PATENT SPECIFICATION (11) 1 602 408

1 602 408

(21) Application No. 14327/78 (22) Filed 12 April 1978 (19)
(31) Convention Application No. 198 350 (32) Filed 12 April 1977 in
(33) Dem. Rep. of Germany (DD)
(44) Complete Specification published 11 Nov. 1981
(51) INT. CL.[3] F16D 3/14
(52) Index at acceptance
F2U 206 208 286 288 310 340 396

(54) TORSIONAL DAMPERS IN MOTOR VEHICLE CLUTCHES

(71) We, VEB RENAK-WERKE, of Dammsteinstrasse 9, 98 Reichenbach, German Democratic Republic, a Corporation organised under the laws of the German Democratic Republic, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to a torsional damper in a motor vehicle clutch having a plate part carrying friction linings and a hub part, the torsional damper comprising two friction devices, one of which is operative in the no-load range and the other in the load range of the clutch.

Torsional dampers of this type are known e.g., from German Democratic Republic Specifications 54 147, 57 771 and 62 494. The frictional devices are arranged coaxially on the hub, whereby a central adjustment of the damping force is rendered possible. The accommodation of the frictional devices on one side of the clutch plate has disadvantageous consequences. As a result of it the entire axial space round the hub is covered, which leads to a large axial space requirement of the torsional damper in the region of the hub and hence to accommodation difficulties. Furthermore, the mode of arrangement of the frictional device necessitates the construction of an expensive and complicated hub.

Federal Republic of Germany Specification 23 23 872 partially overcame these disadvantages in that a shortening of the hub was achieved. However, the one-sidedness of the arrangement still remained whilst the construction of the components and their assembly became even more expensive.

According to Federal Republic of Germany Patent Specification 24 30 160 and Gebauchsmuster Specifications 74 20 306 and 75 01 962 it is also possible to arrange the frictional devices on both sides of the hub flange, while the hub flange has a particularly large diameter and is provided with ports to accommodate the compression springs. The central arrangement of the friction plate and the better distribution of the weight have favourable effects upon the hub and the entire arrangement. The space requirement of the torsional damper in the region of the hub, however, has not been reduced. The number of components has increased compared to the known constructions, more particularly the assembly has become more expensive. The production of the components necessitates production to close tolerances. Dampers of this type are generally non-variable and can be used, e.g. only for a specific type of vehicle. It is scarcely possible to adapt them to other vehicles by dimensional modifications or enlargement of the frictional devices.

The invention aims at developing a multi-stage torsional damper which is relatively simple in construction and easy to assemble. Whilst economical to produce, the torsional damper is intended to be variable in its construction.

The invention also aims at overcoming the disadvantages of the known constructions. The hub is to be simple in construction, and the frictional devices are removed out of the radial zone immediately surrounding the hub. The frictional devices are so conformed that by variations in diameter the frictional elements and additional frictional devices of the torsional dampers can be modified as to size and effect.

To this end, the present invention consists in a torsional damper in a motor vehicle clutch having separate frictional devices for no-load and load ranges, which devices are arranged on respective opposite sides of an intermediate plate carrying the clutch plate, which intermediate plate is disposed radially outwardly of a hub flange for angular movement relative to said hub, andthe no-load friction device is loaded by a helical spring which is located on the side of the intermediate plate remote from said no-load friction device to be operative between the intermediate plate and a part of the load friction device, the latter device being coaxial with and radially spaced from the hub.

In order that the invention may be more readily understood, reference is made to the accompanying drawings which illustrate diagrammatically and by way of example several embodiments thereof, and in which:—

Fig. 1 is an axial section of one embodiment of torsional damper;

Fig. 2 is a similar view of another embodiment;

Fig. 3 is a section taken on the line B—B of Figs. 1 and 2;

Fig. 4 is a modification of Fig. 3; and

Fig. 5 is a detail of Fig. 4 showing part of another embodiment.

