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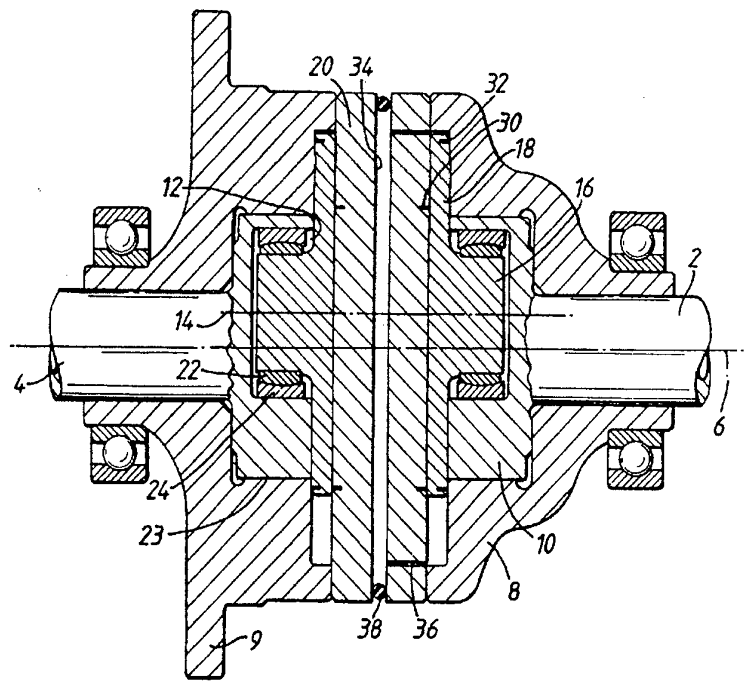
(54) Title: DIFFERENTIAL DRIVE MECHANISMS

(57) Abstract

A differential drive mechanism comprises a cage (8), which is rotatable about an axis (6) and represents the input, two coaxial output shafts (2, 4) which are rotatable with respect to the cage (8) about the said axis, a coupling (16) which is connected eccentrically to the two output shafts (2, 4) to transmit relative contrarotational movement between them by connections (12, 22, 24) which permit relative rotation of the coupling (16) and the output shafts about an axis substantially parallel to the said axis and a restraint member (20) which is coupled to the cage (8) and to the coupling (16) such that the coupling (16) is rotatable with respect to the cage (8) about an axis substantially perpendicular to the said axis and is capable of reciprocating movement in a direction perpendicular to the said axis but prevented from movement in a direction parallel to the said axis (6). The eccentric connection of the coupling (16) and the output shafts (2, 4) includes a respective eccentric hole (12) formed in the inner end (10) of each output shaft (2, 4) in which the associated end of the coupling (16) is received, the ends of the coupling (16) having a part-spherical engagement surface. One or more portions of the coupling (16) or of a member connected to it (18; 2, 4; 20) constitute a piston which is slidably received in the respective cylinder defined by a further component or components (8, 20; 8) of the differential drive mechanism. The said portion(s) of the coupling (16) or of the said member (18; 2, 4; 20) or one of the said further components (8, 20; 8) defines a restricted passage (36; 42; 44; 58) which connects the cylinder to a further space. When differential rotation of the output shafts (2, 4) occurs a fluid, such as oil, within the cylinder is cyclically discharged into and withdrawn from the further space through the restricted passage.



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DIFFERENTIAL DRIVE MECHANISMS

The present invention relates to differential drive mechanisms and is concerned with such mechanisms of the type which comprise a cage which is rotatable about an axis and represents the input, two coaxial output shafts which are rotatable with respect to the cage about the said axis, a coupling which is connected eccentrically to the two output shafts to transmit relative contra-rotational movement between them by connections which permit relative rotation of the coupling and the output shafts about an axis substantially parallel to the said axis and a restraint member which is coupled to the cage and to the coupling such that the coupling is rotatable with respect to the cage about an axis substantially perpendicular to the said axis and capable of reciprocating movement in a direction perpendicular to the said axis but prevented from movement in a direction parallel to the said axis, the eccentric connection of the coupling and the output shafts including a respective eccentric hole formed in the inner end of each output shaft in which the associated end of the coupling is received, the ends of the coupling having a part-spherical engagement surface. Such a differential is disclosed in German Patent No. 819628.

An inherent characteristic of such differentials is that if the resistance to rotation of one of the output shafts should decrease substantially, e.g. if the wheel attached to that output shaft is in contact with mud or ice, that output shaft commences to rotate very much faster than the other output shaft. This results in a transfer of torque from the one output shaft to the other but only to a limited extent which is governed by the torque bias ratio of the differential which is the maximum possible ratio of the torque applied to the two output shafts. In normal operation the torque bias ratio desirably has a value of say 2 to 3 but in the event that one output shaft is rotating very much more rapidly than the other, that is to say the differential is differentiating at high speed, it is desirable that the torque bias ratio should have as high a value as possible so that the maximum amount of torque is transmitted to the more slowly rotating output shaft. This is desirable so as to ensure that power is not expended spinning a vehicle wheel and that the maximum amount of power is available to the other wheel to drive the vehicle over the patch of mud or the like. It is, therefore, desirable that the torque bias ratio is strongly speed dependent, that is to say increases substantially with increased speed of differentiation.

Differentials of the type referred to above inherently have little or no speed sensitivity of the torque bias ratio due to the fact that the frictional torque is substantially independent of speed. A speed dependent torque transfer is desirable and it is therefore an object of the present invention to provide a differential of the type referred to above which has a substantial speed sensitivity of the torque bias ratio.

According to the present invention a differential drive mechanism of the type referred to above is characterised in that one or more portions of the coupling or of a member connected to it constitute a piston which is slidably received in a respective cylinder defined by a further component or components of the differential drive mechanism, that the said portion(s) of the coupling or of the said member or one of the said further components defines a restricted passage which connects the cylinder to a further space, whereby when differential rotation of the output shafts occurs a fluid within the cylinder is cyclically discharged into and withdrawn from the said further space through the restricted passage.

The present invention is based on the recognition that when a differential of the type referred to above differentiates the coupling or a member connected to it, which may be the restraint member, reciprocates linearly with respect to the cage perpendicular to the said axis and that the ends of the coupling or members connected to it, which may be the output shafts, reciprocate with respect to the cage parallel to the said axis and that one or more of the said reciprocating members may form one half of a piston/cylinder unit and that, if a restricted passage is formed in either the piston or cylinder of the unit, fluid contained within the cylinder will be pumped through the passage which will generate the pressure which resists differentiation of the differential. The magnitude of this pressure increases with increasing speed of differentiation, and in fact increases with the square of the velocity of the oil through the restricted passage, whereby the torque transferred across the differential during differentiation will increase substantially with increasing differential speed. Such speed dependent torque transfer across the differential will result in the torque bias ratio increasing substantially with increasing speed.

