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Improved disk brake assembly

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(56) Related Art
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US 4102438
US 3469658

ABSTRACT

An annular disk brake assembly having a housing mounted to a vehicle and a rotor disk mounted to a wheel of the vehicle. Annular brake pads extend parallel to the rotor disk within the housing and are mounted thereto with at least one brake pad being movable axially by means of an oil applied bladder mounted to the housing and moving the first brake pad axially against the disk brake. The rotor disk is adapted to slide axially to engage the second brake pad when pressure is applied to the rotor disk by means of the first brake pad and the bladder.

AUSTRALIA

Patents Act 1990

COMPLETE SPECIFICATION

STANDARD PATENT

Applicant:

YVON RANCOURT

Invention Title:

IMPROVED DISK BRAKE ASSEMBLY.

The following statement is a full description of this invention, including the best method of performing it known to me/us:

IMPROVED DISK BRAKE ASSEMBLY

Technical Field

The present invention relates to disk brakes and more particularly to improvements in large area contact disk brakes for vehicles.

Background Art

The disk brake of the present invention is a disk brake of the type described in US 5,330,034 issued July 19, 1994 and US RE 35055 issued Oct. 10, 1995 referring to full annular disk brakes for larger vehicles such as trucks. The concept of the full annular disk brake is now proposed for automobiles and light trucks and the present invention relates to a structure of a full annular disk brake for such vehicles.

There are obvious advantages in having a complete annular array of friction pads contacting an annular disk on both sides of the disk. The braking or thermal energy distribution is related directly to the thermal resistance associated with both sides of the interface where the heat is generated. In a full annular brake there is a large area to distribute the braking energy more efficiently.

It has also been found that vibrations between the inner and outer pads are the major causes for brake squeal.

The analysis of vibration response is of considerable importance in the design of brakes that may be subjected to dynamic disturbances. Under certain situations, vibrations may cause large displacements and severe stresses in the brake. The velocity of a vibrating system is in general, proportional to its frequency and hence a viscous damping force increases with the frequency of vibration. Forces resisting a motion also arise from dry friction along a non-lubricated surface. It is usually assumed to be a force of constant magnitude but opposed to the direction of motion. In addition to the forces of air resistance and external friction, damping forces also arise because of imperfect elasticity or internal friction, called hysteric damping, within the body. The magnitude of such a force is independent of the frequency but is proportional to the amplitude of vibration or to the displacement.

In a brake system, dynamic loading produces stresses and strains, the magnitude and distribution of which will depend not only on the usual parameters encountered previously but also on the velocity of propagation of the strain waves through the material of which the system is composed. This latter consideration, although very important when loads are applied with high velocities, may often be neglected when the velocity of application of the load is low. Since dynamic loading is conveniently considered to be the transfer of energy from one system to another, the concept of configuration (strain energy) as an index of resistance to failure is important. One of the important concepts is that the energy-absorbing capacity of a member, that is, the resistance to failure is a function of the volume of material available, in contrast to the resistance to failure under static loading, which is a function of cross-sectional area or section modulus.

One of the main problems in adapting the technology of a full annular brake system of the type described in the above mentioned patents is the consideration of weight and cost. It would be unrealistic, no matter what the advantages, to assume that the a new full annular brake system would be accepted on the market at a price substantially higher than present day disk brakes. Furthermore any increase of weight compromises the fuel consumption.

Disclosure Of The Invention

It is an aspect of the present invention to provide a disk brake assembly for a vehicle wheel wherein the wheel includes a hub journaled to an axle on the vehicle, the disk brake assembly comprising a housing mounted to the vehicle and at least an annular rotor disk within the housing and means mounting the disk to the wheel, the disk having at least a first radial planar annular friction surface and the housing including a first annular brake shoe provided adjacent the first friction surface of the disk and the brake shoe being movable axially towards and away from the first friction surface, means provided for restraining the first brake shoe from rotating with the disk, the housing including an annular radial wall parallel to the first brake shoe, and an annular fluid expandable bladder extending between the first annular brake shoe and the radial wall, whereby upon expansion of the bladder the first brake shoe moves axially to frictionally engage the first friction surface of the disk, the disk brake assembly further comprising at least a first elastically deformable means for disengaging the first brake shoe from frictional contact with the first frictional surface of the rotor disk upon release of the fluid from the bladder using energy stored in said first elastically deformable means due to the deformation of said first elastically deformable means.

In a more specific embodiment of the present invention the radial disk is provided with a second annular friction surface, parallel to the first and on an opposite side of the rotor disk wherein the first and second friction disks have different radii, and a second annular brake shoe adjacent the second annular friction disk wherein brake squeal will be reduced.

In a still more specific embodiment of the present invention, the means for retaining the first brake shoe includes a brake shoe backing plate having an annular periphery and the housing includes a concentric wall having an internal surface radially adjacent the periphery of the first brake shoe while the inner surface of the concentric wall and the periphery of the first brake shoe have mating interdigital elements which allow axial movement of the first brake shoe relative to the concentric wall but prevents peripheral movement of the first brake shoe relative to the concentric wall of the housing.

In a still more specific embodiment of the present invention, the means for disengaging the first brake shoe from the first friction surface of the rotor disk is at least one rolling seal provided between axially generated adjacent surfaces of the annular radial wall of the housing and the first brake shoe.

The features of the present invention can be utilized for large trucks as well.

Brief Description Of The Drawings

The invention will now be described in derail having reference to the accompanying drawings in which:

Fig. 1 is an exploded fragmentary perspective view of an embodiment of the disk brake in accordance with the present invention;

Fig. 2 is a fragmentary radial cross-section taken through the assembled disk brake;

Fig. 3 is a radial cross-section similar to Fig. 2 but including further elements;

Figs. 4a and 4b are enlarged fragmentary cross-section taken along the same section as Fig. 3 but showing the elements in a different operative position;

Fig. 5 is a fragmentary radial cross-section similar to Fig. 3 but showing another embodiment;

Fig. 6 is a fragmentary perspective view, partially in cross-section, of another embodiment of the present invention;

Figs. 7a and 7b are enlarged fragmentary radial cross-sections of the embodiment of Fig. 6 showing certain elements in different operative positions;

Fig. 8 is a fragmentary perspective view, partly in cross-section, of the embodiment shown in Figs. 6 and 7;

Fig. 9 is an exploded fragmentary perspective view of yet another embodiment of the present invention; and

Fig. 10 is a fragmentary enlarged radial cross-section of the embodiment shown in Fig. 9.

Mode For Carrying Out The Invention

Referring now to the drawings, and more particularly to Figs. 1 to 4, a disk brake assembly 10 for an automobile is illustrated having a housing in the form of a shell 12. The housing has a cylindrical wall 14 with a corrugated inner surface 16 having valleys 16a and ribs 16b. The housing 12 includes a radial annular wall 18 provided with an annular brake pad lining 20. The ribs 16b are relatively flat and represent valleys on the outer surface 17 while ribs 17a correspond to valleys 16a.

The cylindrical wall 14 also includes a radial flange 15.

The housing 12 also includes an annular radial wall 22 to which is mounted an annular cylindrical corrugated rim 24 adapted to fit within the corrugated inner surface 16 of the wall 14 and is retained therein by flange 15. The ribs 24a of the corrugated rim 24 fit in the valleys 16a of surface 16 while the valleys 24b correspond to the ribs 16b of the housing wall 14. Thus, the housing 12 will be locked against circumferential movements relative to the radial wall 22. The radial wall 22 has a hub portion 26 which can be bolted to a flange on an axle (not shown) of the vehicle. The radial wall 22 also includes an annular radial planar wall portion 28 and a cylindrical flange 30 as shown in Fig. 2.

An indented detent 70 is provided in the housing wall 14 in order to lock the housing 12 against axial movement relative to the radial wall 22. The detent 70 protrudes inwardly to engage the edge of rim 24.

An annular rotor disk 32 includes radial planar friction surfaces 34 and 36 and a cylindrical annular rim 38 having an inner corrugated concentric surface 40 with ribs 40a and valleys 40b. A hub adapter 42 includes a radial wall portion 44 adapted to be mounted to a vehicle wheel (not shown in the embodiment of Fig. 8) and a cylindrical corrugated wall 46. The wall 46 has ribs 46a and valleys 46b which are adapted to fit within the inner surface 40 of the rim 38 of rotor disk 32. Thus, the rotor disk 32 will be locked against rotational movement relative to the hub adapter 42 but is slidable axially thereon. Since the hub adapter 42 is mounted onto a vehicle wheel the rotor disk 32 will rotate with the wheel. The rotor disk 32 is ventilated and therefore has radially extending ventilation passages 48 communicating with openings 49 in housing wall 14. As shown in Figs. 1, 2 and 3, there are axial opening 48a that intersect radial openings 48 so as to ensure that as much air as possible passes through the rotor disk 32.

