(12) UK Patent Application (19) GB (11) 2 171 631 A

(43) Application published 3 Sep 1986

(21) Application No **8601993**

(22) Date of filing **28 Jan 1986**

(30) Priority data

(31) **3506695** (32) **26 Feb 1985** (33) **DE**

(71) Applicant
Robert Bosch GmbH (FR Germany),
Postfach 50, 7000 Stuttgart 1, Federal Republic of Germany

(72) Inventors
Manfred Bleicher,
Ulrich Bohne,
Karl Wanner

(74) Agent and/or Address for Service
A. A. Thornton & Co., Northumberland House, 303/306
High Holborn, London WC1V 7LE

(51) INT CL[4]
B25D 11/12 B23B 45/16

(52) Domestic classification (Edition H):
B4C 14E 1A1 1B4A 1C 1D1 1F 6B1 7 9BX 9C X
F2L 5M5 9BU
U1S 1647 B4C F2L

(56) Documents cited
None

(58) Field of search
B4C
F2L
Selected US specifications from IPC sub-class B25D

(54) **Hammer drill**

(57) A Hammer drill with a motor-driven air-cushion percussion mechanism 4 which is driven via a gear which contains a swash plate 26. The drum (16) of the plate (26) forms an axially displaceable moveable coupling part 39, 40 which, under the effect of the mechanism (4), can be pressed against the other coupling part (22, 41) supported fixedly in the housing (1). The impact reaction forces are transmitted via the coupling and supported by the fixed coupling part, with the result that the transmittable coupling torque is intensified as a result of the compressive force of the percussion mechanism. To separate the coupling, the drum (16) is shifted out of the coupling position via springs 47 and at the same time is braked via braking surfaces 62, 63. Thus, under no-load conditions, the swash plate is not taken up. This produces a perfect no-load behaviour (Figure 1).

The coupling part 39, 40 may be releasably coupled to the drum 16 or coupled via positive coupling members. Spur teeth arranged respectively on the drum 16 and the hub 22 may alternatively function as the coupling parts.