The fluid which is pumped back and forth by the piston/

cylinder unit(s) may be air but it is preferred that it is a liquid, such as oil, and in practice it may be an oil/air mixture. The further space into which the fluid is pumped may be outside the differential drive mechanism and may constitute, for instance, simply an oil sump. However, it is preferred that the further space is defined within the differential drive mechanism and that its volume increases as that of the associated cylinder decreases and vice versa. Whilst there are several spaces within differential drive mechanisms of the type referred to above whose volume cyclically increases and decreases as differentiation occurs, that is to say as differential rotation of the output shafts occurs, it is preferred that there are two pistons received within associated cylinders which are connected together by the restricted passage, whereby each cylinder constitutes the said further space for the other cylinder. Thus in this construction the oil or other fluid is simply pumped back and forth between the two cylinders, the combined volume of which remains constant.

In one embodiment the restraint member is fixed relative to the cage, the coupling member is mounted to reciprocate in rotation about the restraint member and linearly parallel to the length of the restraint member, the coupling member including a tubular portion which extends around the restraint member and whose ends constitute annular pistons received in respective cylinders defined by the cage and the restraint member, the restricted passage constituting a bore within the restraint member extending substantially parallel to its length and two further bores which extend transverse to its length and connect the bore to respective cylinders.

In a further embodiment the coupling member is connected to move with the restraint member, the two ends of which constitute pistons within respective cylinders defined by the cage, the restricted passage comprising a bore within the coupling member which connects the two cylinders together.

In a further preferred embodiment in which the fluid is not simply pumped back and forth between two cylinders, the ends of the coupling are restrained within the eccentric holes so as to be fixed parallel to the said axis, but nevertheless still rotatable about the axis of the holes, whereby when differential rotation of the output shafts occurs they reciprocate longitudinally, whereby the inner ends of the output shafts constitute pistons within cylinders defined by the cage. The restricted passage may then be formed through each inner end. Alternatively, a space may be provided within each output shaft and the restricted passage is formed in each output shaft which connects the associated space to the associated cylinder. In the latter case it will be appreciated that there is an inherent tendency for the total volume of the space occupied by the oil or other fluid to vary as the piston reciprocates but this may be compensated for in the event that the space is filled with oil by the provision of a piston which is in sliding sealing engagement with the wall of the space and is biased into engagement with the oil by a spring. The piston thus moves automatically in response to changes in volume of the oil space.

It will be appreciated that the piston(s) inherently move with a generally sinusoidally varying speed in the associated cylinder(s). This tends to result in the frictional torque and thus the torque bias ratio varying cyclically over each differential revolution. In a further preferred embodiment this cyclical variation is substantially smoothed by configuring the components so that the resistive force exerted by the oil and thus the torque bias ratio varies in a manner similar to that of a square wave rather than a sinusoidal wave. Since the torque bias ratio is dependent in part on the instantaneous value of the pressure generated in the fluid when differential rotation of the output shafts occurs, this may be effected by providing a variable restriction in or associated with the restricted passage which automatically varies the flow area of the restricted passage in inverse relation to the velocity of the fluid whereby, when the piston is moving the resistive force exerted by the oil is smoothed. This variable restriction may have a variety of forms but in one embodiment of the invention one end of the restricted passage is opposed to the surface of a member which moves relative to the restricted passage when differential rotation of the output shaft occurs, the said surface of the member having a profile which is so shaped that the spacing between the end of the restricted passage and said surface varies cyclically, whereby the resistance to flow presented by the restricted passage varies cyclically also. Alternatively, the restricted passage may include a variable restrictor valve member which is biased into contact with the surface of a member which moves relative to the restricted passage when differential rotation of the output shafts occurs, the said surface of the member having a profile which is so shaped that the position of the variable restrictor valve member moves relative to the restricted passage cyclically, whereby the resistance to flow presented by the restricted passage varies cyclically also.

Further features and details of the invention will be apparent from the following description of certain specific embodiments which is given by way of example with reference to the accompanying drawings, in which:

Figure 1 is a sectional view of the first embodiment of the invention;

Figure 2 is a similar view of the second embodiment;

Figure 3 is a similar view of the third embodiment;

Figure 4 is a sectional view at right angles to that of Figures 1 to 3 which shows the fourth embodiment on the right hand and a modification thereof on the left hand side;

Figure 5 is a scrap sectional view on the line Z-Z in Figure 4;

Figure 6 is a sectional view similar to Figure 3 of the fifth embodiment;

Figure 7 is an end view of the fifth embodiment from the left in Figure 6 with the portion on the line Z-Z in section;

Figure 8 is a view similar to Figure 1 of the sixth embodiment;

Figure 9 is a sectional view on the line IX-IX in Figure 8; and

Figure 10 is a scrap sectional view on the line X-X in Figure 9.

The differential of Figure 1 includes two output shafts 2 and 4, which are rotatable about a common axis 6 and pass through, and are rotatable with respect to, a cage 8 which is also rotatable about the axis 6. The cage has an end cover 9 which is the final drive flange through which rotational movement is transmitted to the cage. At their inner ends the output shafts have a thickened portion 10 in which a cylindrical hole or recess 12 is formed, the axis 14 of which is parallel to but offset from the axis 6. Received in the holes 12 are the ends of a coupling bar 16 at whose centre there is a tubular portion 18 defining a hole whose axis is perpendicular to the axis 6. Slidably and rotatably received in this hole is a restraint bar 20.

The ends of the restraint bar 20 are fixedly secured and sealed in respective holes in opposed sides of the cage 8. At its ends the coupling bar 16 carries part-spherical segments 22 whose surface engages the complementary internal surface of a respective sleeve 24 which is slidably received in the associated eccentric hole 12.

The external surface of each thickened portion 10 is of circular shape and is spaced by a clearance from the opposed circular portion of the internal surface of the cage, which is less than the clearance between the output shafts 2 at the point at which they pass through the cage, so that contact may not occur at that point. A sliding interface 23 is thus defined between each thickened portion and the cage.

The tubular portion 18 of the coupling 16 is a close sliding fit between the outer surface of the restraint bar 20 and an opposed circular section portion of the inner surface of the cage 8 and seals 30 and 32 on the outer surface of the restraint bar 20 and the tubular portion 18 ensure a seal between the opposed sliding surfaces. Each end of the tubular portion 18 might thus be thought of as constituting an annular piston within an annular cylinder defined by the restraint 20 and the cage 8. Provided within the restraint bar is an axial bore 34 which communicates with the interior of each cylinder by way of a respective small radial bore 36. The outer ends of the bore 34 are closed by respective seals 38.