A brake shoe 50 includes brake linings 52 and a backing plate 54. The brake shoe 50 includes a corrugated peripheral edge 51 engaging the inner surface 16 of the cylindrical wall 14. Thus, the brake shoe 50 can slide axially but is retained against rotational movement relative to the housing 12.

An annular bladder 56 is provided between the wall 28 and the backing plate 54. When fluid such as oil is fed into the bladder 56 it will expand, moving the brake shoe 50 axially towards the friction surface 36 of rotor disk 32. The rotor disk 32 will also slide axially on the hub 42, in response to the force exerted by the bladder 56, and the radial friction surface 34 will come in frictional contact with the brake linings 20. Thus, when it is necessary to apply the brakes, the bladder 56 is expanded. However, to release the brakes the oil is allowed to drain from the bladder 56, thereby releasing the axial force on the brake shoe 50, allowing the disk rotor 32 to rotate freely within the housing 12.

Referring now to Figs. 3, 4a and 4b, the rolling seals 62, 64 will be described. A pair of rolling seals 62 are located, in the present embodiment, on the outer surface of corrugated wall 46 of the hub adapter 42 and are formed to the contour of the corrugated surface. Pairs of circumferentially extending grooves 46c, 46d are defined in wall 46 to receive the rolling seals 62a and 62b respectively. As shown in Fig. 3, the pair of rolling seals 62a and 62b are pre-compressed when inserted between the hub 42 and the rim 38 of the rotor disk 32. Retainer ring 63 may be provided to hold seal 62a in place. Seal 62b is retained by groove 65 formed in wall 46. Retainer ring 63 is formed with convexly curved surface 63b to support seal 62a and control the deformation of the seal 62a as will be described. Likewise the groove 65 is also formed with convexly curved surface 65b to control the deformation of seal 62b. When the rotor disk 32 slides on the hub adapter 42, as previously described, the rolling seals 62a and 62b will be deformed in the direction of the path of the rotor disk 32, as illustrated by the arrow in Fig. 4b, when force is exerted by the bellows 56 on the brake shoe 50. When the brakes are released, the rolling seals 62a, 62b will be restored because of the energy stored therein, and will return to the shape as shown in Fig. 4a, thereby moving the rotor disk 32 and thus drawing the friction surface 34 away from the brake pad 20. The rolling seals 62a and 62b can be selected to provide the right amount of clearance to avoid the drag which might occur when the rotor disk 32 remains in contact with the friction pad 20. It is important that only a slight clearance be provided in order to avoid undue pedal movement.

In the same manner, rolling seal 64 which is located in circumferential groove 30a on the flange 30 engages the flange of backing plate 54 on the brake shoe 50, and will act to return the brake shoe 50 away from the friction surface 36 of the rotor disk 32 when the fluid is drained from the bellows, in order to eliminate drag of the brakes. Wiper 66 on the housing 14 seals the brake shoe from debris and dust.

Referring now to Fig 5 there is shown a modification to the brakes of the present invention. The elements which in Fig. 5 are similar to those in Figs. 1 to 4 have been raised by 100.

More specifically, the housing 112 is a shell having a cylindrical wall 114 that now includes a smooth cylindrical portion 155 adjacent the corrugated portion 116. Likewise, the radial wall 122 has a smooth cylindrical wall portion 160 adjacent the corrugated peripheral wall 124. Thus, when the radial wall 122 is received within the shell or housing 112 the smooth wall portion 160 of radial wall 122 will fit in the smooth cylindrical wall portion 155 of the housing 112. A ledge 155a is formed between the corrugated wall portion 114 and the smooth wall portion 155 which acts as a stopper for the radial wall 122 having complementary peripheral surfaces, that is between the corrugated portion 124 and the smooth portion 160. This will eliminate the need for indents 70 as shown in the embodiment of Figs. 1 to 4.

The cross-section of Fig. 5 is taken through the radial wall 122 at exactly the position where the bleed openings 170 and 172 for the bladder 156 are located.

Referring back to Fig. 1, the wall 28 is adapted to receive strain sensor 60. These strain sensors 60 may be the type known under Trademark MULTIDYN and described in U.S. Patent No. 5,522,270 issued June 4, 1996 to THOMSON-CSF. The strain sensor 60 can provide valuable information on the braking efficiencies and the wear of the brake shoes.

The strain sensor 60 extends somewhat tangentially to the wall 28 and can, therefore, monitor the torque being applied between the hub 26 and the cylindrical flange 30 of spider 22. With the information which can be obtained from strain sensor 60, the temperature of the brakes can be monitored by means of suitable micro processors. For instance, when the brakes are applied, the pressure is known, and if the heat should increase the torque will be reduced. Increased temperature of the brakes will normally signal brake deterioration or malfunction.

Other criteria can also be determined logically from the known pressure, and the torque information provided by the strain sensor 60.

A further embodiment of the present invention is disclosed in Figs. 6 to 8. The reference numerals in these figures, designating elements which correspond to similar elements in the embodiment of Figs. 1 through 4, have been raised by 200.

The disk brake 210 is shown mounted to the hub H of a wheel W (Fig. 8). Thus, the hub adapter 242 is mounted to the hub H by means of bolts. The hub adaptor 242 includes a corrugated wall 246 (Figs. 6, 7a and 7b) including ribs 246a and valleys 246b which mate with the corrugated inner surface 240 of rim 238 which is an integral part of the rotor disk 232.

Fig. 6 illustrates the various elements of this embodiment but without the rotor disk 232. The rotor disk 232 is illustrated in Figs, 7a, 7b and 8.

As previously described, the rotor disk 232 is restrained against circumferential rotation relative to the hub adapter 242 but the rotor disk 232 can slide axially relative to the hub adapter 242. The rim 238 is notched along each edge thereof to receive rolling seal housings 263 and 265 respectively. Each rolling seal housing 263 and 265 is made of thin wall stamping and is formed as an annular channel having a lateral width which is greater than the diameter of the rolling seals 262a or 262b respectively. The area of the channel is represented by the numeral 263b and 265b in Figs. 7a and 7b. The bite portion of the channel forms a ramp which is sloped downwardly from left to right in Figs. 7a and 7b. Thus, when the rotor disk 232 is slid from right to left to engage the brake shoe represented by brake pad 220, the rim 238 and rolling seal housings 263 and 265 will move towards the left from the position shown in Fig. 7a to the position shown in Fig. 7b.

Observing the position of the rolling seals 262a and 262b, in Fig. 7b, one would recognize that the rolling seals are somewhat squeezed by ramps of the channels 263 and 265. Thus, the rolling seals have stored energy which can overcome the forces applied to the rotor disks 232 by the bladder 256 when the fluid is released from the bladder 256, as will be described. Thus, the rolling seals 262a and 262b will draw the rotor disk 232 away from the brake pad 220 to a position shown in Fig. 7a. The rolling seals 262a and 262b will slide on surface 246 in order to compensate for wear of the brake pad 220. The rolling seals 262a and 262b also serve as a suspension to dampen the vibrations between the rotor disk 232 and the hub adaptor 242.

In the present embodiment, the housing shell 212 represented by cylindrical wall 214 and radial wall 218 is a thin wall stamping. A skirt 218a is formed at the inner edge of the wall 218 to allow the brake pad 220 including a backing wall 221 to be snapped into position within the housing as shown in Figs, 7a and 7b. The shell 212 may be assembled from the left end side of Figs. 7a and 7b, with the portion 255 extending over and concentric with the cylindrical wall portion 224 of the radial wall 222. A cap 283 which may be hinged in two parts surrounds the enlarged collar portion formed by the extension 255 and has a radial skirt on each edge thereof to form a channel to lock the wall 224 of the radial wall 222 within the housing 212.

Fig. 6 shows how the two-part cap 283 with short extensions 283a and 283b overlap each other. A coupling member 284 extends over the joint so formed by the ends of the hinged cap 283. The coupling member 284 includes openings 286 through which pins 288 can pass. These pins are shaped and pass in an area coincident with the valleys in the cap 283.

The bladder 256 is shown here with a U-shaped membrane 256a having leg portions which are inserted into slots 276 and 278 within the radial wall 222. Reinforcement rings 280 and 282 are also placed in these slots to prevent the membrane 256a from expanding radially.