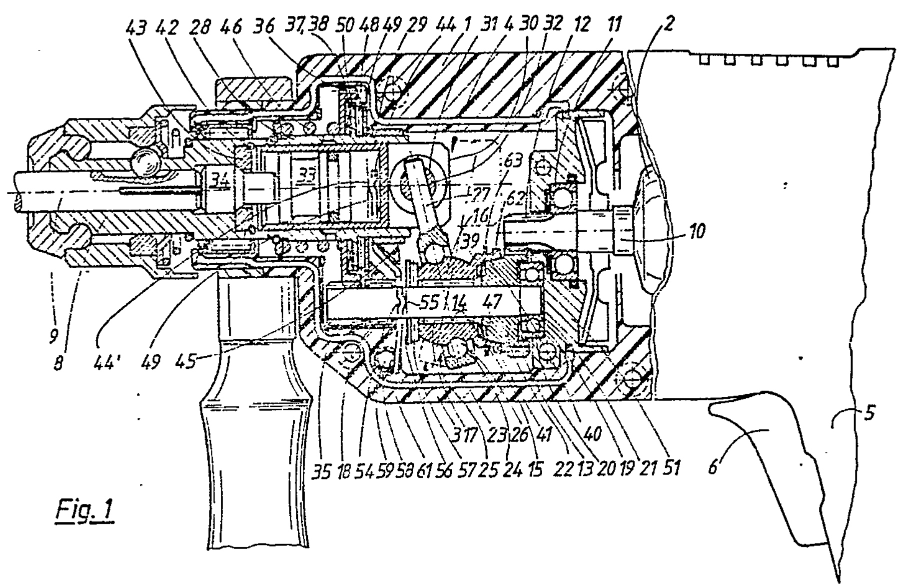

Fig. 1

GB 2 171 631 A

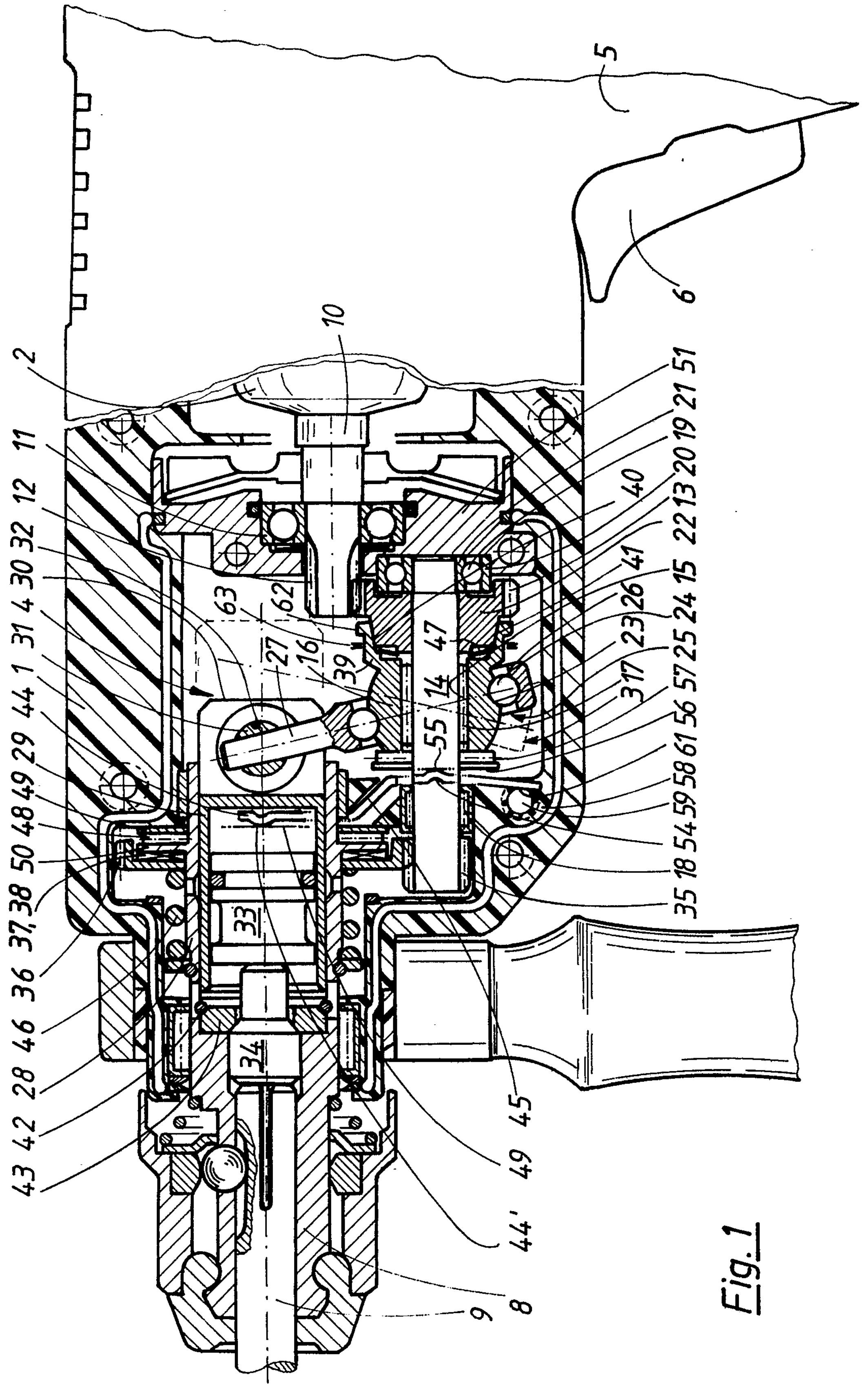
5
6
10
2
11
12
32
30
4
31
1
44
29
49
48
50
37,38
36
46
28
42
43
51
21
19
20
13
22
15
24
25
57
56
61
58
59
54
18
35
40
41
26
23
317
63
62
16
27
39
47
14
55
33
34
45
49
44'
8
9

Fig.1

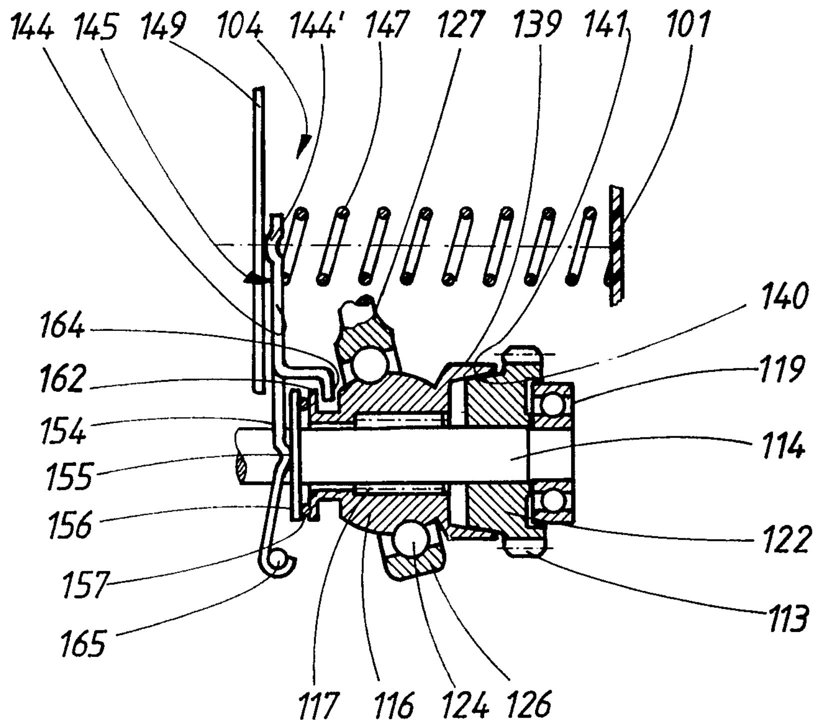
144
145
149
104
144'
147
127
139
141
101
140
119
114
122
113
164
162
154
155
156
157
165
117
116
124
126

