DRIVING MECHANISM FOR PRODUCING TURNING MOVEMENTS SUBSTANTIALLY WITHOUT SIDE FORCES

Arthur Krávits, Budapest, Hungary, assignor to Mírköz Múszaki Irodai és Korszerűségéi Cikkelhet Gyártó és Javító K. Sz., Budapest, Hungary, a firm

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Previously known distributor pumps for fast-running multi-cylinder engines have the disadvantage that individual parts thereof subjected to friction wear out very quickly due to large lateral pressures so that the consumption of the engine is increased rapidly.

According to the invention, this disadvantage is overcome in that the relative positions of the loaded parts are determined, not by unbalanced individual forces, but by means of devices which may be called automatic force compensators which function in the manner of a balance in the direction in which friction is caused and automatically generate self-compensating forces. By this means the moving parts are not pressed transversely to their guide means. This applies to both mechanically static and mechanically dynamic forces and also to the static forces produced by the liquid pressure and, finally, to those forces produced by the dynamic effect of the flow of liquid (gas oil).

From the above it can be seen that the reciprocating and rotating parts cannot suffer any appreciable wear so that the service life of the distributor pump is almost unlimited, while the economy factor of the engine remains unchanged, even when it is used for a particularly long time.

FIG. 1 is an axial longitudinal section of an embodiment of the distributor pump of the invention having automatic force compensators. In this embodiment the compensating sleeve used for the drive is in the form of an oil pump and the piston thereof is hydrostatically and hydrodynamically compensated.

FIGS. 2, 2a, 3, 3a, 4, 4a, 4b, 4c, 4d, 4d', 5 and 5a illustrate diagrammatically automatic force compensating devices and other details.

In FIG. 1, the pump casing is denoted by 1 and the pump cylinder by 2. 3 denotes a pump piston which is driven by a driving sleeve 4 and a pin 39 so that it reciprocates and rotates simultaneously. This piston is constructed in such a manner that it is compensated both hydrostatically and hydrodynamically. 4 denotes the driving and compensating sleeve of the invention for relieving lateral load, that is, for compensating the movement of the piston. This sleeve also serves to convey the gas oil in such a way that it acts as a lubricant.

The balls 5 (see also FIGS. 2, 2a, 3, 4, etc.) are compensating members of an automatic torque compensator, which may be called a centripetally operating compensator. With this device, as soon as the torque of the claws 6 of the pinion 6 (FIG. 1) is transmitted by the balls 5 to the claws 4 of the compensating sleeve 4, the forces acting on the individual claws 4 must form a compensated couple of forces which cannot have any lateral components if the angle α of the bearing surfaces is greater than the angle of friction.

The bearing surfaces of the claws adjacent the balls can also be curved to reduce specific surface pressure.

The balls do not prevent the reciprocating movement of the sleeve 4 in the axial direction, as the balls can roll freely in this direction.

In the state of rest the balls 5 lie loosely between the claws 4' and 6'. On rotation in the direction of the arrow, a claw 6' strikes first of all the ball 5 and displaces it until it is pressed against the counter claw 4'. As the angle α is greater than the angle of friction, the system is not self-locking, but the said ball 5 moves concentrically due to the unavoidable eccentricity of the two shafts, until it presses the counter ball between the corresponding claws 6' and 4'. The transmission of force then commences. It is clear that if one of the balls 5 transmits a greater force than the other, the balls will be immediately pushed into the position of balance in which the two balls transmit equally large forces.

The function of the device is as follows:

First of all it should be noted that FIG. 1 shows the distributor in the sucking stroke, in which the gas oil (or other motor fuel) is admitted under pressure at the screw thread 26 of the head 25 and it flows through the bore 28 of a regulating valve situated over 27 and thereafter it streams in the direction of the geometrical axis of the device, that is to say through the central bore, from which it arrives into the radial bores 30. These bores are connected to the bores of a packing 25' and to the channels 31 from which the oil flows into radial bores and then into the radial sucking channels 32 of the piston 3. Now the oil oil comes into a central channel shown in dotted lines in the drawing and forming part of the working space of the pump.

If the shaft 45 is turned by means of a device screwed onto the head part 46, the claws 6' are also rotated (see FIG. 2) and thus the torque is transmitted through the two balls 5 to the claw 4'. Consequently, the sleeve 4 is also turned, which can be shifted and turned on the cylinder 2. On this sleeve are situated cams, the number of these cams corresponding to the number of the cylinders being present in the engine. More exactly, the cams are formed on the disc-like extension 12 of the sleeve 4. These cams are under the influence of rollers 14 (FIG. 6) which engage at the same time and which in turn are pivoted on a spring-controlled cross-shaped holder 15. Thus, the said cams engage the roller 9 pivoted on a holder 7 (FIGS. 5, 5a, etc.), so that the peripheral cam 45 of the sleeve 4, in consequence of the rotation of this sleeve, causes an axial movement of the sleeve, because holder 7, by means of its pivots 8', bears on a fixed part of the pump, but may be swung on this part. The shape of these cams may vary, as is well known, in accordance with the movement it is desired to impart to sleeve 4.

