

MOUNTING OF A PLANETARY GEAR ASSEMBLY IN A CASING

5 This invention relates to a mounting of a planetary gear assembly in a casing.

Planetary gear assemblies have a wide scope of uses as retarders or accelerators.

10 Planetary gear assemblies are able to be used in various manners. In most cases the outer, inner-toothed gear is fixed in a casing. In such cases the outer, inner-toothed gear is tightly fixed in the casing by screws parallel with the axis of the gear assembly.

15 Recently planetary gear assemblies have increasingly been adopted as reduction devices, eg. for electric washers, instead of belt-pulley reduction devices.

20 Japanese Patent Publication No. 60-27320 (published on June 28, 1985) disclosed a single-tub electric washer having a planetary gear assembly as the reduction device.

Because a planetary gear assembly can be used as an integrated unit, it enables simplification of the manufacturing process of an electric washer.

In the prior art, axially-directed screws tightly fix an outer, inner-toothed gear in a casing. Because of the screw-fixing, the outer, inner-toothed gear is tightly coupled with the casing. Four screws  
5 or eight screws are used in the fixing.

However the screw-fixing is time consuming. A worker must align the holes bored through the outer, inner-toothed gear with the screw holes bored in the casing, insert screws into the  
10 aligned holes and turn the screws with a screwdriver.

To simplify this process, a more convenient fixing was devised. Several protrusions were formed in the axial direction on a casing, the protrusions corresponding to the holes of the outer,  
15 inner-toothed gear. By inserting the protrusions of the casing into the holes of the outer, inner-toothed gear and pushing the outer, inner-toothed gear by a ring-shaped presser plate, the outer, inner-toothed gear is tightly fixed in the casing.

20 There is no clearance between the holes of the outer, inner-toothed gear and the protrusions of the casing. Thus the outer, inner-toothed gear is firmly fixed without backlash. This fixing prevents relative movement between the casing and the outer,

inner-toothed gear.

However this fixing is still time consuming, because a plurality of holes of the outer, inner-toothed gear must be fitted on the corresponding protrusions.

In general, reduction gears generate noise from the engagement of gear teeth. Apart from in electric washers, reduction gears employed in any machines generate noise more or less.

Of course other parts, apart from the reduction gear, also generate noise, because a machine consists of many movable parts.

Planetary gear assemblies have been well known for a long time. However they have hardly been used in practice as reduction devices because of practical shortcomings. Because a planetary gear assembly has a high symmetry, its transmission efficiency ought to be high. But that is not always the case.

Unlike other gear-train systems, the freedom of gear engagement is very low in a planetary gear assembly, because of the many engagement points between the gears. Small dimensional inaccuracies in parts frequently generate reaction forces which impede

relative rotation of the gears. This fact has reduced the merit of planetary gear assemblies.

5           Because this difficulty can be resolved by raising the precision of gear-shaping, good planetary gear assemblies have been manufactured by assembling gears which have been finished accurately without dimensional error.

10           Accordingly the conventional planetary gear assemblies must be expensive. This fact may inhibit the adoption of planetary gear assemblies as reduction devices.

Common sense has taught us that planetary gear assemblies with low precision were useless.

15           Thus it was thought that an outer, inner-toothed gear should be tightly fixed on a casing with the least dimensional error. For that reason fixing by screws, protrusions and holes has been adopted for a long time for mounting a planetary gear assembly in a casing.

20           This inventor has invented a planetary gear assembly which may be in conflict with the prevailing common sense (Japanese Patent Application No. 56-193113, laid open Japanese Patent Application No. 58-94656, laid open on June 4, 1983).

The planetary gear assembly disclosed in JP-A-5894656 has a plurality of planetary gears which consists of a ring-shaped gear and two side discs having a diameter larger than the diameter of the addendum circle of the gear. The ring-shaped gear is sandwiched between the two side discs.

In correspondence with the structure of the planetary gears, the outer, inner-toothed gear consists of a gear part and two side parts having an inside diameter which is larger than the diameter of the dedendum circle of the gear part.

Because the planetary side discs roll on the inner-cylindrical surfaces of the side parts of the outer, inner-toothed gear, radial displacement of the planetary gears is effectively suppressed, which ensures smooth rotation of the gears.