Fig. 2

SPECIFICATION

Hammer drill

State of the art

The invention starts from a hammer drill according to the pre-characterising clause of the main claim. A hammer drill of this type is already known (German Patent Specification 2,449,191), and in it the drum of the swash plate is mounted on an intermediate shaft which is itself mounted in the housing of the hammer drill. In the gear train between the intermediate shaft and the drum of the swash plate there is a non-positively acting coupling which is normally disengaged by means of the spring, so that the drum together with the swash plate is not driven via the intermediate shaft, but which, when as a result of the pressure of the tool against the workpiece to be treated a corresponding reaction force takes effect, is brought in response to this reaction force, counter to the effect of the spring, into the closing position in which a drive torque is transmitted non-positively from the intermediate shaft to the drum with the swash plate and consequently to the percussion mechanism. The moveable coupling part consists, here, of a trunkated cone connected fixedly in terms of rotation to the intermediate shaft. The other fixed coupling part is formed by the drum with swash plate which is firmly supported axially relative to the housing. The effectiveness of the coupling between the motor and the drive member of the percussion mechanism depends on the pressing force exerted by the operator. When the force is sufficiently great, the intermediate shaft, as support for the movable coupling part, is pressed axially counter to the separating spring into an inner cone of the axially supported drum with swash plate forming the fixed coupling part, and the drum with swash plate and the percussion mechanism are driven as a result. This principle makes it more difficult to operate the hammer drill. Also, the reliable take-up of the drive member of the percussion mechanism depends on particular circumstances which vary individually and which cannot be reproduced, since, for example, not every user of the hammer drill can exert the same pressing force. The reaction forces which moreover have to be absorbed by the drum with swash plate and, because this is supported fixedly relative to the housing, also by the housing and which are generated by the percussion mechanism act to disengage the coupling. Precisely when the percussion mechanism has to exert the maximum compressive force at maximum compression and therefore the maximum torque has to be exerted via the coupling in the region of the drum with swash plate, the reaction force acting to disengage the coupling is also at it highest. This also means that the operator has to exert relatively high pressing forces.

Advantages of the invention

In contrast to this, the advantage of the hammer drill according to the invention, having the characterising features of the main claim, is that the pressing force necessary for the operator handling the hammer drill is extremely low during percussion drilling, but the closing force actually acting on the coupling is markedly higher than in known hammer drills. Precisely because the moveable engageable and disengageable coupling part is connected fixedly in terms of rotation to the swash plate and the other coupling part is supported fixedly in the housing and absorbs the reaction forces generated by the percussion mechanism, the recoil forces of the percussion mechanism which act on the swash plate result in an increase in the pressing force between the two halves of the coupling. Consequently, here, the non-positive connection between the two coupling parts is automatically reinforced by the compressive force of the percussion mechanism. Whenever the maximum force is required in the percussion mechanism for driving the piston, this arising in an air-cushion percussion mechanism when the pneumatic spring is pressed together to its greatest extent, because of the recoil effect the torque which can be transmitted by the coupling is also at its highest. The operator therefore finds the hammer drill substantially simpler to handle. The pressing forces to be exerted are substantially lower. The hammer drill overall is light-weight, inexpensive and compact and has a low degree of vibration.

Advantageous developments and improvements of the hammer drill indicated in the main claim are possible as a result of the measures described in the sub-claims. A particularly advantageous embodiment emerges from claim 12. By means of this separating spring, when the pressing force decreases, the moveable coupling part can be shifted in the direction of separation of the coupling. The dimensions of the separating spring are such that it can counteract sufficiently any restoring forces caused by the weight of individual parts of the percussion mechanism and of the gear, whilst at the same time keeping the coupling separated.

A further especially advantageous embodiment emerges from claim 15. This ensures that, in the coupling separation position, the drum with swash-plate is braked and cannot rotate. This results in a perfect no-load behaviour with reduction in any vibrations.

A further advantageous embodiment emerges from claim 16. The lever produces an increase in the pressing force which originates from the tool and which is transmitted to the moveable coupling part in the form of the drum with swash plate in the coupling-up direction. At the same time, the lever, likewise reinforced via its leverage, supports the drum with swash plate in the opposite direction and counter to the force which thereby separates the coupling.

Further advantageous embodiments emerge from claims 19 and 21. Here, the separating spring acts on the lever at a distance from the fulcrum, and via the engaging means the lever pulls off the drum with swash plate, carrying the moveable coupling part, in the coupling separation direction in a simple way and at the same time, by means of a clamping effect, ensures braking against the rotation of the drum. Here, the separating spring is located in the region of non-rotating parts, and this reduces the wear.