The brake shoe 250 including the brake pad 252 and backing plate 254, have a T-shaped configuration with the foot of the T 251 folding back the membrane 256a to form an M, as shown in Figs. 7a and 7b. Thus, when fluid such as oil is injected through the inlet 272 as shown in Fig. 7a, the bladder 256 will expand in the axial direction as shown in Fig. 7b.

A further ring 230 (corresponding to the flange 30 in Figs. 1 to 4) is also inserted into the groove 278 but extends axially from the radial wall 222 to support a rolling seal 264. The backing plate 254 is provided with a channel shaped groove 257 having the same construction as that described with respect to channels 263 and 265 herein. Thus, when the bladder 256 is expanded, the brake shoe 250 moves towards the left in the drawings of Figs. 7a and 7b, applying an axial force against the rotor disk 232 by means of the brake pad 252, frictionally engaging the friction surface 236, and further pressing against the rotor disk 232 such that the friction surface 234 engages the brake pad 220. Once oil is released from the bladder 256, the rolling seal 264 which has been somewhat compressed as shown in Fig. 7b, will overcome the reduced axial force, thereby retracting the brake shoe 250 from the friction surface 236 of rotor disk 232. Simultaneously, the rolling seals 262a and 262b will retract the rotor disk 232 from frictional engagement with the brake pad 220.

A wiper 268 is shown mounted to the backing plate 254 to prevent debris from entering into the rolling seal area 264. Similar wipers (see wiper 66 in Fig. 3) can be provided at other practical locations such as between the backing plate 254 and the cylindrical housing wall 214.

A further embodiment is shown in Figs. 9 and 10. Reference numerals corresponding to elements which correspond to elements shown in the embodiment of Figs. 1 through 4 have been raised by 300. The rotor disk 332 has friction surfaces 234 and 236 at different radial distances from the axis of rotation of the rotor disk. As seen in Fig. 10 more clearly, the opposed friction surfaces 334 and 336 are staggered. The corresponding brake pads 320 and 352 are also constructed to correspond to the radially staggered friction surfaces 334 and 336.

The housing wall 314 is accordingly formed in order to accommodate this difference in radius. It has been found, that the amplitude and difference in amplitude of the vibration between pads such as pads 20 and 52 in the embodiment of Figs. 1 through 4 were the major factors contributing to the generation of brake squeal. Brake squeal has been found to be a result of self induced vibration phenomena of the various parts. Under certain situations, vibrations may cause large displacements and severe stresses in the brake. The velocity of a vibrating system is, in general, proportional to its frequency and enhance a viscous stamping force increases with the frequency of vibration.

It has been found that by having the brake pads 320 and 352 as well as the corresponding annular friction surfaces 334 and 336 on the rotor disk 332 at different radii, these vibrations are at different frequencies and thus reduce the chances of harmonics which helps to reduce the brake squeal and stresses which might occur in the disk brake.

THE CLAIMS DEFINING THE INVENTION ARE AS FOLLOWS:

1. A disk brake assembly for a vehicle wheel wherein the wheel includes a hub journaled to an axle on the vehicle, the disk brake assembly comprising a housing mounted to the vehicle and at least an annular rotor disk within the housing and means mounting the disk to the wheel, the disk having at least a first radial planar annular friction surface and the housing including a first annular brake shoe provided adjacent the first friction surface of the disk and the brake shoe being movable axially towards and away from the first friction surface, means provided for restraining the first brake shoe from rotating with the disk, the housing including an annular radial wall parallel to the first brake shoe, and an annular fluid expandable bladder extending between the first annular brake shoe and the radial wall, whereby upon expansion of the bladder the first brake shoe moves axially to frictionally engage the first friction surface of the disk, the disk brake assembly further comprising at least a first elastically deformable means for disengaging the first brake shoe from frictional contact with the first frictional surface of the rotor disk upon release of the fluid from the bladder using energy stored in said first elastically deformable means due to the deformation of said first elastically deformable means.

2. A disk brake assembly as defined in claim 1, wherein the means for restraining the first brake shoe from rotation with the rotor disk comprises providing the housing with a concentric wall having an internal surface radially adjacent the periphery of the first brake shoe, providing the internal surface of the concentric wall and the periphery of the first brake shoe with mating inter-digital elements which allow axial movement of the first brake shoe relative to the surface of the concentric wall, but preventing the circumferential movement of the first brake shoe relative to the concentric wall of the housing.

3. A disk brake assembly as defined in claim 2, wherein the inter-digital elements include a plurality of circumferentially spaced-apart axially extending ribs on the inner surface of the concentric wall mating with corresponding valleys on the periphery of the first brake shoe.

4. A disk brake assembly as defined in claim 2, wherein the housing further includes an annular radial skirt depending from the concentric wall located on the opposite side of the rotor disk from the first brake shoe, and a second brake shoe is provided on the annular skirt facing a second friction surface on the rotor disk.

5. A disk brake assembly as defined in claim 2, wherein the means for disengaging the first brake shoe from the first friction surface of the rotor disk is at least one rolling seal provided between an axially generated surface of the brake shoe and an axially generated cylindrical surface of the first radial wall of the housing which extends parallel to and adjacent the axially generated surface of the brake shoe such that the rolling seal can store energy when force is being applied on the brake shoe to frictionally engage the frictional surface of the rotor disk by means of the fluid expandable bladder and whereby the stored energy is sufficient to retract the brake shoe from the first friction surface of the rotor disk when fluid is released from the expandable bladder.

6. A disk brake assembly as defined in claim 4, wherein the means mounting the rotor disk to the wheel comprises a hub adapter adapted to be mounted for rotation with the wheel, the hub adapter including a cylindrical outer surface, the rotor disk including a central opening defined by an inner cylindrical surface, and inter-digital elements are provided on the

outer surface of the hub adapter and the inner cylindrical surface of the rotor disk, whereby said interdigital elements mate to allow the rotor disk to slide axially on the hub adapter but to restrain the rotor disk against rotary circumferential movement relative to the hub adapter, and at least one rolling seal is located between the outer cylindrical surface of the hub adapter and the inner cylindrical surface of the rotor disk and arranged such that when the rotor disk is moved axially against the second brake shoe under the axial force which is applied by the fluid within the expandable bladder, the rolling seal is deformed to conserve energy such that when the fluid is released from the expandable bladder, the conserved energy in the rolling seal will be effective to disengage the second friction surface of the rotor disk from the second brake shoe.

7. A disk brake assembly as defined in claim 6, wherein there are two axially spaced apart rolling seals between the inner cylindrical surface of the rotor disk and the outer cylindrical surface of the hub adapter.

8. A disk brake assembly as defined in claim 7, wherein the rolling seals are provided in channels formed on the outer cylindrical surface of the hub adapter.

9. A disk brake assembly as defined in claim 5, wherein the first radial wall of the housing includes a cylindrical flange extending towards the rotor disk and the first brake shoe includes a backing plate having a cylindrical portion and a rolling seal is mounted in a groove on the flange and engages the cylindrical wall portion of the backing plate.

10. A disk brake assembly as defined in claim 7, wherein the rotor disk includes a rim which defines the inner cylindrical surface and a pair of axially spaces apart grooves are provided in the rim and rolling seal channels are provided in the groove on the rim to receive the rolling seals, wherein each channel includes a bite portion having a sloping surface decreasing in depth from one side of the channel to the other and the axial extent of the channel being greater than the axial extent of the rolling seal so that the rolling seal can be compressed as the rotor disk is moved towards the second brake pad, and the rolling seals engage the outer cylindrical surface of the hub adapter.

11. A disk brake assembly as defined in claim 5, wherein the first brake shoe includes a backing plate and the backing plate defines a cylindrical surface opposite the radial cylindrical surface defined by the first radial wall, and a groove is defined in the cylindrical surface of the backing plate to receive the rolling seal, the groove having a radial extent greater that the radial extent of the rolling seal, and a bite portion of the groove has an inclined wall configuration to provide compression to the rolling seal when the brake shoe moves towards the rotor disk.

12. A disk brake assembly as defined in claim 1, wherein the bladder is an annular enclosed envelope of elastic material and extends between the first radial wall of the housing and the first brake shoe.

13. A disk brake assembly as defined in claim 1, wherein the bladder includes an elongated annular membrane of elastic flexible material having parallel edges and the edges being sealingly engaged to the first radial wall of the housing such that the bladder

is formed between the membrane and the first radial wall.

14. A disk brake assembly as defined in claim 13, wherein the first brake shoe includes a backing plate and the backing plate has a cross-section in the form of a T with the leg of the T extending axially away from the rotor disk and engages the membrane forming the bladder, such that the cross-section of the membrane had an M shape in cross-section.

