A mechano-hydraulically operated governor is provided with a main body rigidly mounted on a rotational shaft and a supplemental control valve means operationally connected with the body. In the main body valve means are mounted on the shaft so as to be moved radially or axially relative thereto under the action of the centrifugal force generated by the rotation of the shaft.

Hydraulic output governor pressure can be varied in a stepped mode for a continuously increased or decreased rotation of the shaft.

10 Claims, 11 Drawing Figures
MECHANO-HYDRAULICALLY OPERATED GOVERNOR

This invention relates to improvements in a mechano-hydraulically operated governor.

In the conventional mechano-hydraulically operated governor of the above kind, the hydraulic line pressure fed from a certain source attached to the apparatus is kept in balance with a centrifugal force developed by a centrifugal mass acting as a valve member in the apparatus for a control of the hydraulic pressure output to be delivered.

On the other hand, this kind of governor apparatus is so designed and arranged that the governor output hydraulic pressure or briefly the governor pressure is in proportion to the revolutions of the governor shaft. The hydraulic characteristic of this kind of governor unit is a parabolic or secondary derivative curve which is, however nearly a linear one representing two or three steps, thus varying in nearly linear.

When such governor is used with automatic speed changer for automotive vehicle, as an example, the governor pressure is applied onto one end of a shift valve, so as to shift the valve member against mechanical resilient pressure exerted thereupon in the opposite sense by a return spring. When the governor pressure exceeds the working load of the return spring, a hydraulic operating circuit for a clutch, a brake band or the like means is subjected to an on-off control action according to our practical experience. However, the speed changing points are highly fluctuated in use of such governor unit cooperatingly arranged with the automatic speed changer.

It is therefore an object of the invention to provide an improved mechano-hydraulically operated governor unit, capable of substantially obviating the aforesaid drawbacks inherent in the prior art.

A further object of the invention is to provide a mechano-hydraulically operated governor unit of the kind above referred to, capable of developing a hydraulic output pressure varying in a stepped mode for a continuously increased or decreased rotation of the governor shaft and minimizing otherwise encountered variation in the speed changing points, thus providing an effective operation of the shift valve of an automatic speed changer when it is arranged to cooperate with the governor unit according to the invention.

These and further object, features and advantages of the invention will become more apparent when read the following detailed description of the invention by reference to the accompanying drawings illustrative of several preferred embodiments of the invention.

In the drawings:

FIG. 1 is a substantially axial main section of a mechano-hydraulically operated governor unit according to this invention.

FIG. 2 is a substantially an axial section of a governor pressure control section which is arranged as a supplemental part to the main section shown in FIG. 1.

FIG. 3 is a chart showing the oil pressure characteristic curve of the main section shown in FIG. 1, being plotted against the revolutions of the governor shaft.

FIG. 4 is a similar view to FIG. 1, illustrative of a second embodiment.

FIG. 5 is a similar view to FIG. 3, illustrative of the hydraulic output pressure characteristic of the second embodiment.

FIG. 6 is a similar view to FIG. 1, showing a third embodiment of the invention.

FIG. 7 is a similar view to FIG. 1, showing a control for the embodiment shown in FIG. 6.

FIG. 8 is a chart showing the relationship between the governor revolutions and the line pressure in the case of the mechanism shown in FIG. 6.

FIG. 9 is a chart showing the governor pressure characteristic of the apparatus in FIG. 6.

FIGS. 10 and 11 are similar views to FIG. 1, illustrative of a fourth and fifth embodiments of the invention.

Referring now to the accompanying drawings, a preferred embodiment of the invention will be described in detail.

Numeral 10 denotes a main body of the governor which main body is fixedly mounted on a rotatable shaft 11. In a radially elongated inside space 10b of the main body 10 relative to the shaft 11, a slidable valve member 13 having a substantially mass is mounted and an urging return spring 12 is inserted under tension between the upper recess end of the valve member, on one hand, and a stopper 15 formed into a perforated disc and held in position within the upper end part of said inside space 10b by means of a spring clip 14 fixedly, but detachably attached to the inside wall of the main body, on the other hand, said valve member being urged to occupy nor- normally the shown position keeping in contact with the shaft 11.