Drawing

Two exemplary embodiments of the invention are illustrated in the drawing and explained in more detail in the following description. In the drawing:

Figure 1 shows a diagrammatic and partially sectional side view of a hammer drill according to a first exemplary embodiment.

Figure 2 shows a diagrammatic and partially sectional side view of parts of the gear, with a movement convetor, of a hammer drill according to a second exemplary embodiment.

Description of the exemplary embodiments

The hammer drill illustrated in Figure 1 has a housing 1, in which an electric motor 2, a gear 3 and a percussion mechanism 4 are arranged. The axis 4' of the percussion mechanism 4 is parallel to the axis of the electric motor 2. The housing 1 at its rear end merges into a handle 5, in which is installed a switch which is provided with a trigger 6 and by means of which the electric motor 2 can be started. At the bottom end of the handle 5, a power feed cable is introduced through an elastic bush. At the front end facing away from the handle 5, a tool holder 8 serving for receiving tools, for example a drill bit 9, is arranged on the housing 1.

The electric motor 2 has a drive shaft 10 which is mounted on both sides in ball bearings fastened in the housing 1. The end of the drive shaft 10 mounted in a ball bearing 11 carries a motor pinion 12 which engages a gear wheel 13. The gear wheel 13 is held fixedly in terms of rotation on an intermediate shaft 14, for example pressed on to this. The intermediate shaft 14 is mounted by means of the end adjacent to the gear wheel 13 and on the right in Figure 1 in a ball bearing 19, the outer ring 20 of which is received fixedly in an intermediate-wall part 51, whilst the inner ring 21 rests on a stepped shoulder of the intermediate shaft 14. The intermediate shaft 14 passes through a relatively larger axial bore 15 in a drum 16 which rests on it. Arranged in the axial bore 15 is at least one needle cage 17, by means of which the drum 16 can rotate on the intermediate shaft 14 relative to the latter. The drum 16 can shift axially in the needle case 17 at least slightly in relation to the intermediate shaft 14. The end of the intermediate shaft 14 facing away from the ball bearing 19 is mounted in a housing 1 at a distance from the free shaft end by means of a needle bearing 18. The drum 16 has a track 23 with an axis oblique relative to the axis of the intermediate shaft 14. The track 23 forms the inner race for balls 24 which are parts of a ball bearing 25, the outer ring of which is designed as swash plate 26. A finger 27 formed onto the swash plate 26 drives the percussion mechanism 4.

The percussion mechanism 4 is arranged inside a guide tube 28 which is mounted rotatably in the housing 1 and which, in the exemplary embodiment illustrated, is made in one piece with the tool holder 8. A piston 29 serving as a drive member is guided in a sealed manner and so as to slide in the guide tube 28. The rear end 30 of the piston 29 facing away from the tool holder 8 is made fork-like and carries a pivot pin 31. This has a transverse bore 32, into which the finger 27 engages with play. The finger 27 can thereby move easily in the transverse bore 32 in the axial direction. Guided in a sealed manner and so as to slide in the hollow piston 29 is a striker 33 which acts via a header 34 on the rear shank end of the tool 9.

The intermediate shaft 14 carries, at the free end located opposite the ball bearing 19 and adjacent to the needle bearing 18, a pinion-like toothing 35 which engages into a gear wheel 36. The latter is mounted on the guide tube 28 so as to be displaceable and freely rotatable. It comes under the influence of a compression spring 46 which endeavours constantly to press the gear wheel 36 against a collar 50 of the guide tube 28. The end faces of the gear wheel 36 on the one hand and of the collar 50 on the other hand, which face one another, have projections 37, 38, which are designed so that, when they interact, they serve as a take-up coupling and, under the influence of the compression spring 46, as an overload coupling. The pressed-on gear wheel 13 is provided with a hub 22 which points axially towards the drum 16 and is preferably in one piece with the gear wheel 13, and which carries on its outer peripheral surface an outer cone 41 axially tapering frustoconically in the direction of the drum 16. An inner cone 40 is formed on the drum 16, at the end facing it, within a sleeve 39. The hub 22, together with the outer cone 41, on the one hand, and the drum 16 together with the inner cone 40 arranged inside the sleeve 3, on the other hand, constitutes an engageable and disengageable cone coupling, by means of which, when it is engaged, the drum 16 can be connected fixedly in terms of rotation to the gear wheel 13 and its hub 22 and consequently fixedly in terms of rotation to the intermediate shaft 14. The cone coupling 40/41 is engaged under the influence of the guide tube 28 which is shifted axially as a result of the pressure of the drill bit 9 against the workpiece to be treated.