In use, when a vehicle, to which the differential is fitted, travels in a straight line, the cage is rotated about the axis 6 and thus rotational movement is transmitted through the restraint and coupling bars 20, 16 to the output shafts 2, 4 and all the illustrated components rotate at the same speed and do not move either linearly or in rotation with respect to one another. A substantial proportion of the propulsive torque of the engine is transmitted from the cage 8 to the restraint bar 20 and thence to the coupling bar 16. This load displaces the output shafts by a small distance perpendicular to the axis 6 and thus presses one side of the outer surface of the thickened portion of the inner surface of the cage 8 with a force which increases with the propulsive torque. This contact effectively constitutes a force-locking connection between the cage and the output shafts and a significant proportion of the propulsive torque is thus transmitted directly from the cage to the output shafts and bypasses the restraint and coupling bars which can thus be of lighter construction than usual and are subject to a reduced tendency to failure. If the vehicle should turn a corner, one of the output shafts rotates at a slower speed and thus contra-rotates with respect to the other output shaft. This movement causes linear movement of the coupling bar 16 along the length of the restraint bar 20 and rotational movement about it and this movement is transmitted to the other output shaft as a corresponding increase in rotational speed by the reciprocating rocking motion of the coupling bar. The coupling bar thus reciprocates linearly in the plane of Figure 1 along the restraint bar 20 and oscillates in rotation about the restraint bar 20 at a rate determined by the differential speed of the two output shafts. The oscillatory movement of the coupling bar results in its ends and the sleeves 24 reciprocating back and forth in the eccentric holes 12. The force transmitted from the coupling bar to the thickened portions 10 is transmitted over the relatively large area of the external surface of the sleeves 24 and thus no excessively large contact loads are produced whereby wear of the cooperating surfaces of the sleeves 24 and thickened portions 10 is minimised. The contra-rotation of the output shafts results in the areas of contact between the thickened portions 10 and the internal surface of the cage contra-rotating also. This generates a frictional torque where both the normal force and the coefficient of friction are load dependent, which means that the torque bias ratio of the differential increases with increasing load.

As the coupling bar 16 reciprocates linearly along the length of the restraint bar 20 the pistons constituted by the opposed ends of the tubular portion 18 reciprocate into and out of the cylinders defined by the restraint bar and cage. The available space within the cylinders, that is to say the space not occupied by the pistons, and the bores 34, 36 connecting them are occupied by a fluid. This fluid may be air but in the present case it is oil or some similar liquid due to its higher density and viscosity. As one of the pistons moves into the associated cylinder it displaces the oil therein through the bores 34, 36 into the other cylinder whose volume is of course increasing at the same rate. The pressure generated in the oil by the pumping action is of a magnitude which is related to the speed of reciprocation and thus to the differential rotational speed of the output shafts 4, 6. Accordingly, as the differential speed increases, the oil pressure and thus the resistance to differentiation increases also and this results in a substantial increase in the torque bias ratio and thus transfer of torque from the output shaft which is rotating faster to that which is rotating slower.

The embodiment of Figure 2 is generally similar but the bores 34, 36 have been omitted and the tubular portion 18 of the coupling 16 is shorter and does not form a sliding seal with opposed surfaces of the cage 8 or the inner ends 10 of the output shafts. Furthermore, the ends of the coupling bar 16 are not free to reciprocate within the eccentric holes 12 because the sleeves 24, whilst still free to rotate within the holes 12, are fixed in position, in this case by respective circlips 40. Nevertheless, linear reciprocation of the ends of the coupling bar 16 still occurs but in this case this is transmitted to the output shafts 4, 6. This embodiment is therefore suitable for applications, e.g. in a front wheel drive car, in which the outer ends of the output shafts are connected to universal joints or the like (not shown) which can readily accommodate cyclical longitudinal movement.

Thus when differential rotation of the output shafts occurs, their inner ends 10 constitute pistons which reciprocate within respective cylinders constituted by respective portions of the cage 8. A bore 42 parallel to the axis 6 passes through the inner end 10 of each output shaft on that side of the axis 6 which is remote from the axis 14. Seals 44, 46 on the inner ends 10 of the output shafts 2, 4 and on their smaller diameter portions ensure a reliable seal at the sliding interfaces 23 and between the shaft 2, 4 and the cage 8. As before, the interior of the cage is substantially sealed and occupied by a fluid, which is again preferably oil. Thus as the inner ends 10 reciprocate the oil is transferred or pumped cyclically through the bores 42 from one side of the inner ends 10 to the other, that is to say between the spaces in front of the inner ends 10 bounded by the output shafts, the cage, the coupling bar and the restraint bar and the spaces behind the inner ends 10 bounded by the output shafts and the cage. The sum of the volume of these spaces is of course always constant. The effect of this is the same as in the previous embodiment.

The embodiment of Figure 3 is again generally similar to that of Figure 1 but the coupling bar 16 is integral with the restraint bar 20 and the tubular portion 18 is thus not present. The cage 8 extends over the ends of the restraint bar 20 which are provided with annular seals 48 and are accommodated as a sealed sliding fit within complementarily shaped cylindrical recesses 50 defined by the cage. The restraint bar 20 is again provided with an axial bore 34 but the bores 36 are omitted. The bore 34 is relatively large but accommodates a restrictor or throttle 52. The available volume of the cylinders 50 and the bore 34 are occupied by oil.

When differential rotation of the output shafts occurs, the coupling bar 16 again reciprocates linearly parallel to the length of the restraint member 20. The latter thus reciprocates linearly also (and indeed in rotation) due to the fact that it is integral with it. The ends of the restraint bar 20 thus constitute pistons which reciprocate within their respective cylinders 50. The oil in the cylinders is thus cyclically pumped through the bore 34 between the two cylinders. The effect of this is the same as in the previous embodiments.

The embodiment shown in Figure 4 is similar to that of Figure 2 in that the sleeves 24 are again fixed in position and the output shafts therefore reciprocate longitudinally and the inner ends of the output shafts constitute pistons which reciprocate within cylinders defined by respective portions of the cage 8. However, each output shaft has a respective longitudinal hollow or bore 54 formed in it and the space which is occupied by the oil is constituted by the bore 54 and the space 56 behind the associated inner end 10 which is defined by the inner end 10, the associated output shaft and the cage. The space 56 is connected to the bore 54 by a radial bore 58 in the wall of the output shaft. Situated within the bore 54 is a piston 60 whose outer surface is engaged by one end of a compression spring 62 whose other end is retained fixed by a circlip 64. The spring urges the piston 60 inwardly against the oil so that the spaces 54 and 56 are maintained full of oil at all times.