The lower half part of the valve member 13 is formed into a rigid spool part 16 having four separated and parallel peripheral grooves 18–21, while the upper half of valve member is formed into the hollow cup-shaped and peripheral recess spool part 17. A positioning ball 22 is inserted in a lateral groove 10b and urged into contact with any one of said peripheral grooves 18–21, as will be more fully described hereinafter. For this purpose, the ball 22 is backed up by a coil spring 24 which is also inserted in the lateral groove 10b and held in position by means of a small male-threaded plug 23.

Lateral ports 25–27 are bored through the wall of the hollow main body 10. Port 25 is connected through a piping 30, a reduced passage 29 inserted therein and only schematically represented for simplicity and a further piping 28 with a line pressure source, not shown, for receiving a line pressure, as will be more fully described in detail. Port 26 is connected with a piping 31, and port 27 is connected with a piping 32. Pippings 30–32 are connected respectively with pippings 30–32', in FIG. 2, which are connected to a pressure regulator valve unit 33 through its ports a, b and c, respectively. Valve unit 33 comprises a stepped valve plunger formed with collars 34, 35 and 36 having successively enlarged diameters. Ports a, b and c are so designed and arranged that they are adapted normally for applying hydraulic pressure coming from respective pippings 31' and 32' onto the plunger collars 34, 35 and 36, respectively.

The largest plunger collar 57 is arranged to receive the line pressure supplied from the source, not shown, through a piping 37, a reduced passage 38 therein and a further piping 39 connected therewith. A piping 40 is branched off from said piping 39 for leaking out therefore the governor pressure, as will be mentioned more in detail hereinafter. As shown, the valve unit 33 is further fitted with drain ports 41 and 42.

The operation of the mechano-fluid operated governor is as follows:

When the shaft 11 carrying main body 10 is kept either stationary or in slower rotational speed than prescribed one, with the valve member 13 being positioned as shown, or at a slightly elevated position, line pressure will be conveyed from piping 28, reduced passage 29 and port of piping 30 to port 25. In this case, however, the lower spool part 16 of centrifugal valve member 13 is positioned exactly or substantially at the shown position so that it does not cover drain port 42, thus line pressure applied on the valve member 13 being drained without exerting any effective influence upon the valve 13. Therefore, the hydraulic pressure prevailing in the piping 30 is maintained substantially at nil.

With increase of the rotational speed of shaft 11, the valve member 13 is urged to move outwardly by the correspondingly increased centrifugal force against the spring urging force 12, thereby drain port 42 being closed and thus the line pressure being conveyed from piping 30 (FIG. 1) to 30' (FIG. 2).

With further increase of the rotational speed of shaft 11, the valve member 13 is positioned at a still further outwardly shifted position as to open port 26, thus the line pressure conveyed from port 25 is discharged through the now opened port 26 to the piping 31.

With still further increase of the rotational speed of shaft 11, the valve member 13 is positioned at a still further outwardly shifted position, so as to open the next outwardly posi-
tioned port 27. Therefore, the line pressure is discharged through the now open port 27 to its related piping 32, and vice versa.

The centrifugal control valve unit is so designed and arranged that with the drain port 42 kept open, the positioning ball 22 is kept in engagement with the uppermost peripheral groove 18; with the port 42 kept closed, the ball is kept in engagement with the new inner groove 19; and with the port 26 opened, the ball is kept in engagement with the new inner groove 20. In the similar way, the ball 22 is brought into engagement with the innermost groove 21. The provision of such ball-and-groove serves for positioning the valve member at any one of the abovementioned several steppedly shifted positions and for exerting a returning effect to a retarding effect to a certain degree against the centrifugal shifting of the valve member.

Although the line pressure acts upon the valve member conveyed from the pressure oil source, not shown, through piping 37, reduced passage 38 and further piping 39, and is being exerted upon the plunger collar 57, even with the shaft 11 kept stationary or rotating at a slow speed, the oil pressure will be discharged through the drain port 41 because at this stage the stepped valve member 33 has been shifted to its left-hand end position from the position shown, thus the drain port 42 being kept open and the hydraulic or governor pressure in the output piping 40 being nil at this stage.