The rear shank end of the drill bit 9 thereby presses against the header 34 and presses the latter against a thrust ring 43 retained on the guide tube 28 by means of a spring ring 42. The pressure of the shank end of the drill bit 9 on the header 34 causes the tool holder 8 together with its guide tube 28 to be shifted axially via the thrust ring 43 and the spring ring 42. The collar 50 of the guide tube 28 thereby presses, via an axial bearing 48 and a disc 49, against a fork-like end 44, engaging round the guide tube 28, of a lever 45. Figure 1 shows only one leg of the two fork legs of the fork-like end 44. The other end 54 located opposite the fork-like end 44 of the lever 45 is likewise made approximately fork-like and engages round the intermediate shaft 14. At the height of the axis of the intermediate shaft 14, the lever 45 has, on each of the two legs of the fork-like end 54, a projection 55 which is supported against the drum 16 via a disc 56 and an axial bearing 57. Arranged near a recess 58 in the housing 1 is an eccentric pin 59 which is rotatable about its bearing axis and which is thus adjustable. The eccentric pin 59 serves, when rotated, to adjust the lever 45. During the time when the projection 55 rests against the end face of the disc 56, the surface 61, facing away from the projection 55, of the lower fork-like end 54 of the lever 52 is supported against the periphery of the eccentric pin 59.

When the guide tube 28 and consequently the axial bearing 48 and the disc 49 are shifted as a result of the pressure of the tool 9, inserted in the hammer drill, against the workpiece to be treated, the end face of the disc 49 presses against a stamped-out projection 44′ on the particular leg of the fork-like end 44 of the lever 45, with the result that the latter, since its lower end 54 cannot move aside because its face 61 comes up against the periphery of the eccentric pin 59, is pivoted in the direction of the cone coupling 40/41. The projection 55 thereby presses against the end face of the disc 56, with the result that the drum 16, together with the inner cone 40 formed inside the sleeve 39, is pressed axially on to the outer cone 41 and consequently makes a coupling connection. The lever 52 is made of resiliently flexible material, so that, should the operator possibly increase the pressing force further, although a further shift of the guide tube 28 can occur, nevertheless the effective force acting on the cone coupling 40/41 does not exceed a predetermined value.

The effect of the pressing force exerted on the tool 9 on the transmittable torque of the cone coupling 40/41 is intensified. The intensified effect depends on the ratio of the distance between the projection 55 and the end 44 touching the disc 49 with the projection 44′ there. It is possible in a simple way as a result of adjustment of the eccentric pin 59 to compensate possible production tolerances.

As is evident from Figure 1, the moveable engageable and disengageable coupling part in the form of the sleeve 39 with the inner cone 40 is a fixed component of the swash plate 26 and, together with the latter, can, under the effect of the percussion mechanism 4, be pressed against the other coupling part associated with it, which is formed by the hub 22 with the outer cone 41 and gear wheel 13 which is located firmly on the intermediate shaft 14 and via its ball bearing 19 is supported firmly on the housing 1. Since the drum 16 is axially displaceable on the intermediate shaft 14 at least within limits, when the hammer drill is in operation the impact reaction forces which during hammering act on the piston 29 and from this on the swash plate 26 are absorbed, and under these impact reaction forces the drum 16 shifts axially towards the hub 22, specifically in such a way that the inner cone 40 is thereby pressed even more firmly on to the outer cone 41 and in this way the transmittable torque of the coupling designed as a friction coupling, specifically a cone coupling, is increased. Whenever the maximum force is required in the percussion mechanism 4 for driving the piston 29, this occurring when there is maximum compression between the piston 29 and striker 33, the transmittable torque of the coupling 40/41 is also at its highest. The compressive force generated in the percussion mechanism 4 is supported via the piston 29, the pivot pin 31, the finger 27, the swash plate 26, the balls 24 and the drum 16 with the sleeve 39 and inner cone 40 on the outer cone 41 of the hub 22 with the gear wheel 13 and via the ball bearing 19 on the housing 1. This support takes effect precisely when the maximum transmittable torque of the coupling and consequently the maximum drive torque are needed to ensure the slip-free rotary take-up of the drum 16 with swash plate 26. Consequently, the moveable coupling part 39, 40 as a component of the drum 16 can always be pressed in the opposite direction to the direction of the impact produced on the intermediate shaft 14, that is to say to the right in Figure 1, against the coupling part 21, 41 which is fixed there.

It goes without saying that, in another exemplary embodiment not shown, in a kinematic reversal the relative arrangements of the inner cone and outer cone can even be interchanged. Thus, the drum 16 can carry an outer cone, whilst the hub 22 has an inner cone receiving the latter.

Arranged between the two coupling parts, that is to say the sleeve 39 on the one hand and the hub 22 on the other hand, is a separating spring 47 which here is designed approximately like a cup spring and which is supported by means of its inner annular part on the hub 22 and by means of its outer annular part on a collar within the sleeve 39. The separating spring 47 engages here directly on the drum 16 carrying the moveable coupling part. It is designed as a compression spring which generates an axial pressure axis-parallel relative to the intermediate shaft 14 between the two coupling halves 40/41.

