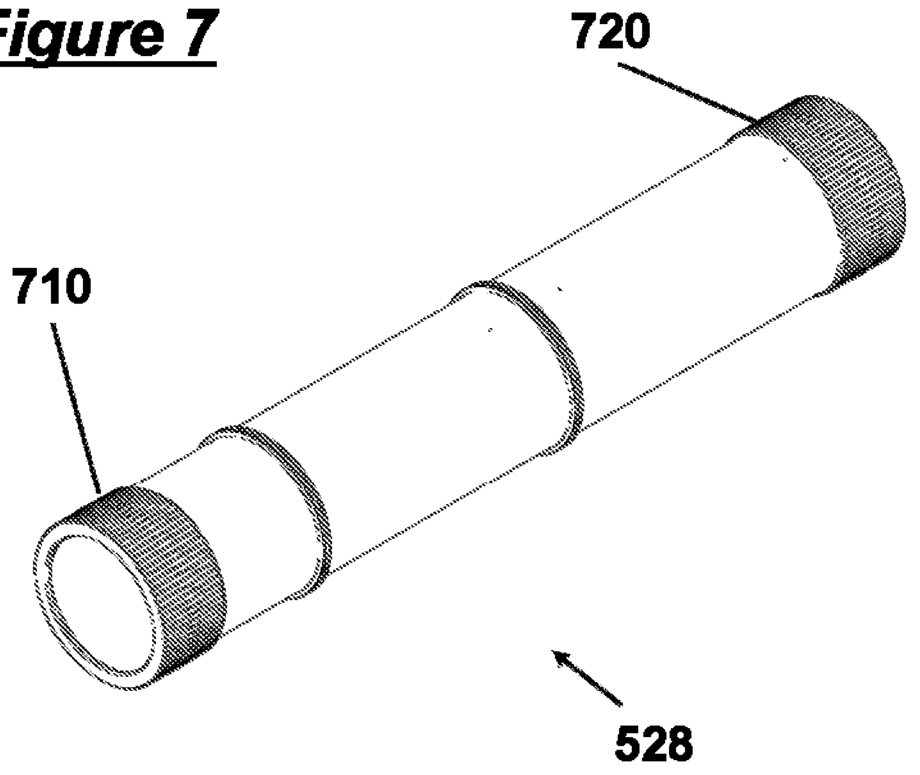
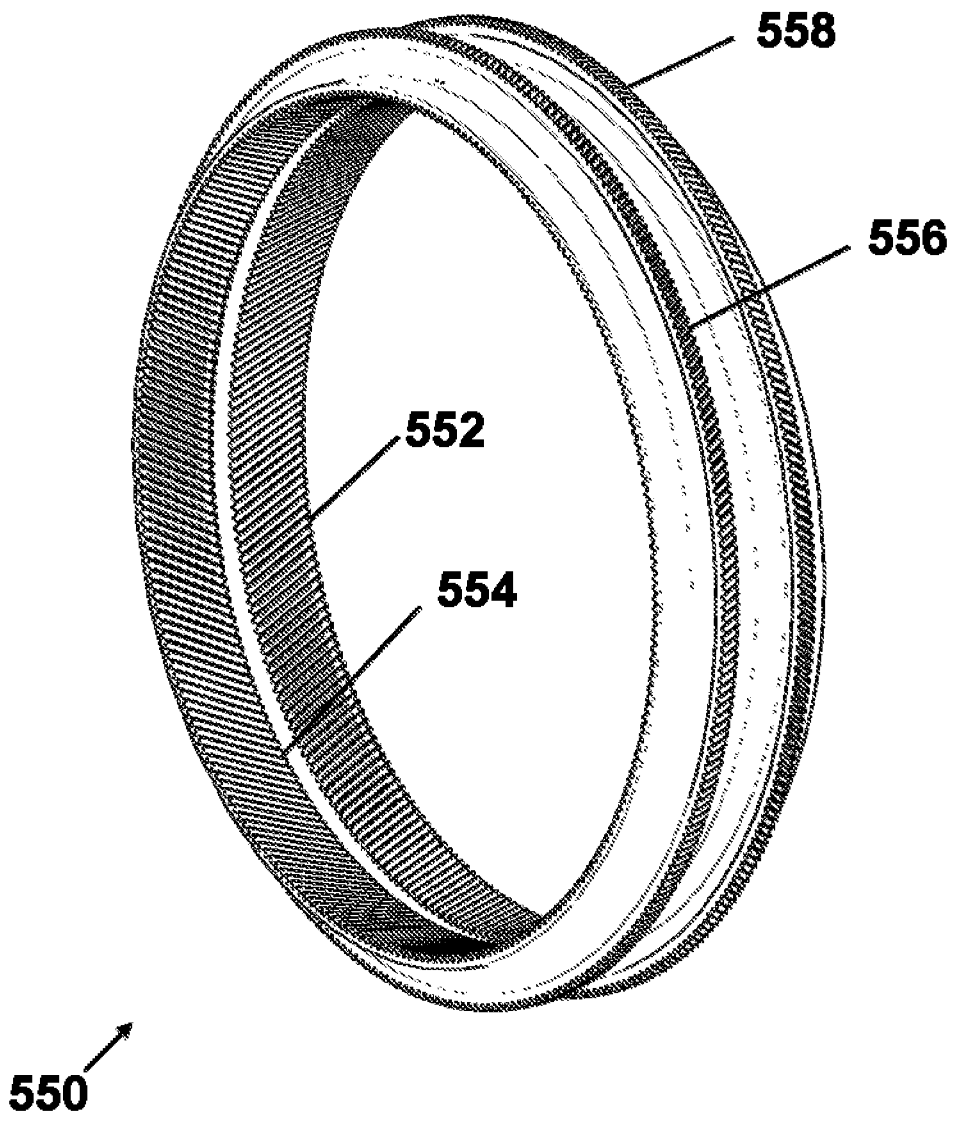