The said axial movement of the sleeve 4 corresponds to the shape of the cam and produces the compression stroke. According to FIG. 1, if the engine has four cylinders, at the end of this compression stroke the sleeve 4 and the piston 3 (coupled to it by means of the pin 39) are turned by 45°. In this position the gas oil (or in general the motor fuel) is flowing into the central channel of the piston 3 and therefrom into the radial bore 34, in which it streams during the compression stroke. Now the oil comes into the compression channels 31' and from this it is pressed through packings (not shown, but like the packing 25') into outlet openings 29, from which it is led to the individual cylinders of the engine.

With the embodiment shown in FIGS. 3 and 3a, the balls do not have a centripetal moment influence but a centrifugal one and the forces are transmitted by means of screws 50a and the ring 51. During operation, the balls move radially, due to inaccuracies in machining, whereby the gaps between the claws 48, 49 also alter slightly whereupon the pressure exerted on the claws is practically completely compensated by transmission of pressure between balls 47 through ring 51. As shown in FIGS. 3 and 3a, ring 51 floats relative to claws 48 and 49.
The screws 59a can be adjusted by means of the nuts 50 so that the angular position of the claws 48 in relation to the claws 49 can be adjusted, which changes the instant of injection.

The above-described devices serve to compensate the torque, but the force of thrust is also compensated, as shown in FIGS. 1, 4 and 4a. The automatic thrust compensator 7 co-operates by means of pins 8 and rollers 9 with the cams on the disc-like extension 12 on the driving shaft 10 to actuate the piston 3 in its pressure thrust without any lateral load. For this purpose the compensator 7 is supported by pins 8 (FIG. 4) which are pivoted on the bearings 42, 42. The rollers 9 are secured by a ring 11 and a pin 10.

As a result of the unavoidable eccentricity of the sleeve 4 in relation to the compensator 7 and of the inaccuracy in machining of the cams on the disc 12, it is possible that one cam may operate before its associated opposite cam, which would give rise to lateral load on the sleeve 4. However, since the compensator 7 can be rotated about the pin 10, the two thrusts cancel each other out in the manner of a balance so that only an axial resultant force is produced.

In FIGS. 4d and 4d' an embodiment of the compensator 7 is shown in which, instead of radial cams 12 and rollers 9, pins 10 and limiting sleeve 11, conical rollers 9' are used which are supported on balls and which co-operate with conical cams on the sleeve 4.

FIGS. 4b and 4c show an embodiment of the thrust compensator 7 in which pairs of conical cams 12' and 12' and pairs of rollers 9'a and 9'a are used. In FIG. 4c the compensator 7 has a two-armed lever 7' which can swing about the pin 8' and which does not transmit any lateral forces, since it co-operates with the compensator 7 in the manner of a balance. In this embodiment, four cams operate simultaneously.

In FIGS. 5 and 5a the piston 3 is moved by the sleeve 4 in its suction stroke without the occurrence of lateral forces, due to the compensating devices shown in these figures. Since the compensator 15 can be turned about the pin 17 and also, since the pins 17' and sleeves 17' can move freely in the axial direction in the recess in the guide piece 42, the two thrusts cancel out in a similar manner to that effected by the compensator 7. Actuation is effected by means of the compression spring 18 which is supported by a ring 19 which turns in the manner of a balance about a fulcrum 20 in a recess of the counter bearing 20a.

The above-described devices are suitable for compensating the dynamic forces which occur when the pump is in use.

The above-described automatic compensators obviate all lateral forces and thus prevent any wear of the parts under a friction load. The advantages of a practically frictionless mode of operation are enhanced by the relief of the pressure side in a closed space by using a specifically designed cam. Such a cam would include a pressure portion, a relief portion and an effective suction portion.

In the pressure portion of the stroke the channels leading from the working chamber to the pressure side open and remain open during the relief portion. On the conclusion of the relief portion and commencement of the suction stroke, the communicating channels from the suction side to the working chamber are opened after a certain angle has been traversed. This avoids the occurrence of currents having too high speeds and large pressure differentials so that the control edges are not worn down by liquid currents and the pumping rate is accurately maintained for a long time.

It is advantageous to control the fuel by throttle means, in particular by a cam profile which rises in the suction stroke, as this can bring about a flexible full charge (dose). Further, it is advantageous to install check valves on the suction side for maintaining the desired initial charge during a decrease in speed.

What I claim is:

1. A driving mechanism for producing turning movement substantially without side forces, comprising a driving member having two claws, a driven member coaxially mounted with said driving member and having two claws opposite the first-mentioned claws, said driving claws having surfaces confronting surfaces on said driven claws, and rolling means disposed one between each pair of the claws engaging said confronting surfaces to transmit driving torque between said members, said rolling means directly engaging each other to transmit force between said rolling means, the angle between said confronting surfaces engaging the rolling means being larger than the angle of friction.

2. A driving mechanism as claimed in claim 1, said rolling means being spheres.

3. A driving mechanism for producing turning movement substantially without side forces, comprising a driving member having two claws, a driven member coaxially mounted with the said driving member and having two claws opposite the first-mentioned claws, said driving claws having surfaces confronting surfaces of said driven claws, rolling means disposed one between each pair of the claws engaging said confronting surfaces to transmit the driving torque between said members, and balancing means in contact with both said rolling means and movable relative to said claws to transmit force between said rolling means, the angle between said confronting surfaces engaging said rolling means being larger than the angle of friction.

4. A driving mechanism as claimed in claim 3, said rolling means being spheres.

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BROUGHTON G. DURHAM, Primary Examiner.

LAWRENCE V. EFNER, Examiner.