In the operating state according to Figure 1, the separating effect of the separating spring 47 is overcome and the coupling 40/41 is in the active position.

Furthermore, the drum 16 carrying the moveable coupling part in the form of the sleeve 39 with an inner cone 40 carries fixed braking surfaces formed on an annular collar 62 integral with the drum. Axially adjacent to the annular collar 62 is an axial stop 63 integral with the housing and having associated braking surfaces.

In the operating state illustrated in Figure 1, the lever 45 is stressed in the way described as a result of the pressing force of the tool 9 against the workpiece to be treated, and as a result the end 54 of the lever 45 shifts the drum 16 axially to the right in Figure 1 via the projection 55 and consequently presses the moveable coupling part with the inner cone 40 against the rotating, but axially non-displaceable and therefore fixed coupling part in the form of the hub 22 with the outer cone 41, so that the coupling is in the active position. The separating effect of the separating spring is thereby overcome.

When the pressing force of the tool 9 decreases, the separating spring 47 presses the drum 16 axially away from the hub 22 in the coupling separation direction, the inner cone 40 disengageing from the outer cone 41. During this axial displacement, the annular collar 62 on the end of the sleeve 39 runs axially up against the axial stop 63. The two surfaces interact as braking surfaces, so that the drum 16 is prevented from rotating. The drum 16 then stops, with the result that all the other parts of the percussion mechanism 4 which are driven from there also stop. In contrast, the rotary drive continues to run and generates a rotary drive movement via the following parts: the engine pinion 12, the gear wheel 13 of the hub 22, the intermediate shaft 14, the toothing 35, the gear wheel 36, the projections 37/38, and the collar 50 with the guide tube 28.

Because the pressure exerted on the coupling is intensified in this way as a function of the percussion mechanism 4, thus generating an increased transmittable torque, the hammer drill has the advantage that it is consequently substantially more comfortable to operate. It is substantially easier for the operator, since he does not have to press the hammer drill so hard against the workpiece to be treated.

When the coupling is separated, owing to the design of the separating spring 47 which at least resembles that of a cup spring, there is scarcely any wear, since the separating spring 47 comes up against the inside of the sleeve 39 on the one hand and the hub 22 on the other hand practically only with circumferential contact.

When the percussion mechanism 4 is engaged and the coupling 40/41 is in the active position, the drum 16 is supported axially, on the left in Figure 1, via the lever 45 which, because of the intensification of force via the leverage in the region of the lower fork-like end 54, can always exert sufficiently high axial supporting forces, in order, during the rotation of the drum 16 with swash plate 26, to support the drum 16 against separation of the coupling and axial displacement to the left in Figure 1. The drum 16 is thus supported axially to the right in Figure 1 via the lower end 54 of the lever 45 and held in this axial position when the coupling is in the active position.

In the second exemplary embodiment illustrated in Figure 2, reference symbols increased by 100 are used for the parts corresponding to the first exemplary embodiment, thus making reference to the first exemplary embodiment to avoid repetition.

In a second exemplary embodiment according to Figure 2, only the parts of the gear and percussion mechanism which are necessary for an understanding are shown. The design otherwise corresponds to that of Figure 1.

The second exemplary embodiment according to Figure 2 differs from the first in that the separating spring 147 is arranged outside the coupling 140/141 and not axially between the drum 116 and the hub 122 with the gear wheel 113. Instead of this, the separating spring 147 engages indirectly on the drum 116 carrying the moveable coupling part. The separating spring 147 is designed here as a cylindrical helical spring which at one end is supported on the upper approximately fork-like end 144 of the lever 145 and which at its other end is supported axially on a fixed part of the housing 101. The separating spring 147 extends approximately at the height of the centre axis of the percussion mechanism 104. Each leg of the fork-like end 144 can be loaded by such a separating spring 147.

The lever 145 carries, at an axial distance from its lower projection 155 which acts axially on the drum 116 via the disc 156 and the axial bearing 157, an engagement means 164 bent approximately in the form of a hook towards the drum 116. This engagement means can be cut out from the material of the lever 145 and bent out. The engagement means 164 has the form of a hook with two hook portions at right-angles to one another.

The drum 116 carries, connected fixedly to it, the annular collar 162 at the other end facing away from the coupling surfaces 140/141. The hook shaped engagement means 164 engages over the annular collar 162. At a distance below the projection 155, the lever 145 is held pivotably on the housing 101 by means of a pin 165.