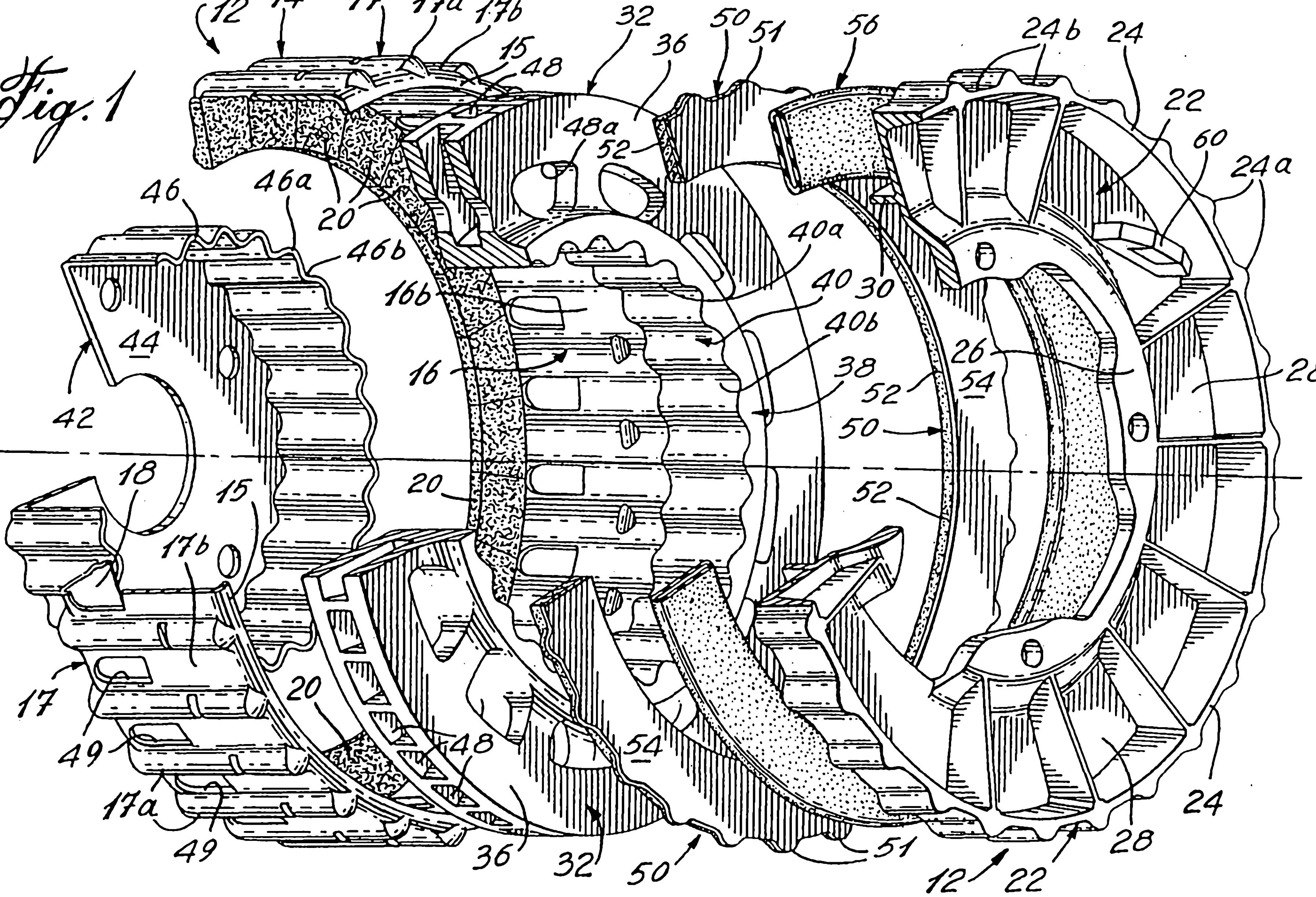
15. A disk brake assembly as defined in claim 13, wherein the first radial wall of the housing is provided with a pair of radially spaced apart circumferential slots, and the ends of the membrane are received in respective slots for sealing engagement with the wall.

16. A disk brake assembly as defined in claim 4, wherein the first and second annular friction surfaces on the rotor disk have staggered radii and the first and second annular brake shoes are located at different radii corresponding to the respective radii of the first and second friction engagement surfaces.

17. A disk brake assembly substantially as described herein with reference to and as illustrated in the accompanying drawings.

Dated this 27th day of January 2004
YVON RANCOURT
By their Patent Attorneys
GRIFFITH HACK
Fellows Institute of Patent and
Trade Mark Attorneys of Australia

Fig. 1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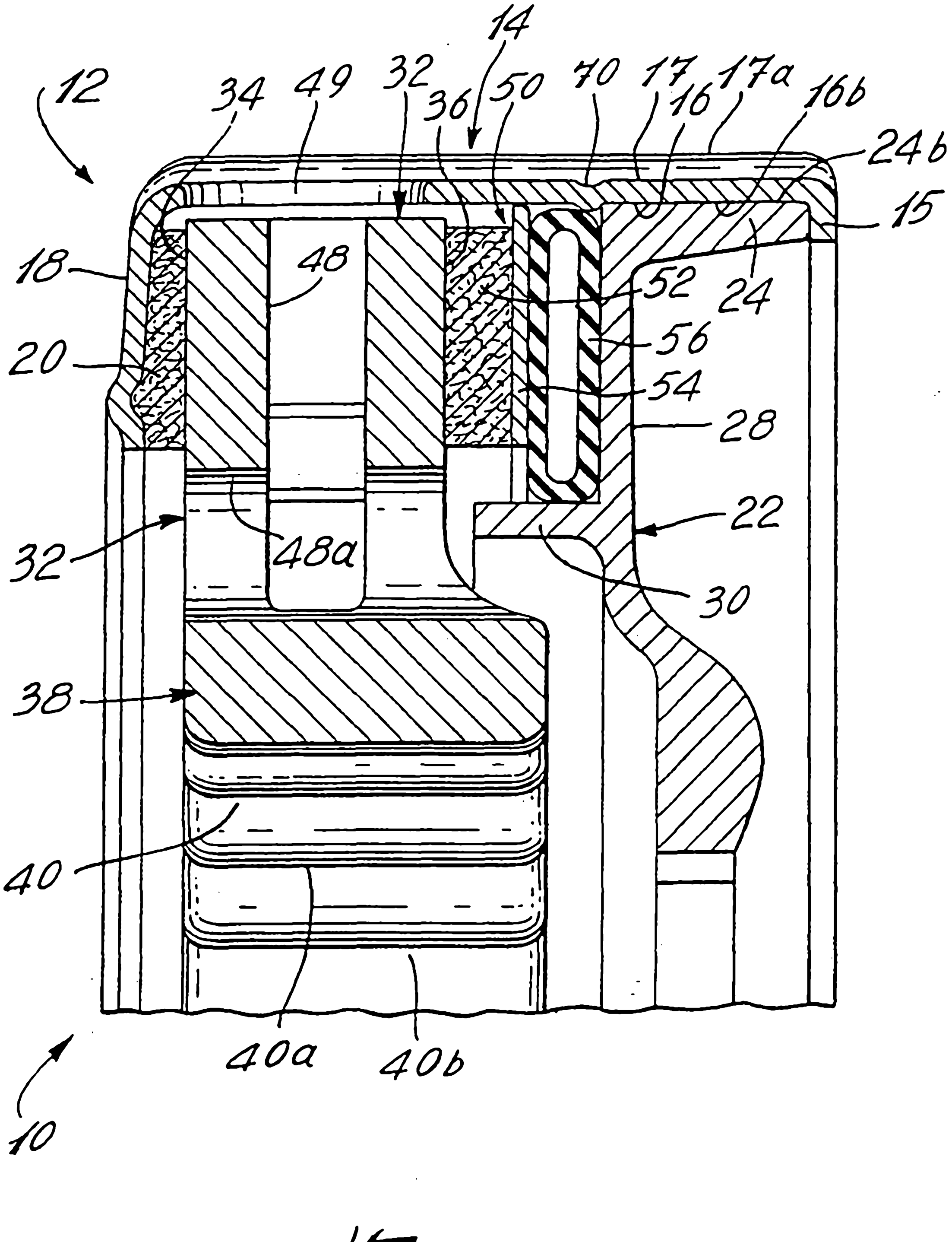
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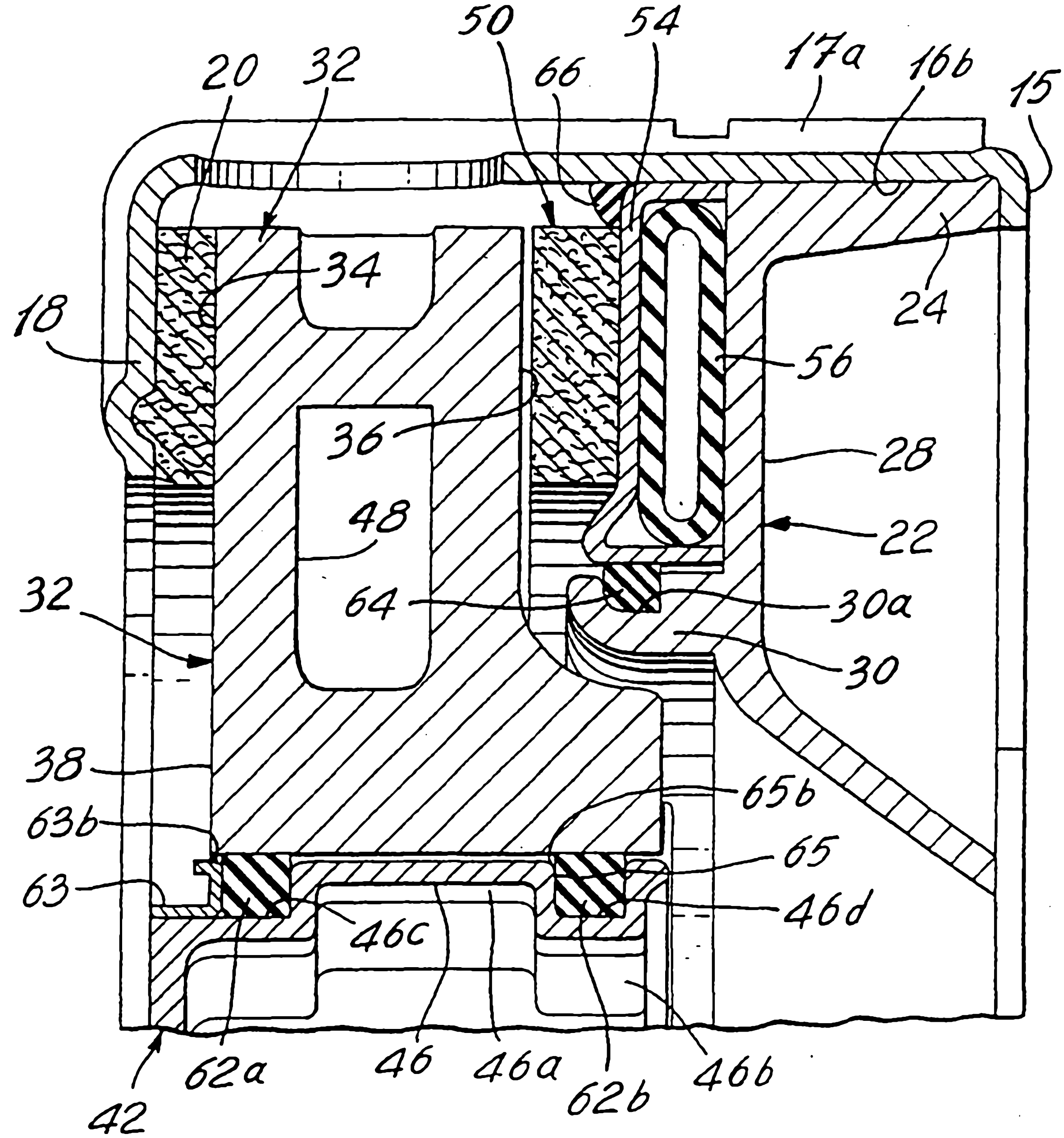
Fig. 2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46b
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Fig. 3