With increased rotation of the shaft 11 and with closure of drain port 42, application of the line pressure through the individual piping 30 and 30’ and port 24 upon the left-hand most reduced plunger collar 34 will provide a delivered pressure in relation to the differential cross-sectional area between the largest collar 57 and the smallest collar 34. With further increased rotational speed of the shaft 11, the line pressure is fed through pipes 30’—31’ and 32’—32 upon the plunger collars 35 and 36, respectively, a correspondingly step-upped governor pressure in relation with the respective differential cross-section between the largest collar 57 and the next larger and still next larger collar 35 and 36, respectively, can be taken out from the output piping 40.

Next, referring to FIG. 3, this chart illustrates the relationship between the governor pressure delivered from the output piping 40 and the revolution speed of shaft 11. In this figure, references a, b and c are illustrative of the respective opening of the ports a, b and c shown in FIG. 2 as shown, the governor pressure varies sharply in a stepped manner. The dotted line curve represents a stepdown operation of the governor with an appreciable hysteresis.

Next, referring to FIG. 4 and the second embodiment of the invention will be described. Corresponding parts shown in FIG. 4 are denoted with respective same reference numerals, each being, however, added with 100 for quicker comparison and identification.

The function of the second embodiment is performed just in the opposite way to that described in the foregoing description with reference to FIGS. 1–3.

With increase of the rotational speed of the shaft 111, ports 127 and 136 are successively closed by the lower spool 116 of the valve member 113. This operation can be explained by reference to FIG. 2. With the governor main body 110 kept stationary or rotating at a slow speed, the hydraulic line pressure is applied successively to the ports a, b and c of the controller shown in FIG. 2. In this case, the line pressure is applied successively and sequentially upon the respective plunger collars 34, 35 and 36, said applied pressure being counteracted by the line pressure applied in the opposite direction upon the largest plunger collar 37. With increase of the rotational speed, the uppermost stage pressure shown at c in FIG. 3 will first be lost and the governor pressure will be correspondingly reduced. With further increase of the rotational speed of the shaft 111, the pressure shown at b in FIG. 3 will be lost and the governor pressure will be reduced correspondingly. The results will be shown in FIG. 5. As seen, the hydraulic characteristic curve has in this case the opposite sense when compared with that shown in FIG. 3.

In the third embodiment shown in FIGS. 6 and 7, respective same numerals as used in the first embodiment are used, yet each being added with 200 for more easier and quicker comparison and identification.

A main body 210 is fixedly attached to a rotateable governor shaft 211 and therein with two pneumatically compartments 210a and 210b in which weight masses or centrifugal valve members 243 and 244 are slidably mounted and subjected to inwardly directing urging force exerted thereupon by respective return springs 245 and 246 mounted also in the space comprising the ports 218 and 216. The centrifugal valve 244 has a larger weight mass than that of valve 243.

Ports 247 and 248 are formed laterally toward the wall of the main body 210 and communicating with the interior spaces of the compartments 210a and 210b and positioned at the opposite sides of the shaft 211 to be off-on controlled by the respective valve members 243 and 244. These ports are connected with respective pipelines 249 and 250 for receiving line pressure fed from a common piping 228 connected with a certain supply source, not shown, and through respective reduced passages 229 and 229’. The pipelines 249 and 250 are connected with respective feed pipes 249’ and 250’, feeding respective ports shown in FIG. 5. With same reference numerals, these ports being formed in a press regulator valve unit 233 shown only partially and in section.

In the body of this valve unit 233, a stepped plunger comprises three successively arranged collars or lands 251, 252 and 259 having steppingly increased diameters. Below the plunger and separated therefrom a certain distance, there is provided a slide valve member 254 urged leftwards by an urging spring 253.