When the pressing force is exerted from the direction of the tool it is transmitted to the end 144 of the lever 145 via the disc 149. The lever 145 is pivoted to the right in Figure 2 with its upper end 144 against the effect of the separating spring 147, about the pin 165 constituting the pivot mounting. The projection 155 at the fork-like lower end 154 presses via the disc 156 and the axial bearing 157 on the annular collar 162 and consequently on the drum 116 which is pressed axially relative to the intermediate shaft 114, to the right in Figure 2, with the inner cone 140 on to the outer cone 141 of the hub 122. The coupling is in the active position in which it transmits the rotary drive torque and in which the drum 116 is driven to rotate and consequently the percussion mechanism 104 operates. As in the first exemplary embodiment, the reaction force arising from the percussion mechanism 104 is absorbed by the hub 122. It is transferred to the hub 122 via the drum 116 with the inner cone 140, with the result that automatically, virtually by means of a servo-effect, the maximum transmittable torque of the coupling 140/141 is available whenever there is in the percussion mechanism 104 a maximum compression between the piston and striker which at the same time demands the maximum drive power for the piston.

When the pressing force arising from the tool decreases, the separating spring 147 relaxes. Via the latter, the lever 145 is pivoted into the initial position about the pin 165, specifically in such a way that the upper approximately fork-shaped end 144 moves to the left in Figure 2. During this pivoting movement of the lever 145, the projected distance between the projection 155 on the one hand and the portion of the hook-shaped engagement means 164 approximately perpendicular relative to the axis of the intermediate shaft, on the other hand, is reduced. In this way, via the engagement means 164 engaging on the annular collar 162, the drum 116 is shifted to the left in Figure 2 relative to the intermediate shaft 114 and the coupling is thereby separated. At the same time, the parts arranged axially between the projection 155 on the one hand and the engagement means 164 on the other hand are clamped axially. The annular collar 162 is consequently clamped axially and the drum 116 is thereby braked and locked against rotation. It is therefore no longer taken up, so that the entire percussion mechanism 104 is stopped simply as a result of the effect of the swash plate drive.

In a second exemplary embodiment, when the coupling is separated the separating spring 147 does not rotate together with one part of the coupling. This prevents wear which would otherwise possibly exist as a result.

In another exemplary embodiment not shown, the moveable coupling part 39, 40 is an element which is independent from the drum 16 and which is connected releasably to the drum 16, for example screwed to it. In another exemplary embodiment not shown, the moveable coupling part 39, 40 is coupled as an independent element to the drum 16 via positive coupling members. These coupling members can consist, for example, of coupling teeth in the form of spur teeth on the one hand on the moveable coupling part 39, 40 and on the other hand on the drum 16 which face one another axially and which are axially engaged with one another. Radial teeth are also possible as coupling teeth. It goes without saying that in contrast to the first exemplary embodiment shown, the needle cage 17 serving for mounting the drum 16 can also be omitted completely, in which case the drum 16 without a bearing is held directly on the intermediate shaft 14 so as to be rotatable and axially displaceable.

In another exemplary embodiment no shown, the coupling does not, as in the first and second exemplary embodiments, consist of a friction coupling acting non-positively, with the outer cone 41 and inner cone 40. Instead, there is a positively acting coupling, for example a coupling provided with spur teeth, and the spur teeth are arranged on the one hand on the drum 16 and on the other hand on the hub 22 on sides facing one another axially and are engaged axially.

Instead of the air cushion described, which constitutes a resilient transmitter between the piston 29 and the striker 33, this transmitter can also consist of another element, for example a spring of a different type.

CLAIMS

1. Hammer drill with a motor-driven percussion mechanism, in which a drive member moved to and fro acts via a resilient transmitter, for example a pneumatic spring, on an axially moveable striker which delivers its energy to a tool guided in the hammer drill, the drive member of the percussion mechanism being moved via a gear having a swash plate as a movement converter and an engageable and disengageable coupling having a moveable coupling part which is engageable as a function of the pressure of the tool against the workpiece to be machined and on which engages an actuator of a transmission device arranged between the inserted tool and the moveable coupling part, characterised in that the moveable engageable and disengageable coupling part (39, 40; 139, 140) is connected fixedly in terms of rotation to the swash plate (26; 126), in that the other coupling part (22, 41; 122, 141) is supported fixedly in the housing (1; 101), and in that the moveable coupling part (39, 40; 139, 140), together with the swash plate (26; 126), can be pressed under the effect of the percussion mechanism (4, 104) against the fixed coupling part (22, 41; 122, 141) absorbing the impact reaction forces.

2. Hammer drill according to Claim 1, characterised in that the fixed coupling part (22, 41; 122, 141) is supported on the housing (1; 101) by means of a bearing, especially a ball-bearing (19; 119).

3. Hammer drill according to Claim 1 or 2, characterised in that the fixed coupling part (22, 41; 122, 141) is arranged fixedly in terms of rotation and fixedly in terms of displacement on an intermediate shaft (14; 114) which belongs to the gear train between a motor pinion (12) and the tool holder (8) and which is mounted on the housing (1; 101) by means of a bearing, especially a ball-bearing (19; 119).

4. Hammer drill according to one of Claims 1 to 3, characterised in that the fixed coupling part (22, 41; 122, 141) is a fixed component of a gear wheel (13; 113) driven by the motor pinion (12) and meshing with it.