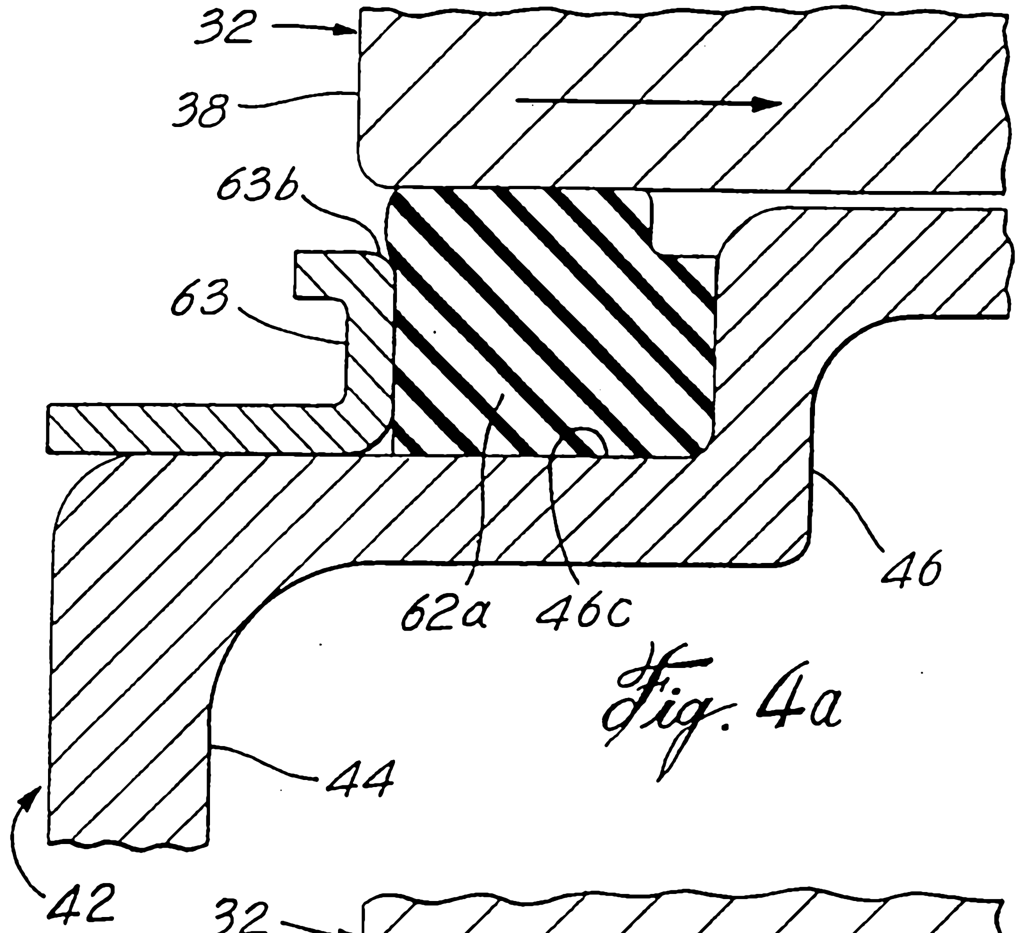

Fig. 4a

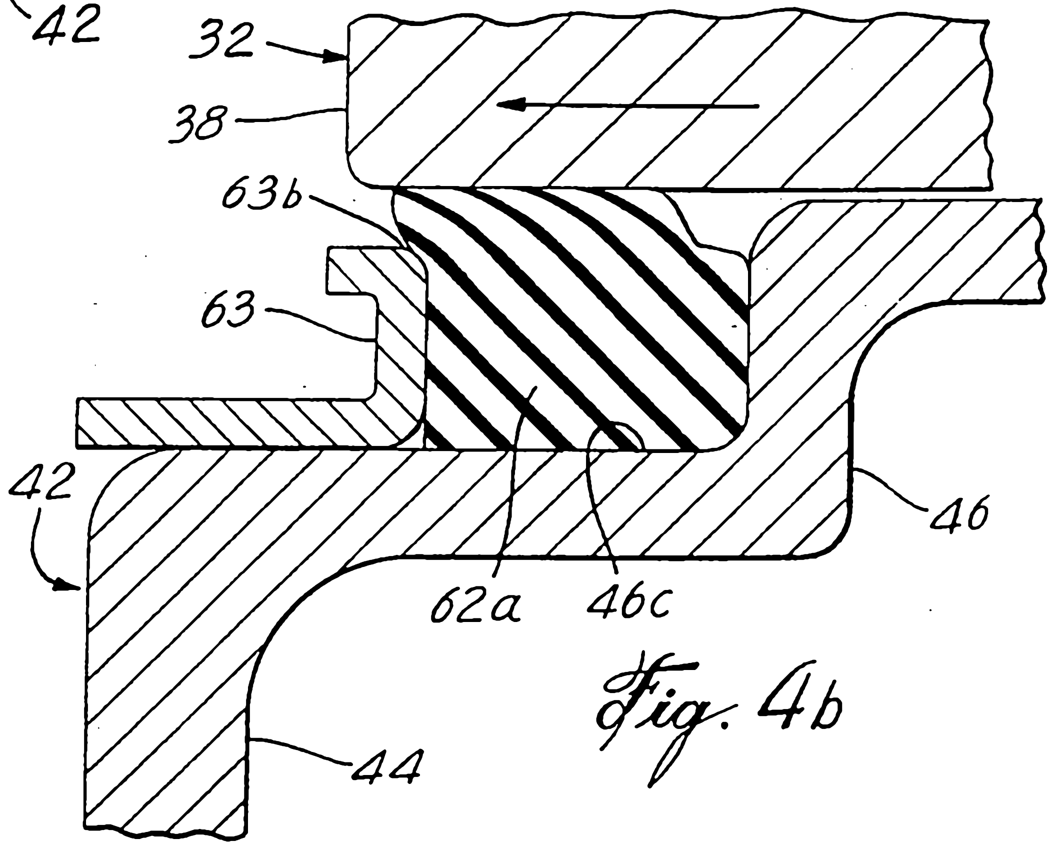

Fig. 4b