The inventor thinks that the planetary gear assembly disclosed in JP-A-5894656 is an unprecedented invention which apparently contradicts the common sense of the field of technology.

The peripheral velocity of a gear is defined as the tangential speed on the pitch circle of the gear.

A complex gear which has one or two side

friction discs has been used for a long time to prevent excessive engagement of the teeth.

5           The diameters of the side friction discs were equal to that of the pitch circles of the gears without exception. If the diameters of the side discs were different from the pitch circles, the peripheral velocities of the side discs were different. To equalize the peripheral velocities of the two discs, the diameters of the side discs must be equal to that  
10           of the pitch circles.

          The inventor has departed from the common sense approach that the diameters of the side discs must be equal to the pitch circles in complex gears.

          The planetary gear invented by this  
15           inventor and disclosed in JP-A-5894656 has side discs with diameters larger than the diameter of the addendum circle of the gear which, in turn, is larger than the pitch circle of the gear. The planetary gear assembly having the novel planetary gear can rotate  
20           despite the difference of the peripheral velocities of the rolling discs. The reason why the gears can rotate will be explained now.

          Of course the peripheral velocities of the rolling side discs are different, but the difference

is very small. This is one reason.

The effect of small differences between peripheral velocities can be cancelled by widening clearances between the discs and the gear parts, because the wider clearance allows slippage between the side discs and the gear parts. This is another reason.

This inventor became aware of the two facts. First the difference  $\Delta v$  of the peripheral velocities of two side discs rolling with each other is given by

$$\Delta v = \delta \Omega \quad (1)$$

where  $\delta$  is the deviation of the disc surface from the pitch circle and  $\Omega$  is the angular velocity of the gear.

The difference  $\Delta v$  may be intuitively estimated as  $2 \delta \Omega$ , because both of the rolling discs and inner cylindrical parts deviate by  $\delta$  from the pitch circles of the respective gears.

But this is wrong. A brief calculation shows us that the difference is only  $\delta \Omega$ .

Furthermore if wide clearances are formed

between the outer surfaces of the discs and the inner cylindrical surfaces of the side parts, the planetary discs can easily slip relative to the side parts. The slippage counteracts the difference between the peripheral velocities.

Besides, the other wide clearances between the planetary discs and the planetary ring-shaped gear make the planetary discs slip relative to the planetary gear.

The concept of wide clearances sharply contradicts with the conventional common sense of planetary gear assembly design.

By providing wide clearances between gears, discs and inner cylindrical parts, the demand for precise finishing is removed. The planetary gear assembly can be constructed with cheap parts shaped with low precision. The low precision can be compensated for by the wide clearances among the parts. Therefore the invention disclosed in JP-A-5894656 enables manufacture of planetary gear assemblies at low cost.

By the invention disclosed in JP-A-5894656, planetary gear assemblies are beginning to be adopted for various purposes. That invention is

one of the highly significant inventions in the history of planetary gear assemblies.

5 There are various forms of dimensional error in a planetary gear assembly. One such error is radial deviation of the sun gear. Another such error is radial deviation of the carrier. These errors are caused by errors in radial location or alignment of planetary shafts.

10 The criterion to determine the precise positions of planetary shafts in a carrier is called "division precision". High division precision would ensure an excellent planetary gear assembly with low noise and high transmission efficiency.

15 However it is very difficult and expensive to increase the division precision.

Furthermore an input shaft and an output shaft should be fitted to the sun gear and the carrier. The shafts are supported in the casing by bearings. In many cases the centre of the bearings is displaced from the centre of the casing.

20

Therefore there are various dimensional errors in finishing parts, determining positions and assembling parts.

This inventor thinks the conventional

fixed mounting of an outer, inner-toothed gear in a casing disturbs the smooth rotation of gears, increases the tooth-edge interference and generates large noise, because dimensional errors cannot be cancelled in the case of a fixed mounting.

One purpose of this invention is to provide a mounting of a planetary gear assembly in a casing with low noise generation.

Another purpose of the invention is to provide a mounting of a planetary gear assembly in a casing which is easy to handle.