Figure 8



INTERNATIONAL SEARCH REPORT

International application No.
PCT/US 09/36372

A. CLASSIFICATION OF SUBJECT MATTER

IPC(8) - F16H 3/08 (2009.01); F16H 1/32 (2009.01)

USPC - 74/363; 475/162

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
USPC: 74/363; 475/162

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
USPC: 74/363, 337, 362, 366, 369, 640, 665; 475/162, 11, 47, 53, 176 ? text limited see below

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
Pubwest (DB=PGPB,USPT,USOC,EPAB,JPAB); Google Scholar
Search terms: gear, planet, spur, helical, teeth, floating, distributing

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X -- Y	US 2,467,226 A (Place) 12 April 1949 (12.04.1949) figures 1, 4	1-2, 9-11, 13, 16-17, 19-20 -- 3-8, 12, 14-15, 18
Y	US 2,909,944 A (Bouvier) 27 October 1959 (27.10.1959) figure 1	4
Y	US 6,615,683 B1 (Voigt) 09 September 2003 (09.09.2003) col 3, ln 20-24	3-8, 12, 18
Y	JP 07105640 A (Kazuo) 21 April 1995 (21.04.1995) Abstract	8
Y	US 3,430,522 A (Hawkins et al.) 04 March 1969 (04.03.1969) col 1, ln 33	14
Y	US 6,024,549 A (Lee) 15 February 2000 (15.02.2000) col 1 , ln 47	15
Y	US 2006/0000300 A1 (Vialle) 05 January 2006 (05.01.2006) Abstract	12
A	US 4,512,213 A (NEWTON) 23 April 1985 (23.04.1985), entire document	1-20
A	US 4,321,842 A (STROMOTICH) 30 March 1982 (30.03.1982), entire document	1-20
A	COY et al., Helicopter Transmission Research at NASA Lewis Research Center, NASA Technical Memorandum 100962 [online], November 1988 (11.1988) [retrieved on 08 June 2009 (08.06.2009)], retrieved from the internet <URL: http://www.dtic.mil/cgi-bin/GetTRDoc?AD=ADA242220&Location=U2&doc=GetTRDoc.pdf>	1-20

☐ Further documents are listed in the continuation of Box C. ☐

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier application or patent but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
04 June 2009 (04.06.2009)	17 JUN 2009

Name and mailing address of the ISA/US	Authorized officer:
Mail Stop PCT, Attn: ISA/US, Commissioner for Patents P.O. Box 1450, Alexandria, Virginia 22313-1450 Facsimile No. 571-273-3201	Lee W. Young PCT Helpdesk: 571-272-4300 PCT OSP: 571-272-7774

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US 09/36372

Box No. II Observations where certain claims were found unsearchable (Continuation of item 2 of first sheet)

This international search report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. ☐ Claims Nos.:
because they relate to subject matter not required to be searched by this Authority, namely:

2. ☐ Claims Nos.:
because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:

3. ☐ Claims Nos.:
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

Box No. III Observations where unity of invention is lacking (Continuation of item 3 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:
This application contains the following inventions or groups of inventions which are not so linked as to form a single general inventive concept under PCT Rule 13.1. In order for all inventions to be examined, the appropriate additional examination fees must be paid.

Group I, claims 1-12, drawn to a gearset.
Group II, claims 13-20, drawn to a gearbox.

The inventions listed as Groups I - II do not relate to a single general inventive concept under PCT Rule 13.1 because, under PCT Rule 13.2, they lack the same or corresponding special technical features for the following reasons: the special technical feature of the Group I claims is a gearset having a distributor gear disposed relative to a spur gear in a manner that produces automatic torque balancing; the special technical feature of the group II claims is a gearbox having a number of planet gears disposed about a distributor gear in a manner that yields a power-to-weight ratio of at least twice that of a corresponding double planetary reduction system; and the groups do not share a special technical feature.

1. ☒ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.

2. ☐ As all searchable claims could be searched without effort justifying additional fees, this Authority did not invite payment of additional fees.

3. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:

4. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

Remark on Protest

☐ The additional search fees were accompanied by the applicant's protest and, where applicable, the payment of a protest fee.

☐ The additional search fees were accompanied by the applicant's protest but the applicable protest fee was not paid within the time limit specified in the invitation.

☒ No protest accompanied the payment of additional search fees.