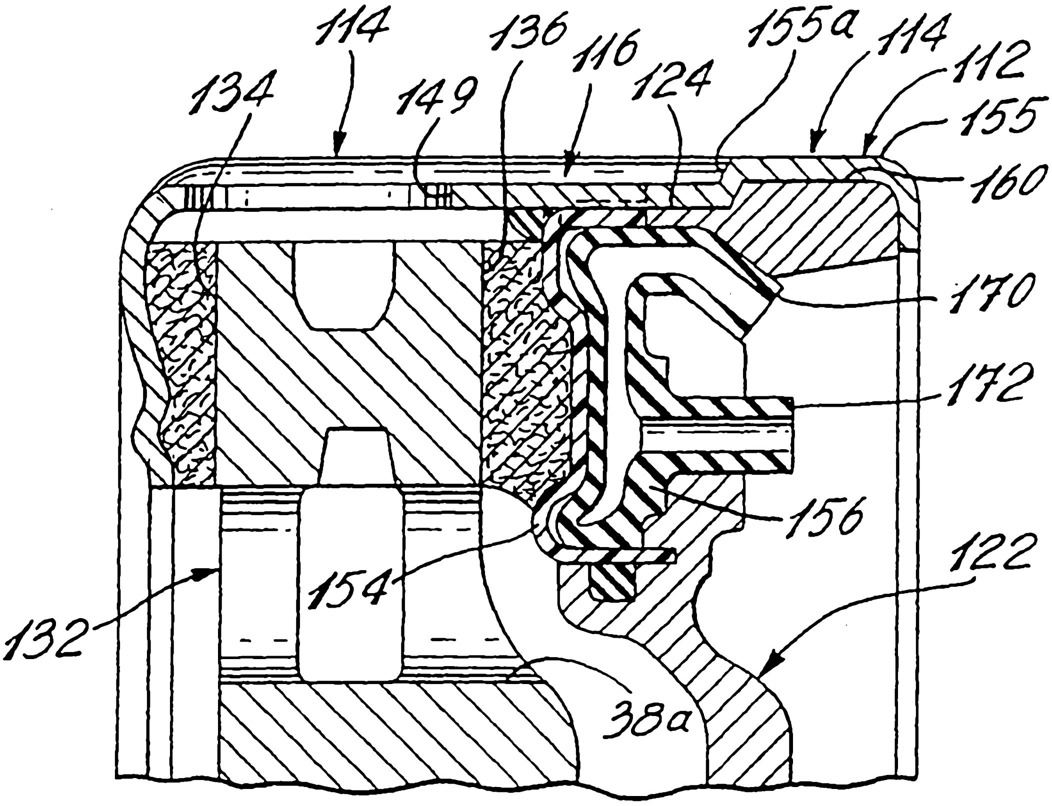
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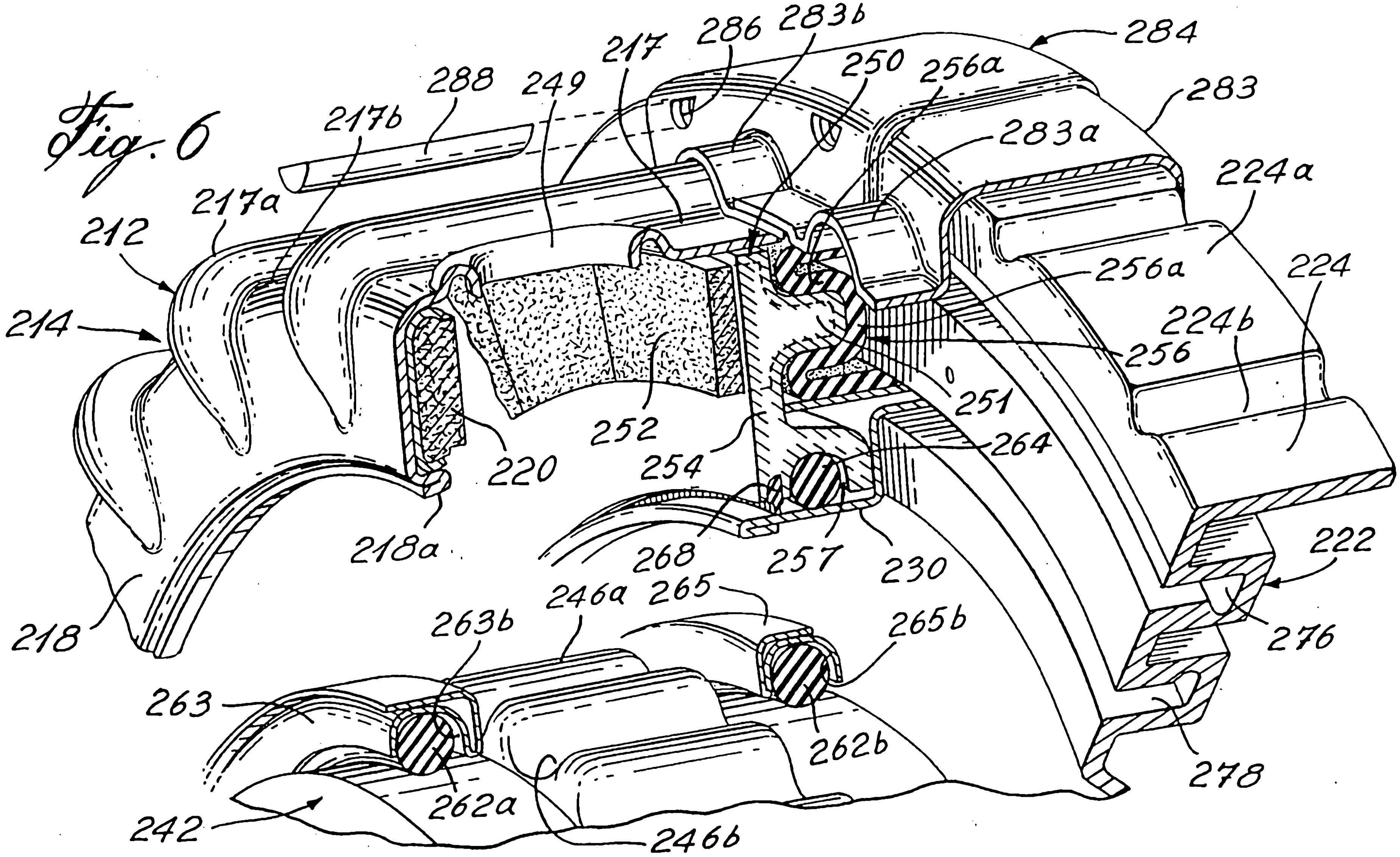
Fig. 5