Therefore this invention proposes a relaxation in the accuracy of the mounting of an outer, inner-toothed ring gear in a casing instead of the conventional fixed mounting.

The relatively loose mounting in which this invention is embodied is defined by permitting radial displacements and preventing relative rotation between the outer, inner-toothed gear and the casing.

The relatively loose mounting requires neither screws, pins nor screwholes formed in an axial direction.

According to this invention there is provided a mounting of a planetary gear assembly

having a sun gear, a plurality of planetary gears and an outer, inner-toothed ring gear positioned in a casing wherein a plurality of arcuate coupling parts projecting from an outer cylindrical surface of the ring gear are loosely fitted within complementary arcuate recesses in an inner surface of the casing, the arrangement being such that a clearance exists between the arcuate coupling parts and the recesses to permit radial displacement of the ring gear within the casing, a major part of the outer cylindrical surface of the ring gear is spaced from the inner surface of the casing by a relatively wide space to permit elastic deformation of the ring gear, and the plane in which rotational forces are transmitted between the planetary gears and the ring gear is displaced axially from the plane in which rotational forces are transmitted between the arcuate coupling parts of the ring gear and the casing.

The characteristic of this invention is the relaxation mounting in place of the fixed mounting.

In the drawings:-

Fig. 1 is a bottom view of a mounting of a planetary gear assembly in a casing in which this

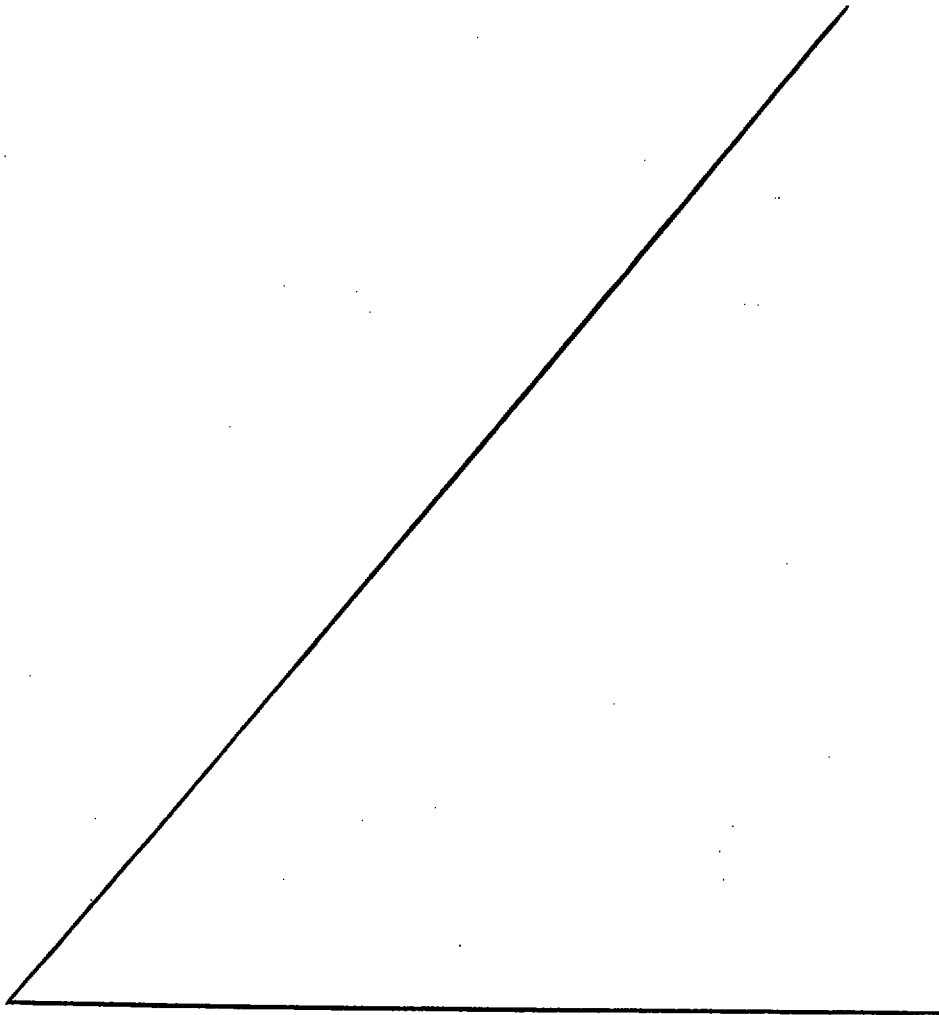
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invention is embodied.