When differential rotation of the output shafts occurs the inner ends 10 reciprocate within their cylinders thus resulting in a cyclical variation in the volume of the spaces 56. This results in oil being cyclically pumped into the space 54 and withdrawn from it. The effect of this is the same as in the preceding embodiments.

The above description relates to the construction illustrated on both sides of Figure 4 but on the left hand side of that Figure and also in Figure 5 a modification is shown whose purpose is to smooth the cyclical variation of the torque bias ratio described above as the cage rotates relative to the output shafts. In the construction shown on the right of Figure 4 the outer end of the bore 58 communicates directly with the space 56 but in the construction shown on the left it is directed towards the wall of the bore in which the shaft 4 is received. This wall has a profile 59 as shown in Figure 5 so that it approaches closer to the outer end of the bore 58 at four positions spaced apart by 90° than at all other positions. At these four positions the resistance to oil flow is increased and these increases are timed to occur at the four periods of minimum velocity of the piston (and in fact occur each time the linear or rotational speed of reciprocation of the coupling with respect to the cage is zero) and are dimensioned so that the variation in pressure of the oil within the cylinder is not generally sinusoidal but has a form approximating to that of a square wave, whereby variations in the oil pressure and thus in the torque bias ratio as the piston moves are reduced or smoothed.

The fifth embodiment shown in Figures 6 and 7 is generally similar to that of Figure 1 except that the restrictor 52 has been omitted and the lower end of the bore 34 is closed by a seal 38 and the bore communicates at its lower end with the cylinder 50 by way of a restricted passage 60 which extends at an acute angle to the radial direction. Situated in the wall of the cylinder are two vertical elongate, circumferentially spaced grooves or recesses 62 which are generally opposed to the outer end of the restricted passage 60. The recesses 62 constitutes a profiling of the surface of the lower cylinder 50 whose purpose is similar to that of the profiling 59 in Figure 5. Thus as differentiation occurs the restraint bar 20 reciprocates longitudinally and in rotation and as it does so oil is pumped between the two cylinders via the bores 34 and 60. The recesses 62 are dimensioned and positioned so that as the downstream end of the passage 60 moves linearly and in rotation it is opposed to one or other of the recesses 62 for most of each cycle but is partially obstructed by the cylinder wall at four separate times spaced apart by 90°. These times correspond to the periods of minimum speed of the pistons. When the passage 62 is partially obstructed in this manner the resistance to oil flow through it is temporarily increased which leads to an increase in the torque bias ratio in the same manner and for the same purpose as that in Figure 5.

Figures 8 to 10 show a further embodiment which is a modification of that shown in Figure 4 but also has the additional advantage of that shown in Figure 5 and also in Figures 6 and 7. In this case the restricted passage 58 cooperates with a variable restrictor valve member 64 which is slidably accommodated in a bore 59 and in the peripheral surface of which an annular groove 68 is formed. The variable restrictor valve body 64 is biased by a spring 66 into contact with the end surface of the coupling 16 which is profiled so that the distance by which the valve body 64 projects out of the passage 58 varies cyclically and reaches four extreme values, i.e. maxima or minima per differential revolution. The flow resistance presented by the valve body 64 is at a minimum when the annular groove 68 is in line with the restricted passage 58. The minima in the flow resistance are timed to occur at the same times as in the preceding embodiments.

It will be appreciated that numerous modifications and further constructions are possible. Thus in the embodiments described above the pistons form a substantial seal with the surface of the associated cylinder by virtue of their relative dimensions and preferably also the provision of one or more seals or piston rings. However, this is not essential and in a modified construction, which is not illustrated, the restricted passage is connected and sealed at each end to the interior of a respective flexible bag or bellows of the type used in an aneroid within which the oil is contained and between which the oil is pumped through the restricted passage as the piston reciprocates. Since the oil is contained within a separately sealed space, a seal between the pistons and the associated cylinders is no longer necessary.

It will be appreciated also that many of the features of the above embodiments may be transferred to different embodiments and that the features of certain pairs of the above embodiments may be combined.

CLAIMS

1. A differential drive mechanism comprising a cage (8), which is rotatable about an axis (6) and represents the input, two coaxial output shafts (2, 4) which are rotatable with respect to the cage (8) about the said axis (6), a coupling (16) which is connected eccentrically to the two output shafts (2, 4) to transmit relative contra-rotational movement between them by connections (12, 22, 24) which permit relative rotation of the coupling (16) and the output shafts (2, 4) about an axis (14) substantially parallel to the said axis (6) and a restraint member (20) which is coupled to the cage (8) and to the coupling (16) such that the coupling (16) is rotatable with respect to the cage (8) about an axis substantially perpendicular to the said axis (6) and capable of reciprocating movement in a direction perpendicular to the said axis (6) but prevented from movement in a direction parallel to the said axis (6), the eccentric connection of the coupling (16) and the output shafts (2, 4) including a respective eccentric hole (12) formed in the inner end (10) of each output shaft (2, 4) in which the associated end of the coupling (16) is received, the ends of the coupling (16) having a part-spherical engagement surface, characterised in that one or more portions of the coupling (16) or of a member connected to it (18; 2, 4; 20) constitute a piston which is slidably received in a respective cylinder defined by a further component or components (8, 20; 8) of the differential drive mechanism, that the said portion(s) of the coupling (16) or of the said member (18; 2, 4; 20) or one of the said further components (8, 20; 8) defines a restricted passage (36; 42; 34; 58) which connects the

cylinder to a further space whereby when differential rotation of the output shafts (2, 4) occurs a fluid within the cylinder is cyclically discharged into and withdrawn from the said further space through the restricted passage.

2. A mechanism as claimed in Claim 1 characterised in that the said further space is defined within the differential drive mechanism and its volume increases as that of the associated cylinder decreases and vice versa.

3. A mechanism as claimed in Claim 1 or Claim 2 characterised in that there are two pistons received within associated cylinders which are connected together by the restricted passage (34, 36; 34; 34, 60), whereby each cylinder constitutes the said further space for the other cylinder.