Referring to Fig. 1, clutch linings 1 are fixed in known manner to a clutch plate 2 which is welded to an intermediate plate 3 which is angularly movable relatively to the hub 15. The transmission of the angular movement is effected, in known manner, by compression springs 13 of different characteristics which are accommodated in apertures 19 in the intermediate plate 3, a cover plate 4, a load cover plate 8 and a load friction entraining means 9. A no-load friction lining 5 is arranged between the intermediate plate 3 and the cover plate 4 which is riveted to the hub 15 in order to generate the no-load friction. The load friction device, constituted by the load cover plate 8, a load friction plate 10, load friction linings 6 and the load friction entraining means 9, is located axially opposite the no-load friction device on the other side of the intermediate plate 3. The components of the load friction device are held together by a friction housing 7 which is riveted fast to the hub 15. A helical spring 11 ensures the necessary contact pressure and is braced against a flange 20 of the load friction entraining means 9 and against the intermediate plate 3. The load friction entraining means 9 is connected by its projections 21 to the intermediate plate 3 and to the compression springs 13 and is entrained in the load range. For this purpose each projection 21 projects through the aperture 19 present in the intermediate plate 3 and abuts against the compression spring 13. The load friction entraining means 9 also has corresponding slits 22 or grooves into which lugs of the load friction plate 10 engage. In the same manner, the load cover plate 8 is also connected to the friction housing 7, so that the entrainment of the complete load friction device is ensured in the load range.

In the embodiment of Fig. 2, a separate load friction spring 12 is provided for the load friction device. The load friction spring 12 is arranged between the load friction plate 10 and an abutment plate 14 which is connected to a flange 20 of the load friction entraining means 9 by flanging or by a similar operation. The abutment plate 14 is not in direct contact with the friction housing 7. It ensures a uniform distribution of the contact pressure. The spring 12 may be a cup spring or annular spring. It will be noted that, in the embodiment of Fig. 2, the housing 7 has an axially flanged rim bearing lugs engaging in grooves in the load cover plate 8, while the helical spring 11 is operative between the intermediate plate 3 and the friction housing 7 in order to load the no-load friction device only.

The compression springs 13 are those for the load range. Separate no-load compression springs 16 are provided for the no-load range. The springs have a different characteristic corresponding to their mode of action. They are arranged in apertures 19 of the intermediate plate 3. As already described, the engagement of the load friction device occurs automatically by the projections 21 of the load friction entraining means 9 in contact with the compression springs 13. Figs. 3 to 5 illustrate various arrangements of the compression springs 13 and of the no-load compression springs 16.

Fig. 3 shows a known arrangement in which pairs of load compression springs 13 and pairs of no-load compression springs 16 are each accommodated in respective apertures 19. The number of the springs is not restricted to four. It is also possible to use e.g., three pairs of springs of different characteristics. The compression springs 13 are arranged centrally in the apertures 19 so that the no-load travel A is equally divided on both sides of the compression spring 13 as A/2. It is, however, also possible to arrange the no-load travel A on one side of the compression spring 13, while the axial extent of the no-load compression springs 16 corresponds to the length of the apertures 19 and the compression spring 13 together with the no-load travel A arranged on its one side amounts to the length of the apertures 19. Because in the no-load range the no-load compression springs 16 are compressed by overcoming the drag torque of the gearbox, the compression springs 13 can oscillate automatically and divide the no-load travel A.

The embodiment according to Fig. 4 shows the arrangement of the no-load compression springs 16 within the load compression springs 13. The no-load travel A is positioned on one side. Within the compression springs 13, the no-load compression springs 16 are braced against spring sockets 17. If the one-sided arrangement of the no-load travel A is impossible, then the no-load compression springs 16 may be arranged counteracting each other (not shown). Because in service one spring is compressed and the other relaxed each time, the mid-position of the compression springs 13 is adjusted in this manner.