Fig. 2 is a sectional view taken along the line II-II in Fig. 1.

Fig. 3 is a plan view of the same  
5 mounting.

Fig. 4 is a bottom view of the casing



alone.

Fig. 5 is a sectional view taken along the line V-V in Fig. 4.

5 Fig. 6 is a plan view of the planetary gear assembly alone.

Fig. 7 is a side view of the outer, inner-toothed gear alone.

10 Fig. 8 is a sectional view of a mounting of another planetary gear assembly in a casing in which this invention is embodied.

Fig. 9 is a sectional view of a mounting of yet another planetary gear assembly in a casing in which this invention is embodied.

15 Fig. 1 is a bottom view of a mounting of a planetary gear assembly (A) in a casing (B). In an electric washer the casing (B) and the gear assembly (A) are installed by positioning the assembly plane parallel with the horizontal plane.

20 The inner structure of the planetary gear assembly (A) is arbitrary, because this invention does not relate to the planetary gear assembly itself but to the mounting of a gear assembly in a casing.

Fig. 1 shows an example whose planetary gears have two side discs which are larger in diameter

than the addendum circle of the respective gear. This corresponds to the Japanese laid open Patent Application No. 58-94656.

5 A sun gear (1) is positioned in the centre of the planetary gear assembly. Four planetary gears (2) are disposed symmetrically around the sun gear (1).

The top view of the planetary gear assembly alone is shown in Fig. 6.

10 The planetary gears (2) are engaged with the sun gear (1). However the tooth edges of the planetary gears (2) do not appear in Fig. 1 or Fig. 6, because they are concealed by the planetary side discs (6) and (7) (see Fig. 2).

15 An outer, inner-toothed ring gear (3) is disposed around the planetary gears (2) which are engaged with it. The ring (3) serves as a housing of the planetary gear assembly.

20 A carrier (4) consists of a main carrier disc (4a) and a sub-carrier disc (4b). Four planetary shafts (5) are supported at either end at respective points on the inner surfaces of the carrier discs (4a) and (4b). The shafts (5) are positioned symmetrically with respect to the centre.

The planetary gears (2) are supported for rotation by the planetary shafts (5) which pass through them.

5 Each planetary gear (2) consists of three parts - the two planetary side discs (6) and (7), which are similar, and a planetary gear ring (8) which is sandwiched between the discs (6) and (7).

10 Each planetary disc (6) or (7) has a large disc part (23) and a small column part (21). The outside diameter of the large disc part (23) is larger than the diameter of the addendum circle of the planetary gear ring (8). The small column part (21) contacts the inner surface of the planetary gear ring (8) and supports it.

15 The main carrier disc (4a) and the sub-carrier disc (4b) are coupled together at four symmetrically-located points which are angularly displaced from the adjacent planetary shafts (5) by 45 degrees.

20 The main carrier disc (4a) has four protrusions (10) and four spigots (11) on the inner surface.

The sub-carrier disc (4b) has four protuberances (12) and four holes (13) correspondingly.

In order to couple the carrier discs (4a) and (4b), the spigots (11) of the main carrier disc (4a) are inserted into the holes (13) of the sub-carrier disc (4b). Then the outer ends of the spigots (11) are pressed and widened. The enlarged outer ends (28) urge the sub-carrier disc (4b) against the main carrier disc (4a).

In this example, the carrier (4) is made from aluminium die-casting. The carrier may be made from zinc, steel or plastics.

In the case of plastic carrier discs, two carrier discs are coupled only by fitting the spigots (11) into the respective holes (13). The elasticity of the plastic parts maintains the coupled state. If the outer ends of the spigots are fixed by an adhesive, the coupling is reinforced. The outer ends of the spigots may be welded by a supersonic welding machine.

In the case of zinc carrier discs, the carrier discs are easily molded from a zinc melt by a metallic mold.