When the shaft 211 is kept stationary or rotates at a slower speed than a predetermined value, the valve members 243 and 244 are positioned precisely or nearly in their innermost positions in close contact with the shaft 211 so that the ports 247 and 248 are kept open. The line pressure fed from the common feed piping 228 is supplied through these opened ports to the interior space compartments 210a and 210b. Although not shown, however, these space compartments 210a and 210b are each formed with a permanently open drain port, not shown, corresponding to that shown at 42 in FIG. 1, the thus fed line pressure being discharged through these drain ports to outside of the main body and can not be conveyed through outlet piping 249—249’ and 250—250’. At this stage slide valve 254 is kept at the left-hand end position from the shown position under the influence of spring 53. This position is at most left one, by virtue of projection of a stop means, not shown, for slide valve. Control valve member 33 is at the left-hand end position from the shown position under the influence of the line pressure applied at its right-hand end, as in the similar way as the first embodiment shown in FIGS. 1 and 2. For this purpose, a certain stopper means is provided for the member 33. This kind of stopper means is also provided in the foregoing first and other embodiments, although its graphical representation has been omitted only for simplicity of the drawings.

With increase of shaft revolution speed, however, the heavier valve member 244 is first urged centrifugally to move in the outward direction, thus the port 248 being closed at first, while the port 247 is kept open by a lighter weight of the related centrifugal valve member 243. Therefore, the line pressure is conveyed through piping 250, to the port denoted with the same reference numeral in FIG. 7.

With further increase of the shaft rotational speed, both valve members 243 and 244 are brought into such respective positions as to cover the respective ports 247 and 248 for closing the same. At this stage, therefore, the line pressure is delivered through the pipes 249’ and 250’ upon the collars 251 and 252 of the valve unit 233 shown in FIG. 7.

With still further increase of the shaft rotational speed to the maximum speed range, the heavier valve member 244 is centrifugally moved further outwards beyond to the port 248 which is thus opened again, while the lighter valve member 243 is closing its related port 247. Therefore, the line pressure
can be transmitted at this stage through its related delivery piping 249', while the line pressure being discharged through the port 248.

Therefore, the pressure prevailing in a duct 100 kept in communication with port 250' (FIG. 7) will drop substantially to nil, thus the slide valve 254 being urged to move leftwards by spring force at 253, so as to bring ducts 101 and 102 into mutual communication. Therefore, the line pressure conveyed from piping 249' is fed through ducts 101 and 102 upon the largest collar 258.

The resulting operation can be better understood by reference to FIGS. 8 and 9. In FIG. 8, the upper part illustrates the hydraulic pressure variation prevailing in the piping (port) 249' plotted against the rotational speed of governor shaft 211. In the similar way, the lower part of this figure illustrates the hydraulic pressure variation prevailing in the piping (port) 250'. FIG. 9 illustrates the output governor pressures obtainable as outputs from the combined arrangement shown in FIGS. 6 and 7 in combination.

In these figures, d - g show the abovementioned four operational stages, successively.

The governor shaft 211 is kept stationary or rotating at a slower speed than in the first operational stage of the foregoing operational description, the pressure prevailed in the pipeangs 249' and 250' is substantially nil.

In this case, the control valve unit 233 is actuated only by the line pressure fed through the piping 239 to move into the left-hand end position from the position shown in FIG. 7. Thus, the hydraulic pressure fed through the passage 239 is discharged through the drain port 241. Accordingly, the governor pressure in the outlet piping 240 will become nil at this stage.

With increase of shaft revolution speed corresponding to the second operational stage in the operational description hereinafter, the port 248 only is closed. Thus, the line pressure is applied onto the collar 251 through the piping 250'. The control valve unit 233 is shifted rightwards to close the discharge port 241. Thus, a governor pressure is obtained in the same manner as mentioned hereinafore concerning the first embodiment. The governor pressure thus obtained is represented by step e in FIG. 7.

With further increase of rotational speed of the shaft 211 than that of the second operational description, the both ports 247 and 248 are closed, so that the line pressure is delivered through the passages 250', 100 and 249', 101 onto the collars 251 and 252. Thus, the governor pressure denotes as step f in FIG. 9 can be taken out from the outlet piping 240.

With still further increase of the shaft rotational speed to the maximum range thereof, the line pressure is conveyed upon the plunger 258 through passages 249', 101 and 102 in the same manner as described hereinafore. Thus, the highest governor pressure represented by step g shown in FIG. 9 can be obtained.