Fig. 6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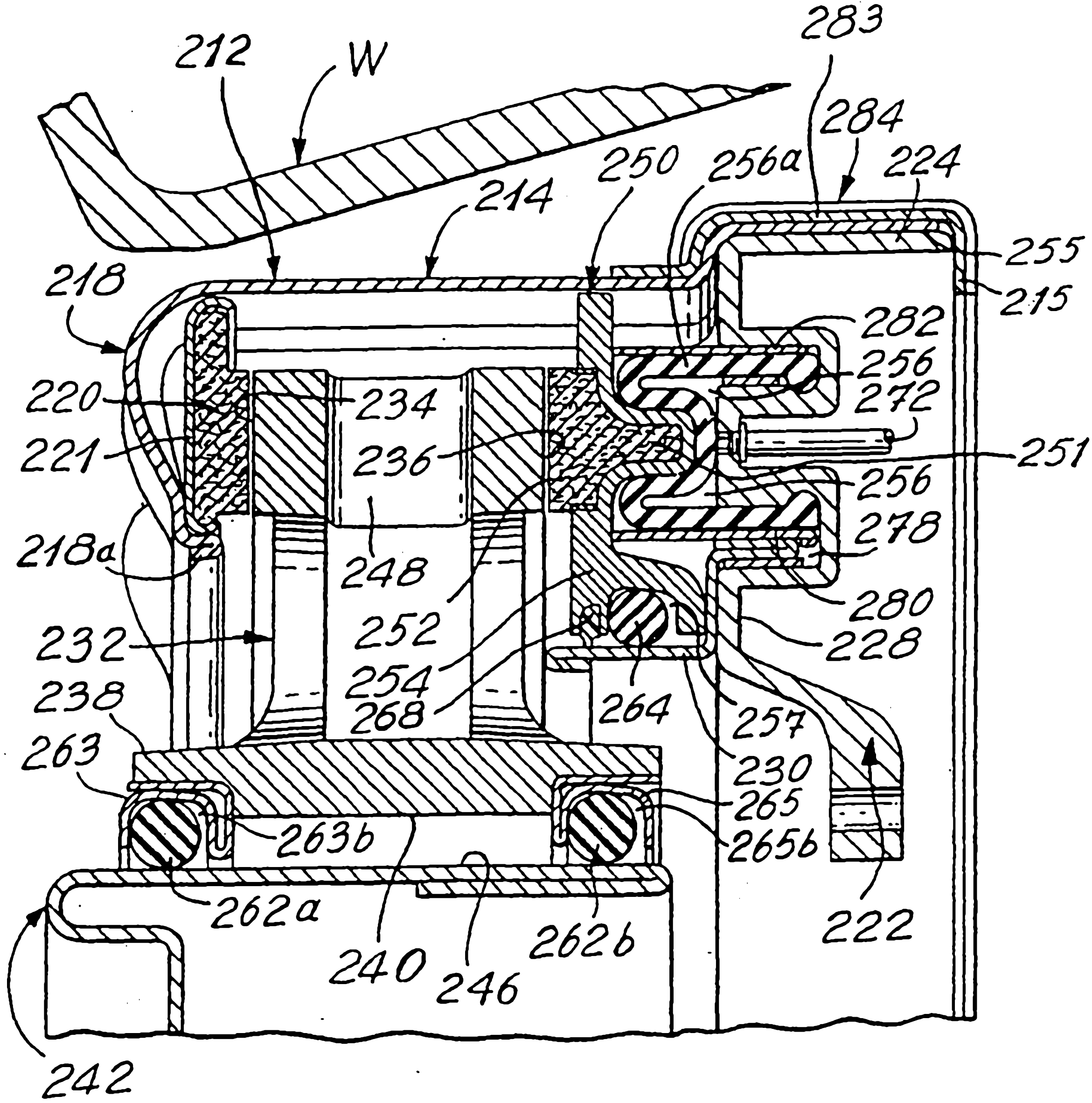
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278
280
252
228
232
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238
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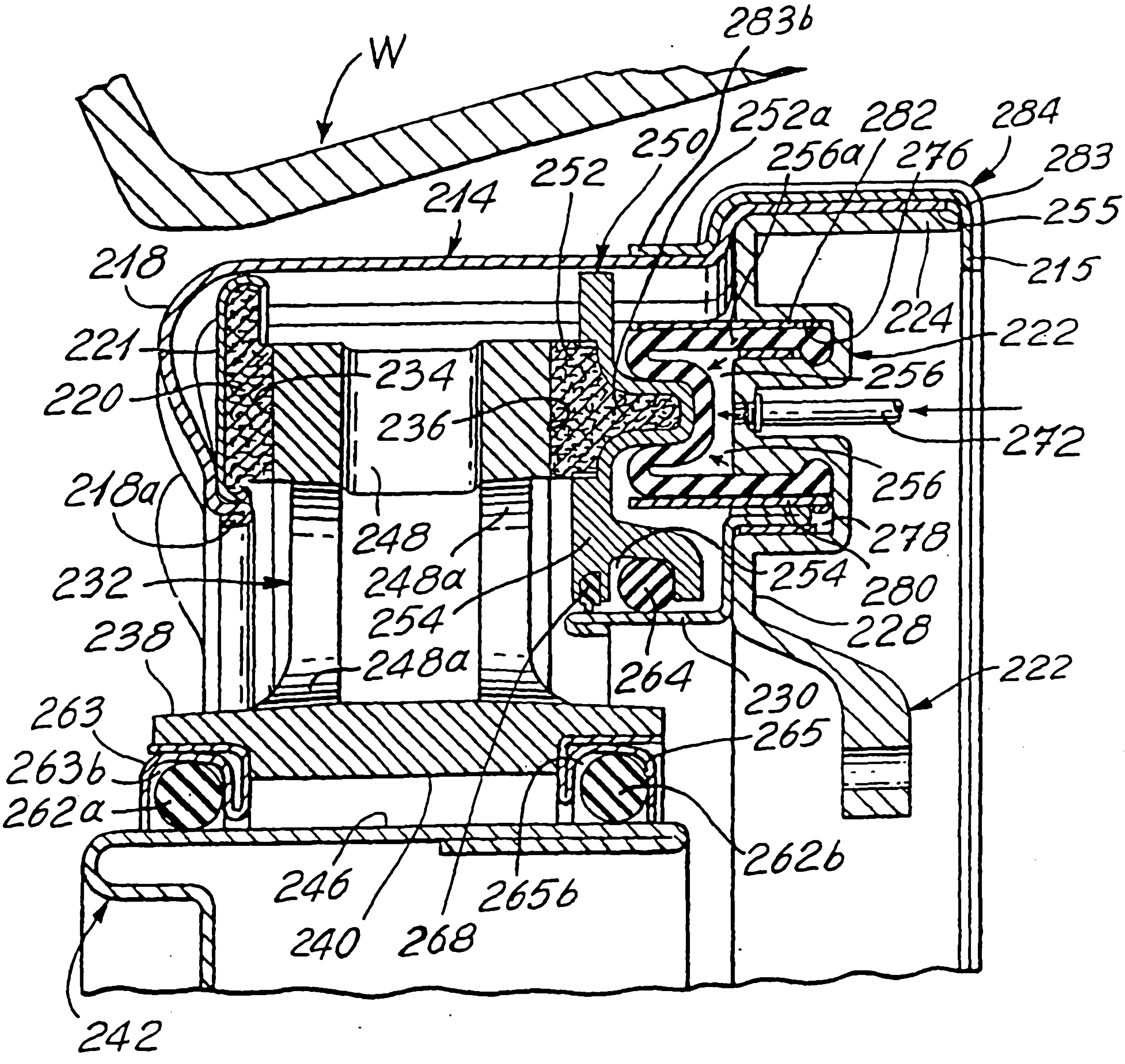
265
263b
265b
262a
240
246
262b
222
242