In the case of steel carrier discs (eg. Chrome-Molybdenum-steel), simple discs are cut. Then the discs (4a) and (4b) are connected by rivets.

The sun gear (1) and the planetary gears (2) are made from sintered alloy, steel, plastics or zinc.

5 The inner surface of the ring gear (3) is formed as a gear part (25) between two cylindrical surface parts (26) and (27). The diameter of the cylindrical surface parts (26) and (27) is larger than the diameter of the dedendum circle of the gear part (25).

10 The gear part (25) is engaged with the planetary gear rings (8). The large disc parts (23) of the planetary discs (6) and (7) are in rolling engagement with the cylindrical surface parts (26) and (27) of the ring gear (3).

15 The large disc parts (23) and the cylindrical surface parts (26) and (27) are displaced from the pitch circles of their respective gears. Thus the peripheral velocities are different. The difference between the peripheral velocities is  $\delta \Omega$ .

20  $\delta$  is the displacement of the larger disc parts (23) or the cylindrical surface parts (26) from the pitch circles of their respective gears.  $\Omega$  is the relative angular velocity of the planetary gear (2) with regard to carrier (4).

However there are clearances between the planetary gear ring (8) and the planetary discs (6) and (7). Also there are clearances between the planetary discs (6) and (7) and the cylindrical surface parts (26) and (27). Therefore these rolling surfaces are not always in contact with each other. Free rotation of the parts is allowed when they are not in contact with each other.

Accordingly despite the difference between the peripheral velocities, the surfaces of the planetary discs (6), (7) and the cylindrical surface parts (26), (27) do not strongly rub with each other. The clearances cancel the difference between the peripheral velocities. The cancellation is complete except in the case of high speed rotation.

The ring gear (3) is conveniently a plastic molding. This is possible, because the cylindrical surface parts (26) are larger than the gear part (25).

Otherwise the ring gear (3) can be made from aluminium die-casting. But in the case of aluminium, the shape of the ring gear (3) must be designed appropriately for aluminium die-casting.

A major part of the ring gear (3) is a

smooth cylindrical surface. Fig. 7 shows the side view of the ring gear (3). A smooth outer cylindrical surface (18) occupies most of the external surface. But four arcuate coupling parts (19) are formed on one side around the ring gear (3).

The arcuate coupling parts (19) are not formed on both sides, nor do they span the width between both sides. The arcuate coupling parts (19) are asymmetric with regard to the central plane of the gears (1), (2) and (3).

The ring gear (3) is loosely coupled with the casing (B) by the arcuate coupling parts (19).

The number of arcuate coupling parts (19) is not necessarily four. Two, three or more than five arcuate coupling parts (19) may be provided.

In this example the arcuate coupling part (19) comprises an external arcuate rib (30), two arcuate grooves (31) and a circular cavity (33). This rib structure is designed to economise on the plastic material without reducing the strength of the arcuate coupling part (19). However the rib structure may be changed arbitrarily.

With regard to the casing (B), the top view is shown by Fig. 3 and the bottom view is shown

- by Fig. 4. Fig. 5 shows the sectional view.

In this example the casing (B) is made from aluminium in order to increase the heat radiation and the mechanical strength. Otherwise the casing (B) can be made from plastics, cast iron, aluminium alloy or other metal.

The casing (B) consists of a large drum part (40), a conical part (41) and a small cylindrical part (42). The drum part (40) accommodates the planetary gear assembly (A). The cylindrical part (42) has an output shaft hole (43) through which an output shaft projects outwardly.

Most of the inner surface of the drum part (40) is formed as smooth arcuate surface portions (44). Four trapezoidal coupling projections (45) are formed on the inner surface of the drum part (40) so that they extend in the axial direction.

The coupling projections (45) of the casing (B) define complementary arcuate recesses (44) to the arcuate coupling parts (19) of the ring gear (3).

When the planetary gear assembly (A) is mounted on the casing (B), the arcuate coupling parts (19) are inserted into the arcuate recesses (44) of

the drum part (40).

The smooth outer cylindrical surface (18) contacts the coupling projections (45).

5 A circular rib (47) and radial ribs (48) are formed on the external surface of the conical part (41) for reinforcing the casing (B).