Fig. 5 illustrates another embodiment. Arranged in the spring socket 17 is a ram 18 which projects through an aperture of the spring socket 17 and extends beyond the socket. The ram 18 is always held in contact with the spring socket 17 by the no-load compression spring 16. When a movement occurs in the direction to relax compression of the spring 16, the ram 18 comes into abutment against the edge of the aperture 19. Further progressive movement leads to a return movement of the ram 18 and to prevention of relaxation of compression of the no-load compression spring 16.

The functioning principle is as follows: in the no-load range only a relative movement occurs between the cover plate 8 and the intermediate plate 3. With increasing load, after a specific travel has been overcome, the load friction entraining means is entrained through its projections 21, whilst the load friction plate 10 is also entrained simultaneously. From this time onwards the friction linings of the load range are also operative. Friction linings are arranged respectively between load friction entraining means and load cover plate, load cover plate and load friction plate and load friction plate and friction housing. All the frictional devices are located radially outwardly of the hub flange so that the radial zone immediately surrounding the hub is clear. The frictional devices can be modified at any time by varying the diameter of the no-load and load friction linings and load friction plate.

The arrangement and mode of operation of the compression springs is generally known. They are in each case spring sets of different characteristics, being on the one hand compression springs for the load range and no-load compression springs. Because the projections of the load friction entraining means project through the apertures of the intermediate plate and the compression springs abut against them, after the no-load travel is overcome the load friction entraining means is automatically entrained and thus the entire load friction device is engaged. Besides the known arrangement, in which the no-load travel is arranged on both sides of the load compression springs, other arrangements are possible. Thus the no-load travel can be located on the traction side of the load compression springs. The apertures are then filled in their total circumferential extent by the no-load compression springs, whilst the length of the load compression springs is reduced by the length of the no-load travel. This construction is possible because under no-load the drag torque of the gearbox has to be overcome, whereby the no-load compression springs are pre-tensioned to the value of the drag torque. By the choice of appropriate compression springs it is possible to achieve that the compression springs oscillate so that the existing no-load travel is divided on to both sides of the load compression springs. Other travel distributions are possible in case of need by modifying the parameters of the no-load compression springs. In the embodiment in which the no-load compression springs are arranged within the compression springs for the load range, the no-load travel is made one-sided. The no-load compression springs are braced against the spring sockets within the load compression springs. The no-load compression springs act in one direction. This arrangement renders the provision of separate apertures for the no-load compression springs superfluous. In cases where a one-sided positioning of the no-load travel is impossible, the no-load compression springs are arranged counteracting each other in order to fix the central position. During operation one spring is compressed and the other relaxed each time. In the devices with no-load compression springs and compression springs for the load range arranged coaxially to each other, the precompression of the no-load compression springs is cancelled during the movement in the direction in which compression is relaxed. This results in undesirable noise. In order to eliminate this disadvantage, rams are arranged in the base of the spring sockets. When the spring arrangement moves towards the side of reduced compression, the ram comes into abutment against the aperture and prevents any further relaxation of the no-load compression spring. In this way undesirable noise is avoided.

WHAT WE CLAIM IS:—

1. A torsional damper in a motor vehicle clutch having separate frictional devices for no-load and load ranges, which devices are arranged on respective opposite sides of an intermediate plate carrying the clutch plate, which intermediate plate is disposed radially outwardly of a hub flange for angular movement relative to said hub, and the no-load friction device is loaded by a helical spring which is located on the side of the intermediate plate remote from said no-load friction device to be operative between the intermediate plate and a part of the load friction device, the latter device being coaxial with and radially spaced from the hub.

2. Torsional damper according to claim 1, wherein the helical spring also serves to load the load friction device, being operative between the intermediate plate and a load friction entraining means of said load friction device.

3. Torsional damper as claimed in claim 1, wherein the load friction device is accommodated in a friction housing which is connected by its internal radius fast with the hub of the damper and has, at its outer circumference, an axially flanged rim carrying lugs engaging in grooves provided in a load cover plate.

4. Torsional damper as claimed in claim 3, wherein a load friction spring is provided within the load friction device between a load friction plate and an abutment plate, while the abutment plate is connected at its inner periphery to a load friction entraining means and the helical spring is arranged between the intermediate plate and the friction housing.