4. A mechanism as claimed in Claim 3 characterised in that the restraint member (20) is fixed relative to the cage (8), that the coupling member (16) is mounted to reciprocate in rotation about the restraint member and linearly parallel to the length of the restraint member, that the coupling member (16) includes a tubular portion (18) which extends around the restraint member (20) and whose ends constitute annular pistons received in respective cylinders defined by the cage (8) and the restraint member (20) and that the restricted passage constitutes a bore (34) within the restraint member extending substantially parallel to its length and two further bores (36) which extend transverse to its length and connect the bore (34) to respective cylinders.

5. A mechanism as claimed in Claim 3 characterised in

that the coupling member (16) is connected to move with the restraint member (20), the two ends of which constitute pistons within respective cylinders (50) defined by the cage (8) and that the restricted passage comprises a bore (34) within the coupling member (20) which connects the two cylinders (50) together.

6. A mechanism as claimed in Claim 2 characterised in that the ends of the coupling (16) are restrained within the eccentric holes (12) so as to be fixed parallel to the said axis (6) whereby when differential rotation of the output shafts (2, 4) occurs they reciprocate longitudinally whereby the inner ends (10) of the output shafts (2, 4) constitute pistons within cylinders defined by the cage (8) and that a restricted passage (42) is formed through each inner end (10).

7. A mechanism as claimed in Claim 2 characterised in that the ends of the coupling (16) are restrained within the eccentric holes (12) so as to be fixed parallel to the said axis (6) whereby when differential rotation of the output shafts (2, 4) occurs they reciprocate longitudinally whereby the inner ends (10) of the output shafts (2, 4) constitute pistons within cylinders defined by the cage (8), that a space (54) is provided within each output shaft (2, 4) and that a restricted passage (66) in each output shaft (2, 4) connects the associated space (54) to the associated cylinder.

8. A mechanism as claimed in Claim 7 characterised in that the space (54) is filled with oil which is engaged by a piston (60) which is in sliding sealing engagement with the wall of the space (54) and is biased into engagement with the oil by a spring (62).

9. A mechanism as claimed in any one of the preceding claims characterised in that one end of the restricted passage (60; 58) is opposed to the surface of a member (4; 8) which moves relative to the restricted passage when differential rotation of the output shafts (2, 4) occurs, the said surface of the member (4; 8) having a profile (59; 62) which is so shaped that the spacing between the end of the restricted passage (60; 58) and the said surface varies cyclically, whereby the resistance to flow presented by the restricted passage (58) varies cyclically also.

10. A mechanism as claimed in any one of the preceding claims characterised in that the restricted passage (58) includes a variable restrictor valve member (64) which is biased into contact with the surface of a member (16) which moves relative to the restricted passage when differential rotation of the output shafts (2, 4) occurs, the said surface of the member (16) having a profile which is so shaped that the position of the variable restrictor valve member (64) moves relative to the restricted passage (58) cyclically, whereby the resistance to flow presented by the restricted passage (58) varies cyclically also.

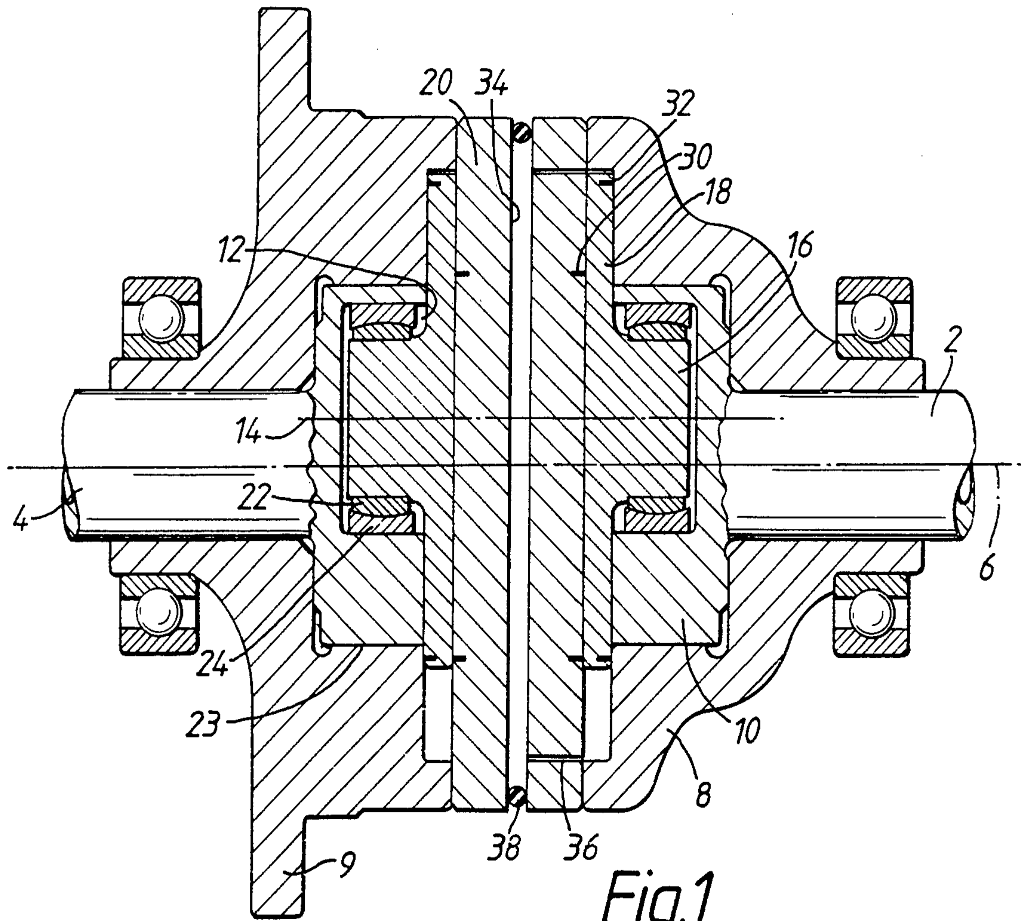
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Fig.1