10 A flange (46) is formed at the forefront of the drum part (40). A cover plate (not shown) is to be fitted on the flange and to be fixed by bolts (not shown) inserted through fixing holes (50).

15 There are clearances (g) between the arcuate coupling parts (19) of the ring gear (3) and the arcuate recesses (44) of the casing (B) when the planetary gear assembly (A) is mounted in the casing (B).

20 For example, the outer diameter of the arcuate coupling parts (19) is 75 mm, and the inner diameter of the arcuate recesses (44) is 76 mm. In the example, the clearance (g) is 0.5 mm in radius or 1 mm in diameter. The clearance (g) has an arbitrary value between 0 and 1 mm.

In general, an appropriate value of the clearance (g) is 0.5 mm to 3 mm.

Furthermore there are clearances (e)

between the coupling projections (45) and sloping edges (51) of the arcuate coupling parts.

Also there are wide spaces (F) between the smooth outer cylindrical surface (18) of the ring gear (3) and the arcuate recesses (44) of the casing (B).  
5 The width is denoted by (f).

The functions of the mounting in which this invention is embodied will now be explained.

An input shaft (not shown in the figures) is fitted in the sun shaft hole (14) (see Fig. 2). An  
10 output shaft (not shown in the figures) is fitted in the carrier shaft hole (15). Splines or serrations are formed on the shafts and the shaft holes (14 and 15) to couple the shafts with the parts of the  
15 planetary gear assembly (A) in which the respective shaft holes (14) and (15) are formed.

The cover plate (not shown) is fitted on the flange (46) and is fixed by bolts to the casing (B) when the planetary gear assembly (A) is mounted in  
20 the casing (B).

When the input shaft rotates, the output shaft also rotates at a reduced speed, because the planetary gear assembly (A) acts as a reduction device.

However the ring gear (3) can be displaced radially, because it is not fixed but is loosely mounted in the casing (B).

" $g_m$ " denotes the maximum value of (g), so  
5 that the clearance (g) between the casing and the ring gear (3) can take an arbitrary value between 0 and  $g_m$ .

The output shaft and the input shaft may be eccentric, one relative to the other, to a small extent, that is, the centres of the shafts may deviate  
10 from the centres of the casing (B) and the planetary gear assembly (A).

Further there may be an error in the relative angularly-spaced locations of the planetary shafts; that is to say the angular spacing of at least  
15 some of the adjacent pairs of the planetary shafts may be different rather being the same as it should be. Such dimensional errors would induce resistance to rotation of the gears, if the ring gear was tightly fixed and very narrow clearances were given at the  
20 contact points of the parts.

However the wide clearances allow radial displacement of the gears (1), (2) and (3). Thus such radial displacements of the gears lead to equalisation

of the forces acting on the contact points. The dynamical balance of the forces reduces resistance forces caused by dimensional errors.

5 Therefore even if the planetary gear assembly itself is finished with low precision, the gears (1), (2), (3) and the carrier (4) are able to rotate without resistance. Smooth rotation of gears lead to high transmission efficiency.

10 Furthermore this relatively loose mounting allows random radial displacements of the gears (1), (2), (3) and the carrier (4). The radial displacements equalize all forces acting on arbitrary points and reduce noise generation. Low noise is the most important advantage of this relaxation mounting.

15 Vibrations with various frequencies and amplitudes occur at the contacting points of teeth. However high frequency vibrations are not transmitted to the casing (B), because the ring gear (3) is tightly mounted in the casing (B).

20 Regarding the low frequency vibrations, vibrations having amplitudes which are less than the clearance (g) are not transmitted to the casing (B). Such vibrations with small amplitudes are absorbed by the clearance (g).

Most of the external part (smooth outer cylindrical surface (18)) of the ring gear (3) is spaced from the casing (B) by the width (f). The narrow side parts (arcuate coupling parts (19)) of the ring gear (3) contact the casing (B). Hence the ring gear (3) can freely deform elastically. There are two modes of deformation.

Elastic deformation of the ring gear (3) can be radial and angular. Radial deformation signifies repeated dilation and shrinkage of the smooth outer cylindrical surface (18). Because of the clearances (g) and (e), the ring gear deforms towards an ellipsoidal shape or towards a square shape.