5. Torsional damper as claimed in claim 4, wherein the load friction spring is a cup spring or an annular spring.

6. Torsional damper as claimed in any of claims 1 to 5, wherein, to transmit angular movement, load compression springs and no-load compression springs are provided which are arranged in apertures of equal circumferential extent and the arrangement is made so that the axial length of the no-load compression springs corresponds to the circumferential extent of the apertures and the axial extent of the load compression springs is determined by the length of the no-load travel A, while the no-load travel A arranged on the one side of the load compression springs together with the axial extent of the load compression springs, has the same axial extent as the no-load compression springs and the apertures.

7. Torsional damper as claimed in any of claims 1 to 5, wherein load compression springs and no-load compression springs are arranged coaxially to each other and in order permanently to maintain a given minimum compression of the no-load compression springs, spring sockets are provided in the load compression springs to accommodate the no-load compression springs and are provided with a hole which accommodates a ram which comes into abutment against the edge of the respective aperture at a predetermined reduced compression of the no-load compression springs.

8. A torsional damper in a motor vehicle clutch substantially as herein described with reference to and as shown in Figs. 1 or 2 of the accompanying drawings, optionally including the modifications of any of Figs, 3 to 5.

VENNER, SHIPLEY & CO.,
Chartered Patent Agents,
Rugby Chambers,
2, Rugby Street,
London WC1N 3QU,
Agents for the Applicants.

Printed for Her Majesty's Stationery Office by Burgess & Son (Abingdon) Ltd.—1981. Published at The Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained.