Fig. 7a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236
272
256
218a
278
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248a
232
254
280
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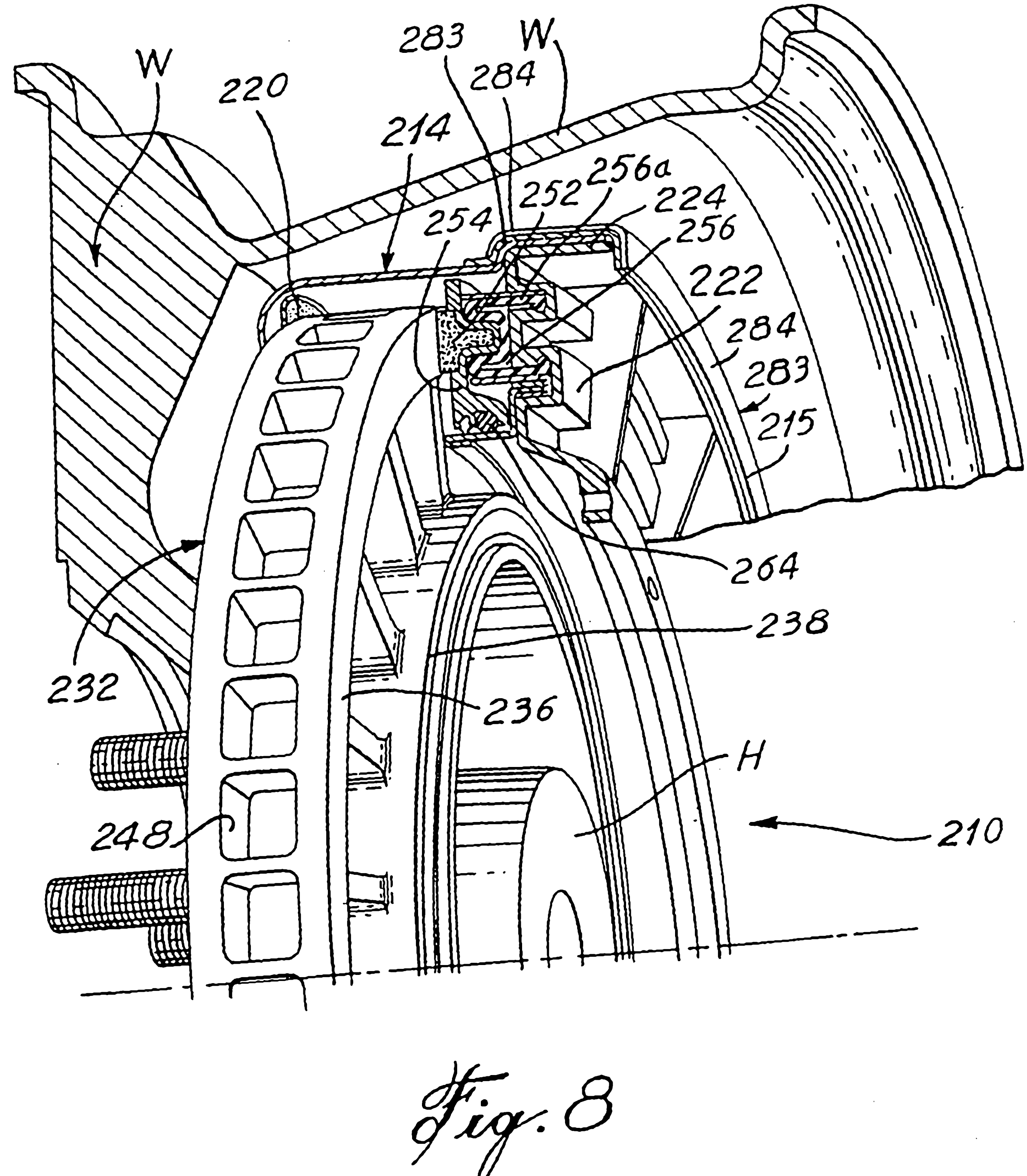
263b
262a
262b
246
265b
240
268
242

Fig. 7b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H
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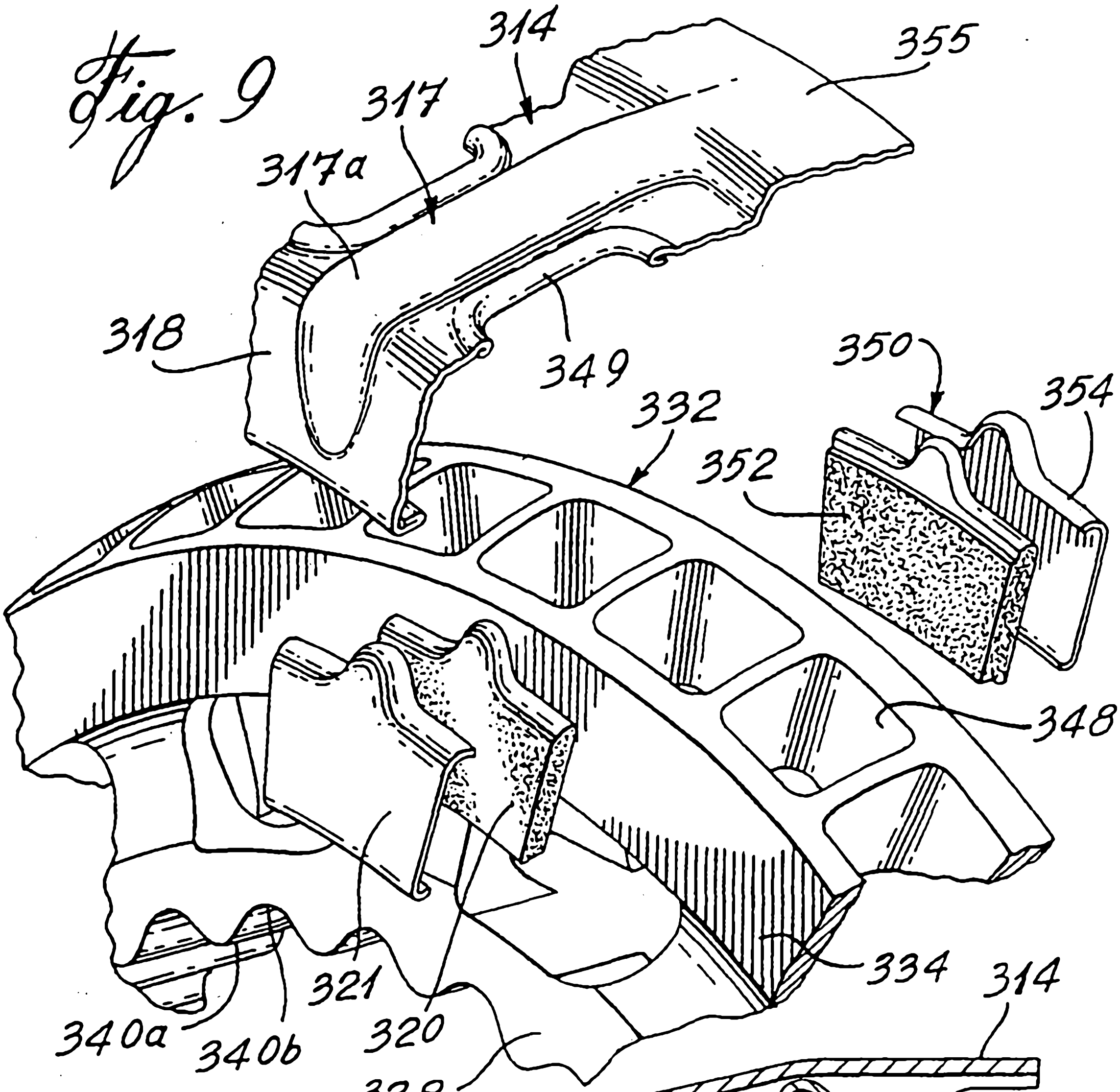
248

Fig. 8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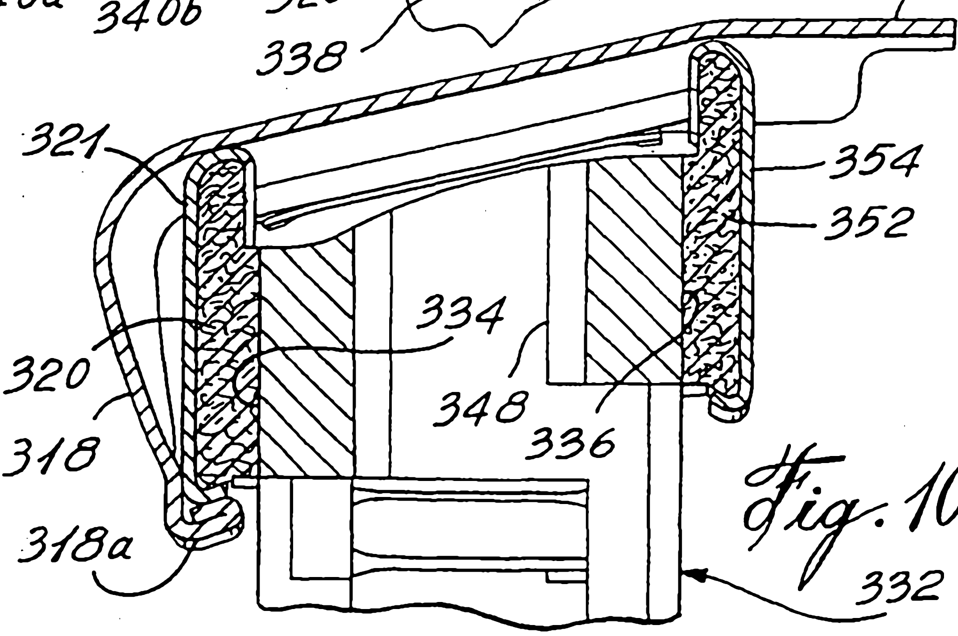
340a
340b
338

Fig. 9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334
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318
318a
332

Fig. 10