Another mode of deformation is twisting of the cylindrical surface (18) about its axis, namely angular deformation. This mode of deformation is caused by the arcuate coupling parts (19) being formed only on one side of the ring gear.

The centre of the rotational forces acting on the ring gear (3) from the planetary gears (2) coincides with the centre line of the outer gear part (25). The line X - X denotes the centre of the rotation forces in Fig. 2.

However the rotational forces are trans-

- mitted to the casing (B) by the arcuate coupling parts (19). Hence the centre of the forces in the casing (B) coincides with the working line Y - Y in Fig. 2.

5 The line X - X or the plane X differs from the line Y - Y or the plane Y. The torque acting on the plane X is balanced by the torque acting on the plane Y. However a twisting torque exists between the plane X and the plane Y. The twisting torque twists the outer cylindrical surface (18) of the ring gear  
10 (3).

If the gears rotate clockwise and counter-clockwise in turn, like reduction gears in electric washers, such twisting deformation alleviates the impulses occurring at the moment of reversion.  
15 Because the impulses are reduced, noise generation is reduced also.

The probable elastic deformation is another advantage of the preferred embodiment of the

invention.

This invention relates only to the mounting of a planetary gear assembly in a casing. The inner structure of the planetary gear assembly is arbitrary. Any variations are available with regard to planetary gear assemblies. There may be only three planetary shafts (5) and associated planetary gears (2) arranged symmetrically about the axis of the assembly.

Fig. 8 shows the structure of another form of planetary gear assembly. A sun gear (1) has two lateral cylindrical bosses (54) and (55) whose diameter is smaller than the diameter of the dedendum circle of the sun gear (1). The cylindrical bosses (54) and (55) contact and roll upon the planetary discs (6) and (7). This planetary gear assembly has been disclosed by laid open Japanese Patent Application No. 60-201146 (laid open on Oct. 11, 1985).

Fig. 9 shows the structure of an asymmetric planetary gear assembly. One planetary disc (6) of each planetary gear (2) is smaller than the dedendum circle. The other planetary disc (7) of the respective planetary gear (2) is larger than the addendum circle.

One cylindrical surface part (26) of the

ring gear (3) is smaller than the addendum circle.  
The other cylindrical surface part (27) is larger than  
the dedendum.

5       The sun gear (1) is formed in a one piece  
body with the sun shaft (59). Furthermore a larger  
sun column (54A) and a smaller sun column (55A) are  
shaped on both sides of the sun gear (1). The larger  
sun column (54A) is larger than the addendum circle.  
The smaller sun column (55A) is smaller than the  
10       tooth-root circle.

      The smaller sun column (55A) rolls on the  
larger planetary discs (7). The larger sun column  
(54A) rolls on the smaller planetary discs (6).

15       The larger planetary discs (7) roll on the  
larger cylindrical surface part (27). The smaller  
planetary discs (6) roll on the smaller cylindrical  
surface part (26).

      This asymmetric planetary gear assembly  
has been disclosed by laid open Japanese Patent  
20       Application No. 61-27337 (laid open on Feb. 6, 1986).  
An advantage of the asymmetric planetary gear assembly  
is the possibility that the sun shaft can be inserted  
and withdrawn after the whole gear assembly has  
constructed.

The advantages of the preferred embodiments of the invention are now explained.

5 (1) Because the ring gear is mounted in the casing in a relatively loose mounting, the ring gear is free for a certain amount of radial displacement. Radial displacement of the ring gear absorbs the effects of various dimensional errors and keeps rotation of gears smooth. Thus radial displacements effectively suppress noise generation.

10 (2) The clearances (e), (g) and (f) between the casing (B) and the ring gear attenuate transmission of vibrations to the casing. Thus noise generation is suppressed also by the clearances.

15 (3) The ring gear can deform elastically. Such elastic deformation increases transmission efficiency and reduces noise generation.

20 (4) The inventor has measured noise generation in electric washers. One electric washer was equipped with a planetary gear assembly fitted in a relaxation mounting in which this invention was embodied. Another electric washer was equipped with a

planetary gear assembly fixed in the casing by screws. A maximum weight (4.2 kg in the example) of clothes were washed by the washers.