In the fourth embodiment shown in FIG. 10, corresponding parts shown therein are denoted with respective same numerals used in FIG. 1, however, each being added with 'a'.

A circular main body 310 is fixedly mounted on a rotatable governor shaft 311, only a part of the body 310, however being shown in FIG. 10. In the circular inside space of the main body 310, an axially slideable valve member 313 having a circular recess 317 is mounted on the shaft 311 and an urging return spring 312 is inserted between the valve 313 and a spring mount 350 rigidly mounted on the shaft 311. A passage 356 is formed in the shaft 311 and connected with a chamber 310b formed by the shaft 311, the main body 310 and the valve 313.

In this embodiment, however, port or passages 325-327 are performed as shown 311, which are connected with passage 30'-31a in FIG. 2, respectively. The passage 325 is further connected with a hydraulic source (not shown) through a piping 328 and a reduced passage 329 inserted therein.

The operation of this embodiment is as follows: When the shaft 311 is kept either stationary or slower rotational speed than a predetermined one, the valve member 313 is kept in its left-hand position as shown to open the port 325.

The line pressure fed through the passage 325 is discharged through a discharge port (not shown). Therefore, no line pressure can be conveyed through the passages 326 and 327, which lead to the port b and c of the control valve unit shown in FIG. 2.

With increase of the rotational speed of the shaft 311, fluid fed constantly through the passage 356 into the chamber 310b is compressed into the outer region therein because of centrifugal force generated by the rotation of the shaft 311. Therefore, the valve member 313 is shifted slightly rightwards by the pressure of the compressed fluid to close the input port 325 by the right hand end of the valve member 313 so that the line pressure through the piping 328 is conveyed through the passages 325, 30 to the port a of the control valve unit shown in FIG. 2.

With further increased rotational speed of the shaft 311, the valve 313 is further shifted rightwards to cover the ports 325 and 326 with the recess 317. Therefore, the line pressure is fed to the port b in FIG. 2 through the passages 325, 326 and 31.

With still further increased speed of the rotational shaft 311 within the maximum range thereof, the ports 325-327 are covered with the recess 317 to be communicated. Thus, the line pressure fed through the passage 325 is conveyed to the ports b and "c" through the respective passages 326, 31 and 327, 32.

On the fifth embodiment of the invention, a circular valve member 460 formed with two projections 460a and 460a' oppositely in the diametral direction is rigidly mounted on the shaft 411. A circular valve member 413 formed with two projections 413a and 413a' axially parallel to the projections 460a and 460a', is slidably mounted on a rotational shaft 411, the valve 413 having a recess 413 which forms a longitudinally extended chamber. A returning or compressed spring 412 is inserted between the member 460 and the projections 413a and 413a' of the valve 413. Centrifugal mass members 455 and 455', each having substantial mass, are linkedly and movably connected with respective projections 460a, 413a and 460a', 413a' through respective arms 446, 447 and 446' and 467'.

The remaining ports 425-429 in FIG. 11 are formed and arranged in the similar way as in the foregoing example shown in FIG. 10.

The operation of the fifth embodiment is as follows: In stationary or near stationary state of the rotational shaft 411, the valve 413 is kept in the position as shown, and the port 425 is opened. In this case, the line pressure is discharged through the output port (not shown). Therefore, the line pressure can not be transmitted to the ports a, b and c of the supplementary means shown in FIG. 2.

With the increased rotational speed of the shaft 411, the mass member 455 and 455' are slightly outwardly shifted by the centrifugal force caused by the rotation, so that the slideable valve 413 is moved leftwards to close the input port 425.

Thus, the line pressure fed from the source (not shown) through the piping 428 is conveyed to the port a shown in FIG. 2.

When the rotational speed of the shaft is further increased, the valve 413 is further shifted leftwards according to the same principle as mentioned hereinafore to cover the ports 425 and 426 with the recess 413' of the shaft 411. Accordingly, the line pressure is conveyed to the port b through the passages 425 and 426.