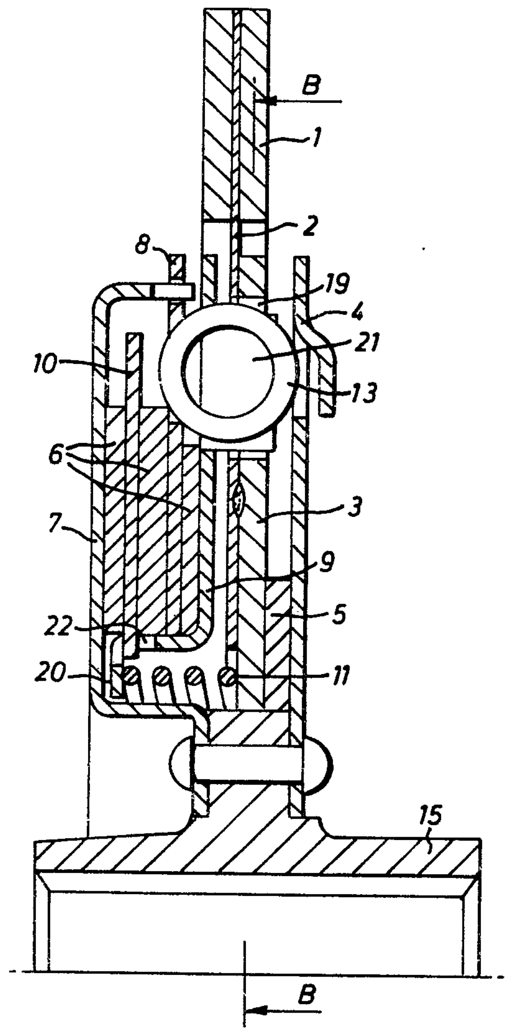

FIG.1

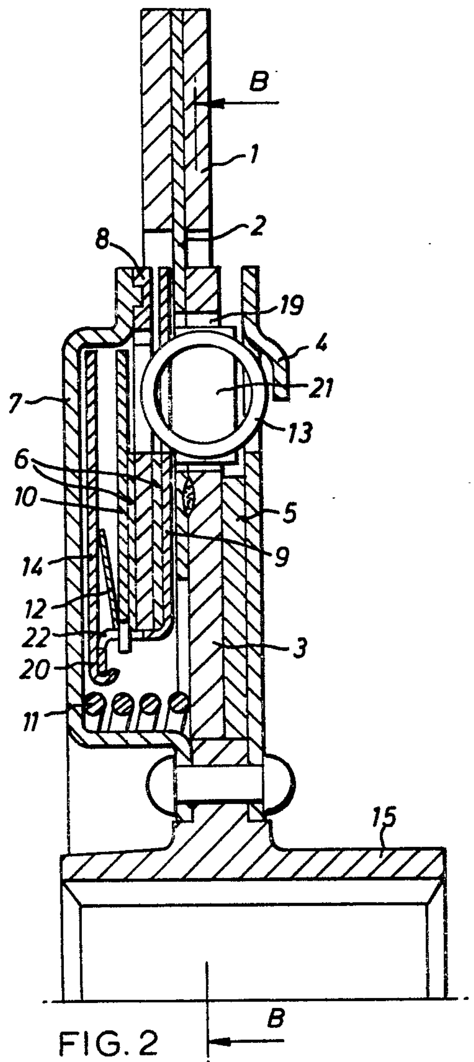

FIG. 2

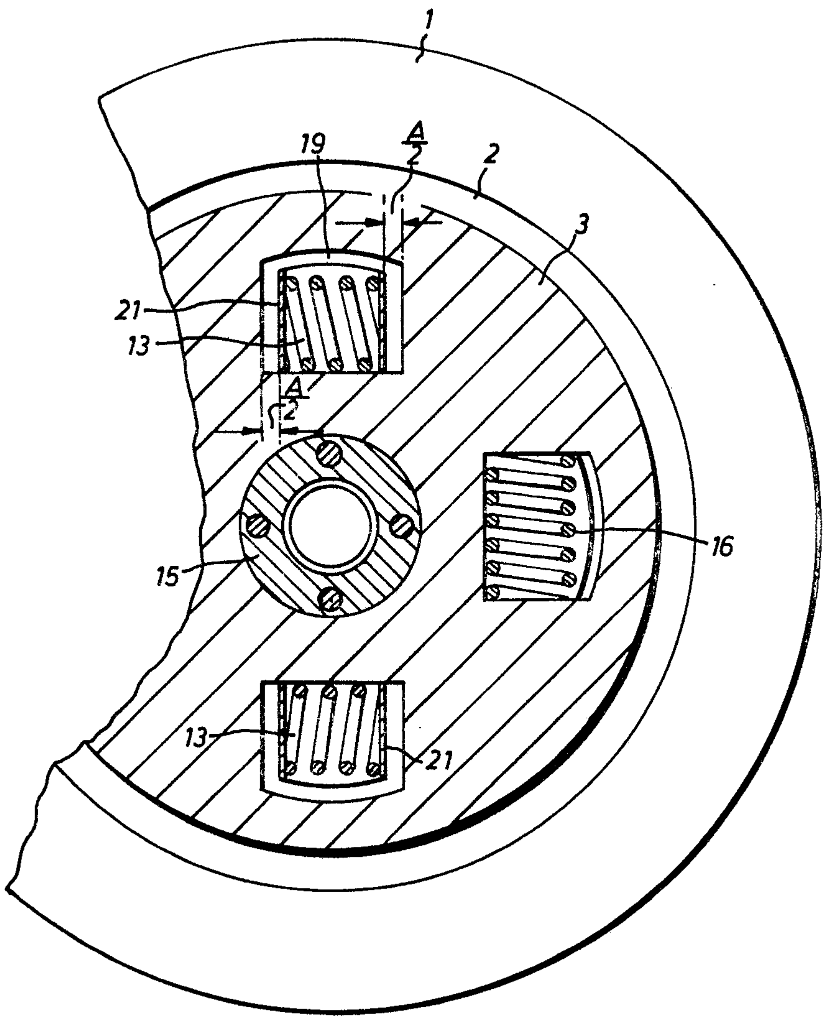

FIG. 3

1602408 COMPLETE SPECIFICATION