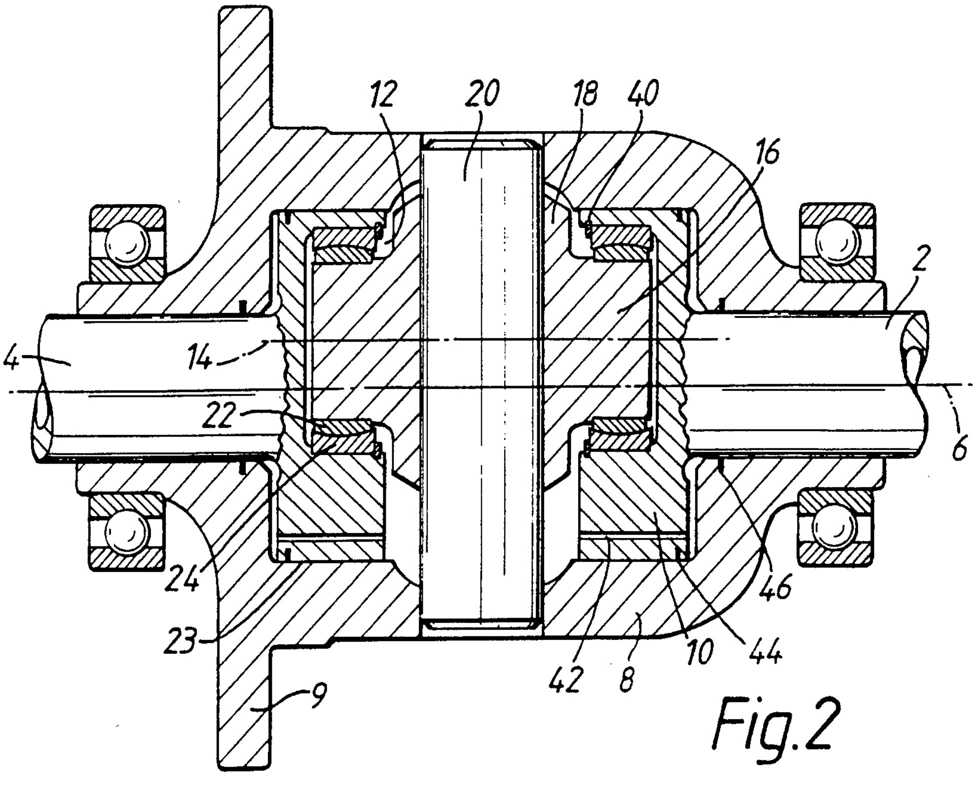
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Fig.2