With the maximum or near maximum rotational speed of the shaft 411, the mass member 455 and 455' are moved to the outermost position so that the ports 425-427 are covered with the recess 413' of the valve 413 shifted in the left-hand end position against the action of the spring 412. Thus, the line pressure fed through the port 425 is transmitted through the passages 425 and 426 to the ports b and c shown in FIG. 2.
In the fourth and fifth embodiments of the invention, it will be easily understood that governor pressure can be obtained under the same operation of the control valve unit shown in FIG. 2 as mentioned hereinabove on the first embodiment.

What is claimed is:

1. A mechano-hydraulically operated governor comprising a rotatable shaft, a main governor body fixedly mounted on the shaft, governor valve means slidably mounted in said main body, spring means for biasing the governor valve means in a predetermined direction, a line pressure supply source, a pressure controller for regulating line pressure to obtain a governor output pressure, said pressure controller having a stepped valve plunger slidably mounted in a housing, said stepped valve plunger having stepped collars with successively enlarged diameters provided thereon, first passage means communicating said main governor body with said pressure controller housing, said line pressure supply source being connected with said first passage means and with the largest of said collars of said valve plunger, a governor output pressure passage connected with the largest of said collars of said valve plunger, an exhaust passage formed in said pressure controller housing, said stepped collars connected with said first passage means respectively for controlling governor output pressure in response to the line pressure in said first passage means controlled by the governor valve means, said line pressure supply source being connected fluidically with the governor output passage, said stepped valve plunger being movable in response to the pressure in said first passage means controlled by said governor valve means to regulate the pressure in said governor output passage.

2. A mechano-hydraulically operated governor as set forth in claim 1 being characterized in that the main governor body is provided with ports directly connected with said first passage means respectively, said governor valve means being slidably mounted normal to the shaft, and the smallest one of said collars on said stepped valve plunger being fluidically connected with the passage connected with said line pressure supply source, said ports being progressively opened as the shaft speed increases to communicate line pressure with each stepped collar successively to increase the governor output pressure.

3. A mechano-hydraulically operated governor as set forth in claim 2, being characterized in that the governor valve means is formed with parallel peripheral grooves and the main governor body is provided with spring biased ball detent means for engagement in said grooves to provide for stepped operation of the governor.

4. A mechano-hydraulically operated governor as set forth in claim 1 being characterized in that the main governor body is formed with ports directly connected with said respective first passage means, said slideable governor valve means being mounted for normal movement relative to the shaft and the smallest one of said collars is fluidically connected with the passage connected with said line pressure supply source, said ports being progressively closed as the shaft speed increases to block communication of line pressure with each collar to progressively decrease the governor output pressure.

5. A mechano-hydraulically operated governor as set forth in claim 4 being characterized in that the governor valve means is formed with parallel peripheral grooves and the main body is provided with spring biased ball detent means for engagement in any one of said grooves to provide for stepped operation of the governor.

6. A mechano-hydraulically operated governor as set forth in claim 1 being characterized in that the main governor body is formed with two compartments radially opposed to the shaft, a valve member provided in each respective compartment, spring means biasing each valve member radially inwardly and port means in each compartment, spring means biasing each valve member radially inwardly and port means in each compartment directly connected with a respective passage means.

7. A mechano-hydraulically operated governor as set forth in claim 6 being characterized in that said valve members are different in weight.

8. A mechano-hydraulically operated governor as set forth in claim 1 being characterized in that said pressure controller further comprises a spring biased slideable spool valve means adjacent said first mentioned stepped valve plunger, and second passage means operatively connecting said first mentioned stepped valve plunger and said spring biased slideable spool valve means.

9. A mechano-hydraulically operated governor as set forth in claim 1 being characterized in that said first passage means are formed in the shaft with said governor valve means being axially slideable relative to the main body, a hydraulically operating chamber formed by the main body, said governor valve means and said shaft for controlling the sliding movement of the governor valve means and a further passage in the shaft for supplying the chamber with hydraulic fluid.

10. A mechano-hydraulically operated governor as set forth in claim 1, further comprising a movable centrifugal mass member operatively connected between the shaft and the governor valve means, the governor valve means being axially slideable on the shaft, with said first passage means being formed in the shaft.