5 The input rotation speed was 820 to 900 RPM (revolutions per minute). The reduction rate of the planetary gear assembly was 1/4.8. A belt-pulley reduction device reduced the motor rotation (1800 to 2000 RPM) to the input rotation (820 to 900 RPM).

10 The level of noise generated was measured at a spot spaced from the electric washers by 1 m. The level of noise from the washer incorporating the relaxation mounting was 48 phons. The level of noise from the washer incorporating the fixation mounting was 55 phons. Many parts other than reduction gears  
15 generate noise. This reduction in level of noise shows an excellent advantage of this invention.

(5) A mounting in which this invention is embodied can be adopted in reduction devices in various machines other than electric washers.

CLAIMS

1.           A mounting of a planetary gear assembly  
having a sun gear, a plurality of planetary gears and  
5           an outer, inner-toothed ring gear positioned in a  
            casing wherein a plurality of arcuate coupling parts  
            projecting from an outer cylindrical surface of the  
            ring gear are loosely fitted within complementary  
            arcuate recesses in an inner surface of the casing,  
10           the arrangement being such that a clearance exists  
            between the arcuate coupling parts and the recesses to  
            permit radial displacement of the ring gear within the  
            casing, a major part of the outer cylindrical surface  
            of the ring gear is spaced from the inner surface of  
15           the casing by a relatively wide space to permit  
            elastic deformation of the ring gear, and the plane in  
            which rotational forces are transmitted between the  
            planetary gears and the ring gear is displaced axially  
            from the plane in which rotational forces are  
20           transmitted between the arcuate coupling parts of the  
            ring gear and the casing.



2. A mounting of a planetary gear assembly in a casing as claimed in claim 1, wherein the ring gear is made from plastics.

5 3. A mounting of a planetary gear assembly in a casing as claimed in claim 1, wherein the ring gear is made from an aluminium die-casting.

10 4. A mounting of a planetary gear assembly in a casing as claimed in claim 2, wherein the ring gear is elastically deformable in radial directions.

15 5. A mounting of a planetary gear assembly in a casing as claimed in claim 4, wherein the ring gear is elastically deformable towards an ellipsoidal shape.

20 6. A mounting of a planetary gear assembly in a casing as claimed in claim 4, wherein the ring gear is elastically deformable towards a square shape.

7. A mounting of a planetary gear assembly in a casing as claimed in claim 2, wherein the ring gear



\* C L N S \*

is elastically deformable so that it can be twisted about its axis.

5           8.           A mounting of a planetary gear assembly in  
a casing as claimed in any one of claims 1 to 7,  
wherein the clearance between the arcuate coupling  
parts and recesses is within the range 0.5 mm to 3 mm.

10           9.           A mounting of a planetary gear assembly in  
a casing as claimed in any one of claims 1 to 8  
wherein each planetary gear consists of a planetary  
gear ring and two side planetary discs with disc parts  
larger than the addendum circle of the respective gear  
ring, and the ring gear comprises a gear part between  
15           cylindrical surface parts, the cylindrical surface  
parts being arranged for rolling upon the disc parts  
of the planetary discs.

20           10.           A mounting of a planetary gear assembly in  
a casing as claimed in claim 9, wherein the sun gear  
has side cylindrical bosses arranged for rolling on  
the disc parts of the planetary discs.



11. A mounting of a planetary gear assembly in a casing as claimed in any one of claims 1 to 10, wherein each planetary gear consists of a planetary gear ring, a side planetary disc larger than the addendum circle of the respective gear ring and a side planetary disc smaller than the dedendum circle of the respective gear ring, the ring gear comprises a gear part, a larger cylindrical surface part and a smaller cylindrical surface part being arranged for rolling upon the respective discs of the planetary gears, and the sun gear has a smaller column for rolling on the larger planetary discs and a larger column for rolling on the smaller planetary discs.

12. A mounting of a planetary gear assembly in a casing as claimed in claim 11 wherein, the sun gear is formed integrally with a sun shaft.

13. A planetary gear assembly mounted in a casing substantially as described hereinbefore with reference to the accompanying drawings and as shown in Figures 1 to 7, of Figure 8 or Figure 9.

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