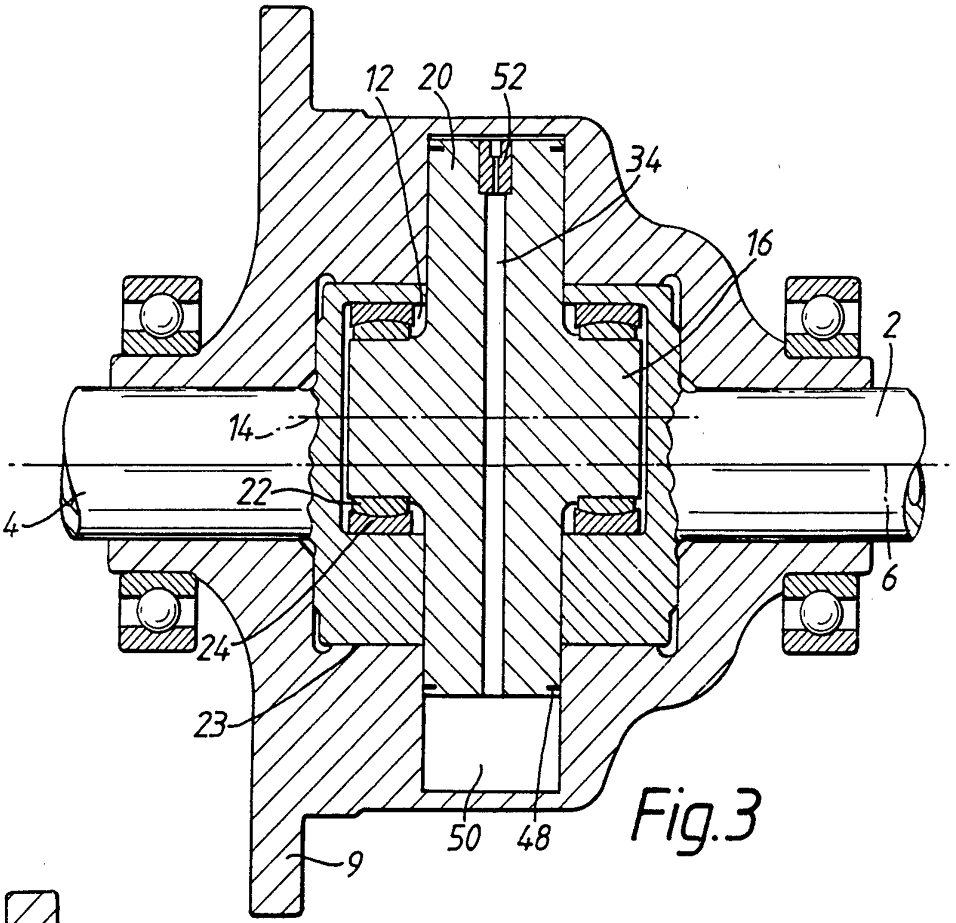

Fig.3

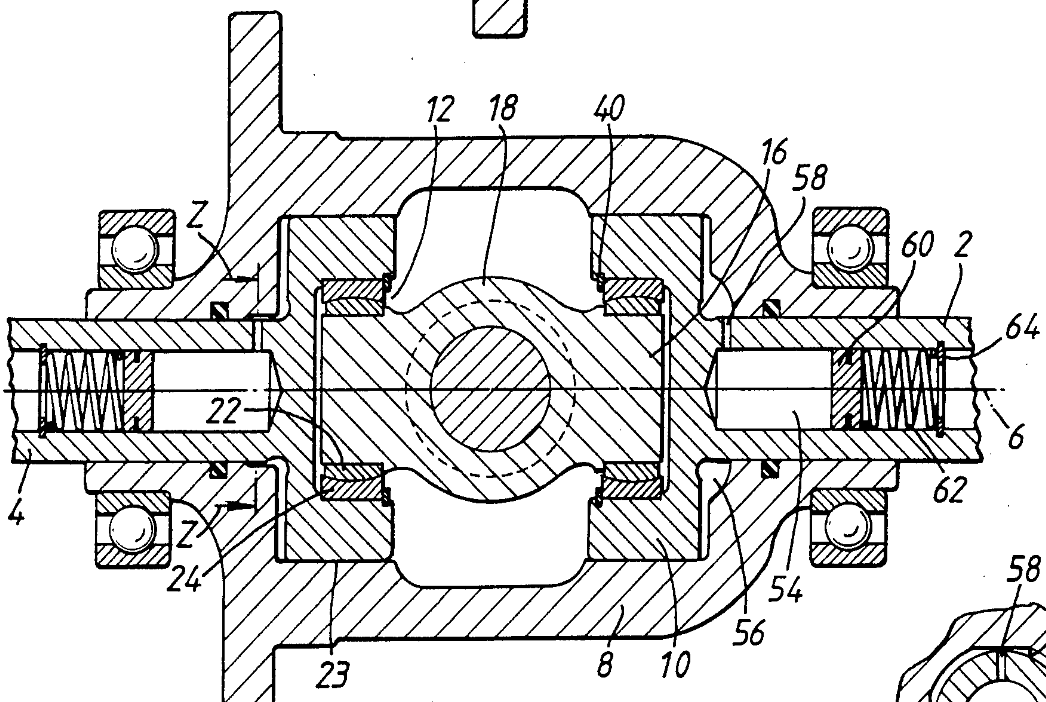

Fig.4

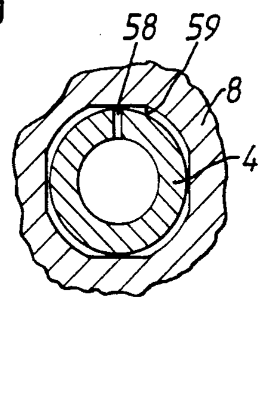

Fig.5

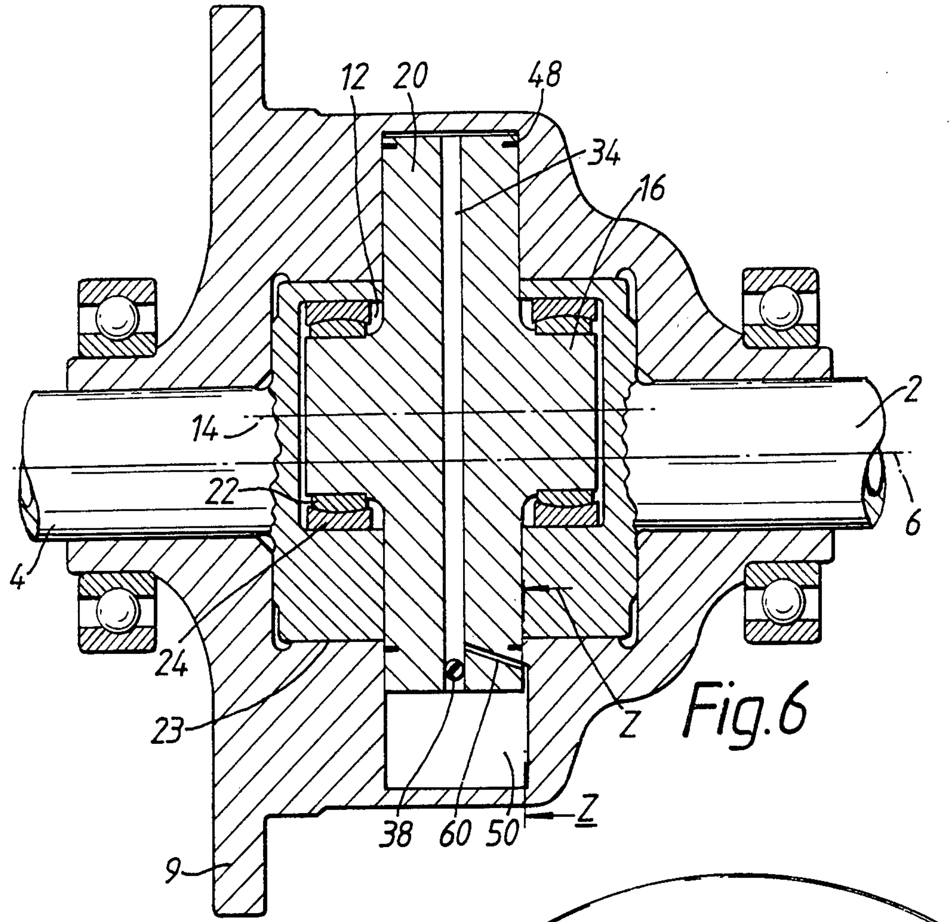

Fig.6

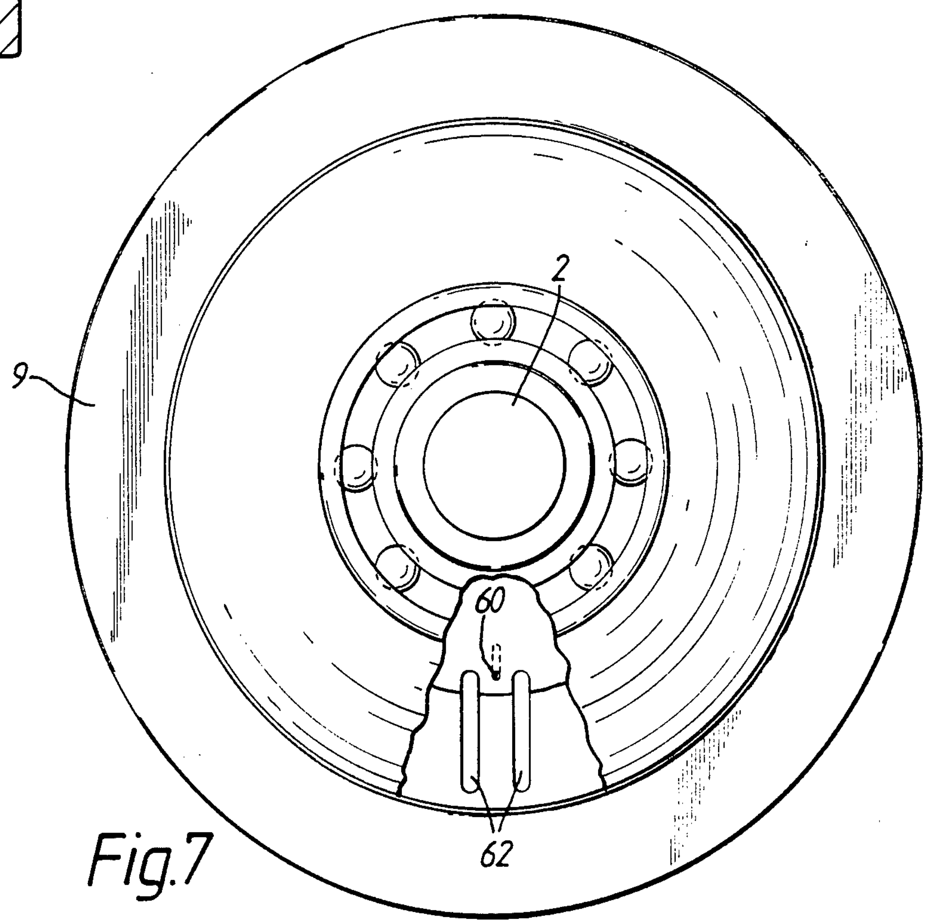

Fig.7

4/4

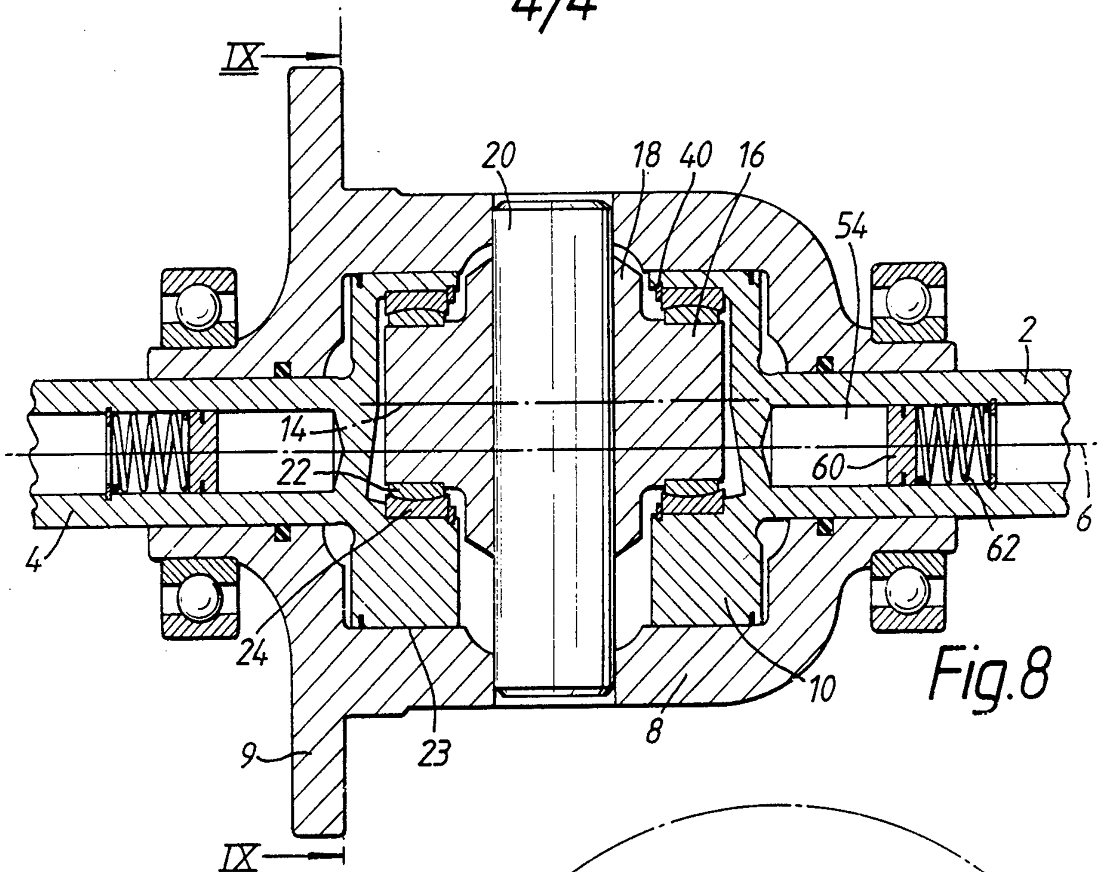

Fig.8

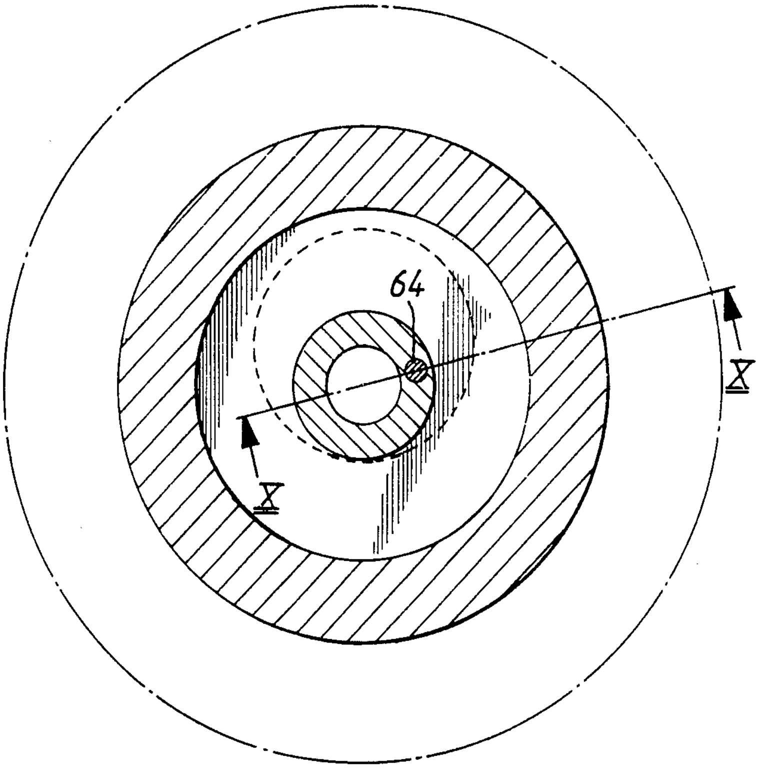

Fig.9

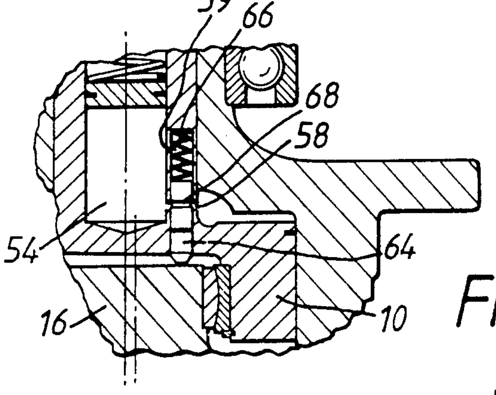

Fig.10

INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 94/00273

A. CLASSIFICATION OF SUBJECT MATTER
IPC 5 F16H35/04

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)
IPC 5 F16H

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	FR,A,1 181 029 (CHAUDE) 11 June 1959 see the whole document ---	1,2
A	DE,C,819 628 (ALTMANN) 13 September 1951 cited in the application see the whole document -----	1

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

☒ Patent family members are listed in annex.

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"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

1

Date of the actual completion of the international search	Date of mailing of the international search report
21 April 1994	27. 04. 94

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+ 31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+ 31-70) 340-3016	Mende, H

INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 94/00273

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
FR-A-1181029		NONE	
DE-C-819628		NONE	