5 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 4

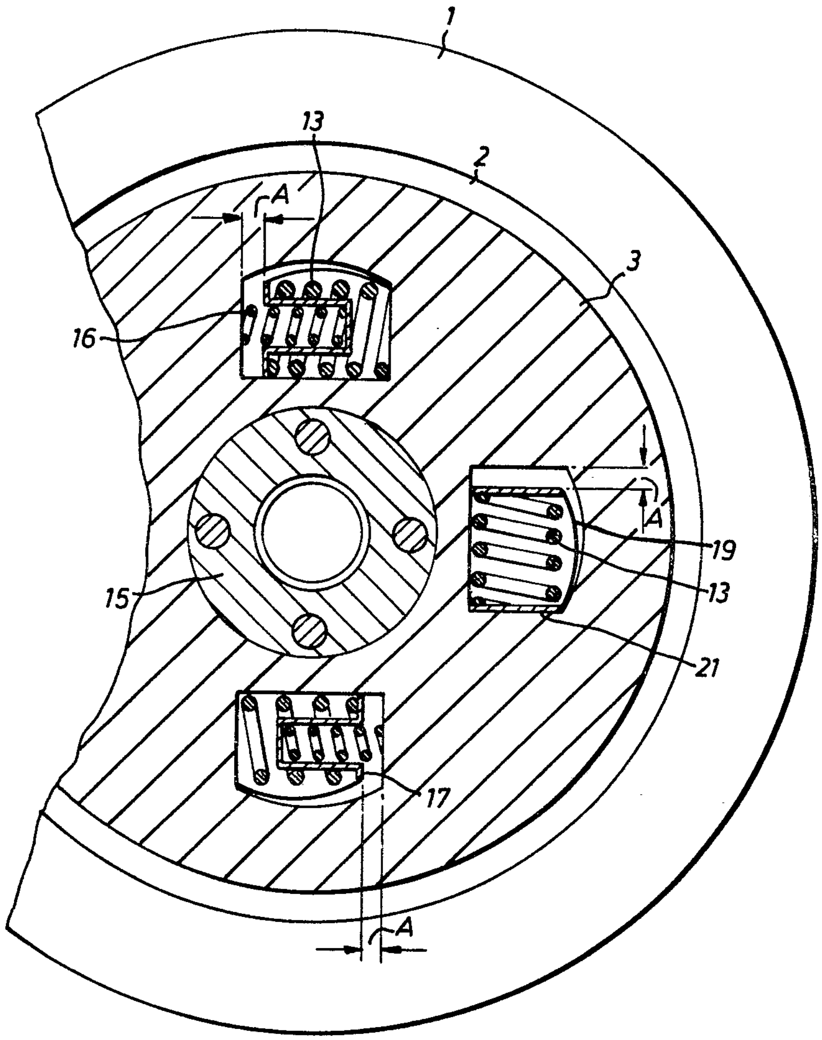

FIG.4

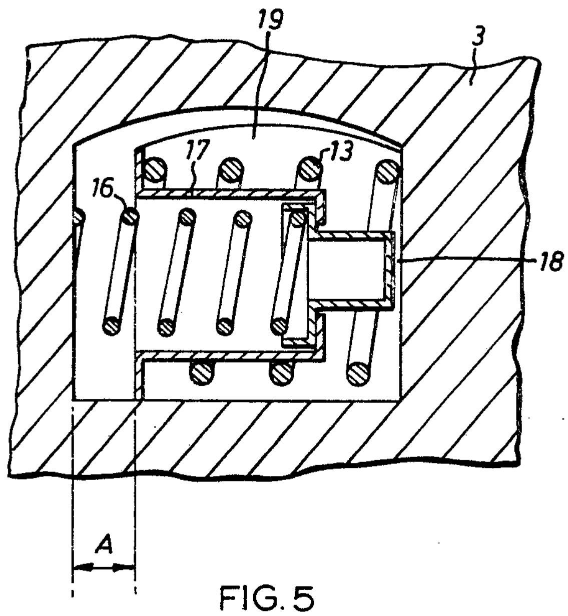

FIG. 5