5. Hammer drill according to one of Claims 1 to 4, characterised in that the moveable coupling part (39, 40; 139, 140) is connected releasably to the drum (26, 126) of the swash plate (16;116), preferably coupled to it via positive coupling members, especially coupling teeth, for example spur teeth.

6. Hammer drill according to one of Claims 1 to 4, characterised in that the moveable coupling part (39, 40; 139, 140) is a fixed component of the drum (16; 116) of the swash plate (26; 126), for example is in one piece with it.

7. Hammer drill according to one of Claims 1 to 6, characterised in that the drum (16; 116) of the swash plate (26; 126) is held on the intermediate shaft (14; 114) so as to be rotatable relative to the latter and axially displaceable at least within limits, preferably by means of a bearing, especially a needle bearing (18; 117).

8. Hammer drill according to one of Claims 1 to 7, characterised in that the moveable coupling part (39, 40, 16; 139, 140, 116) can be pressed against the fixed coupling part (22, 41; 122, 141) on the intermediate shaft (14; 114) in the opposite direction to the direction of the impacts generated by means of the percussion mechanism (4; 104).

9. Hammer drill according to one of Claims 1 to 8, characterised in that the coupling (40, 41; 140, 141) is designed as a non-positively acting friction coupling, especially as a cone coupling, or is formed from a positively acting coupling, for example a coupling with spur teeth.

10. Hammer drill according to Claim 9, characterised in that the moveable coupling part (39, 40, 16; 139, 140, 116) carries an inner cone (40; 140) and by means of this can be pressed on to an associated outer cone (41; 141) of the fixed coupling part (22; 122).

11. Hammer drill according to Claim 10, characterised in that the outer cone (41; 141) of the fixed coupling part (22; 122) is formed by a frustoconical surface tapering towards the moveable coupling part (39, 40, 16; 139, 140, 116) and towards the tool (9).

12. Hammer drill according to one of Claims 1 to 11, characterised by at least one separating spring (47; 147) which engages directly or indirectly on the moveable coupling part (39, 40, 16; 139, 140, 116) in the coupling separation direction.

13. Hammer drill according to Claim 12, char-

acterised in that the separating spring (47; 147) is designed as a compression spring.

14. Hammer drill according to Claim 12 or 13, characterised in that the separating spring (47; 147) is designed as a cup spring or a spring at least similar to this or as a cylindrical helical spring.

15. Hammer drill according to one of Claims 1 to 14, characterised in that the drum (16, 116) carrying the moveable coupling part 39, 40; 139, 140) carries braking surfaces (62, 162) fixed on it, by means of which, when the coupling is separated, the moveable coupling part (39, 40, 16; 139, 140, 116) can, under the effect of the separating spring (47; 147), be pressed against associated braking surfaces (63; 164) and stopped in its rotation.

16. Hammer drill according to one of Claims 1 to 15, characterised in that the actuator of the transmission device is formed by a lever (45; 145) which at an approximately fork-like end (44; 144) can be subjected to the pressing force of the tool (9) against the workpiece to be treated and which, by means of the portion (54, 55; 154, 155) extending at a distance from the said end, transmits this pressing force as an axial pressure force exerted in the same direction, intensified via the leverage, to the drum (16; 116) carrying the moveable coupling part (39, 40; 139, 140).

17. Hammer drill according to one of Claims 12 to 16, characterised in that the separating spring (47) is arranged and supported between the two coupling parts (39, 40, 16 and 22, 41).

18. Hammer drill according to one of Claims 15 to 17, characterised in that the braking surface of the drum (16) is formed on an adjacent annular collar (62), to which an axial stop (63) is axially adjacent as a braking surface integral with the housing.

19. Hammer drill according to one of Claims 1 to 16, characterised in that the lever (145) carries, at an axial distance from its pressure portion (155, 154), an approximately hook-shaped engagement means (164) which via an associated annular collar (162) engages on the drum (116) carrying the moveable coupling part (139, 140).

20. Hammer drill according to Claim 19, characterised in that the lever (145) is held pivotably on the housing (101) at a distance from the engagement means (164).

21. Hammer drill according to Claim 19 or 20, characterised in that the separating spring (147), acting in the striking direction, engages on the end (144) of the lever (145) which is subjected from the percussion mechanism (104) to the pressing force of the tool against the workpiece to be treated, and, when the pressing force decreases, the lever (145) can, under the effect of the separating spring (147), pivot into an initial position in such a way that, because the engagement means (164) engages on the annular collar (162), the drum (116) can be pulled axially in the coupling separation direction and at the same time stopped in its rotation by means of an axial clamping effect between the pressure portion (154, 155) and the engagement means (164) of the lever (145).

22. A hammer drill substantially as herein described with reference to Figure 1 and Figure 2 of the accompanying drawings.

Printed in the UK for HMSO, D8818935, 7/86, 7102.
Published by The Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained.