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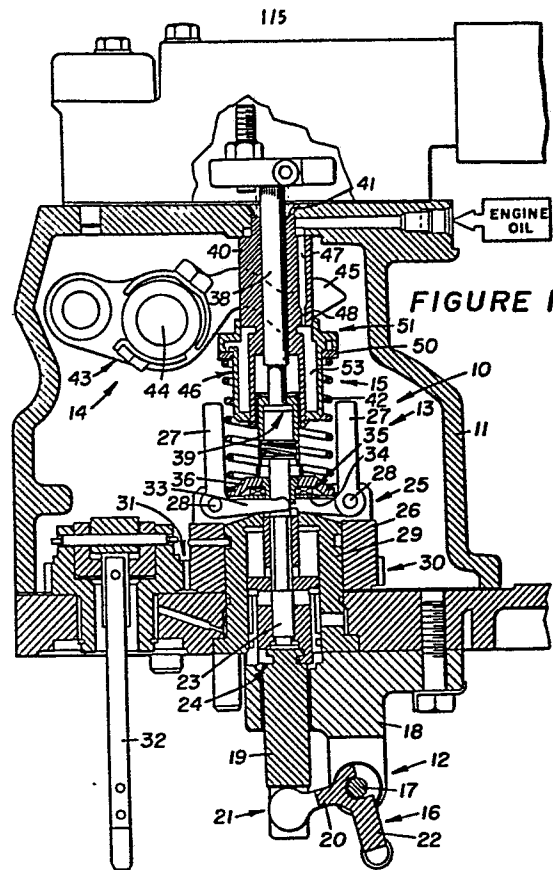
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(54) **Servo booster apparatus for engine governor.**

(57) The pedal effort required to change the setting of a governor (13) of an engine increases sharply with increased engine speed. Booster controls have been provided in the art for applying a boost force (F_B), additive to an operator input force (F_L), for opposing the force (F_s) of a governor spring (42), thus to reduce pedal effort at high engine speeds. The improved booster control (15) herein includes an actuating chamber (53), and a control arrangement (51) for controlling fluid pressure in the chamber (53) when the setting of the governor (13) is changed by the operator and in a predetermined ratio to the operator's input force. The booster control (15) has wider application wherein a boosting force (F_B) is desired to supplement an input force (F_L) for controlling movement of a member (23) having an opposing force (F_s) applied thereto.



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SERVO BOOSTER APPARATUS FOR ENGINE GOVERNOR

This invention relates to a servo booster governor control (hereinafter called a booster) particularly adapted for automatically reducing throttle control effort during operation of an internal combustion engine.

5. Fuel injection systems of the type employed in diesel engines, typically run at speeds of from 650 to 2400 rpm, and must be precisely designed to exhibit trouble-free operation over an extended period of time. Such systems include a throttle control, usually
10. including an accelerator pedal, for increasing the power of the engine at the will of the operator. The accelerator pedal is connected through a suitable linkage to a governor control lever which functions to compress a spring and associated flyweights thus moving the
15. governor to a higher setting. The governor spring controls actuation of another linkage, interconnecting the governor with the injection pumps of the engine, to control closely injection of fuel into the cylinders of the engine. Thus, depression of the accelerator
20. pedal by the operator will provide him with the additional power he requires and, simultaneously, the governor will balance out at this higher setting.

- One of the problems encountered with conventional fuel injection systems of this type is that when an
25. engine is running at speeds above low idle, pedal effort

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increases with speed, requiring greater pedal effort by the operator to compress the governor spring to change the increased setting of the governor. This phenomenon is depicted in Figure 6 of the drawings by the curve labeled "Standard Governor".

- A solution to this problem comprises the utilization of a booster apparatus for applying an additional force to the governor spring, additive to the force applied to the spring by depression of the accelerator pedal by the operator, whereby the pedal effort is reduced. Such booster apparatuses have previously included means for communicating air intake manifold pressure or engine lubricating oil to the governor to counteract the opposing force of the governor spring.
10. To date, such booster apparatuses have been found to be unduly complex and expensive to manufacture and do not always ensure the precise booster and governor control effort required. In addition, leakage and related problems may be occasioned to affect further the precise control of the system. Also, only one specific magnitude of boost force is normally designed into conventional systems.

- According to this invention a booster apparatus comprises a control member, first means for
25. applying a first force to the control member to move it in a first direction, second means for applying a second force to the control member in opposition to the first force, and third means for applying a third force to the control member
30. additive to the second force to assist the second means in urging the control member in a second direction opposite to the first direction, characterized in that the third means includes control means for controlling the third force in a predetermined ratio with the second
35. force.

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Preferably, third means includes an actuating chamber, the control means maintaining a fluid pressure in the chamber proportional to the value of the second force.

5. The booster apparatus of this invention provides a non-complex and economical means for precisely controlling a booster assist to an operator of a vehicle upon his depression of an accelerator pedal. The operator is thus able to move the governor means
10. to a higher setting to provide the additional power needed without requiring an unduly high pedal effort. The response from throttle to fuel change is with the governor adjusting to the higher setting, dictated by amount of depression of the accelerator pedal by the
15. operator. Also, the booster apparatus may be designed to have its boost force selected from a range of boost forces to effect a desired ratio in respect of the input force of the operator.

- Two examples of apparatus according to the invention
20. will now be described with reference to the accompanying drawings, in which:-

- Figure 1 is a sectional view through a fuel control system employing a first booster apparatus embodiment of the present invention therein, shown in a starting
25. stage of operation;

Figure 2 is an enlarged sectional view of the booster apparatus generally taken in the direction of arrows II-II in Figure 3, shown in a second stage of operation;

30. Figures 3 and 4 are cross-sectional views, taken in the direction of arrows III-III and IV-IV, respectively, in Figure 2;

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FIG. 5 is a view similar to FIG. 2, but illustrates a second booster apparatus embodiment of the present invention shown in a starting stage of engine operation; and

5 FIG. 6 graphically depicts throttle torque and oil pressure curves.

FIG. 1 illustrates a fuel control system 10
10 mounted in a housing 11, secured in a conventional manner on an internal combustion engine. Fuel control system 10 comprises a fuel control means 12 for controlling supply of fuel to the fuel injection nozzles of a diesel engine,
15 governor means 13 for automatically controlling supply of such fuel in response to the speed of the engine, and operator input means 14 for selectively applying a first force to governor means 13 to override a counteracting force of the governor means.
20 This invention relates to a booster means 15 for applying a second, downward force to governor means 13 which is additive to the first downward force applied thereto by operator input means 14, to aid the operator input means in opposing the force of
25 governor means 13 to selectively give the governor means a higher setting.

Partially illustrated fuel control means 12 comprises a bellcrank 16, pivotally mounted by a pin 17 on a bracket 18 secured to housing 11, for
30 pivoting in response to axial movements of an output shaft 19, slidably mounted in bracket 18. A first

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arm 20 of bellcrank 16 is pivotally connected at a ball and socket connection 21 to a lower end of shaft 19 whereas a second arm 22 of bellcrank 16 is suitably connected by a standard linkage (not shown) to a series of fuel injection pumps (not shown) of the engine to control the quantity of fuel injected into the combustion chambers thereof.

The upper end of output shaft 19 is connected to a reciprocal riser shaft 23 at a bayonet connection 24 to reciprocate simultaneously therewith. As shown in FIGS. 1 and 2, governor means 13 comprises a carrier 25, including an annular gear member 26. A pair of flyweights 27 are each pivotally mounted on carrier 25 by a pin 28 whereby upon rotation of carrier 25 about a fixed bearing 29 flyweights 27 will pivot radially outwardly.

A first gear 30 is formed on member 26 and meshes with a second gear 31. Gear 31 is suitably attached to an engine-driven input shaft 32 whereby rotation of gear 31 will, in turn, rotate gear 30 and carrier 25.

Referring to FIG. 2, rotation of carrier 25 to pivot flyweights 27 radially outwardly will engage an arm 33 of each of the flyweights with an annular race 34 of a thrust bearing assembly 35 which is mounted on riser shaft 23 to reciprocate therewith. A spring riser 36 is secured on riser shaft 23 by a cross pin 37 and has an upper end thereof mounted on an axial rod 38 at a flanged connection 39. Rod 38 is mounted within a stationary column 40 which may have its upper end suitably secured to housing 11 at a flared portion 41.

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A selected or balanced standard governor spring 42, mounted between riser 36 and booster means 15, is adapted to be compressed in an upward direction upon radial outward pivoting of flyweights 27 and to be compressed in the opposite direction by the additive forces of operator control means 14 and booster means 15. In particular, operator control means 14 (FIG. 1) comprises a control lever 43 which is mounted on a pivotal shaft 44 so that a pair of bifurcated arms 45 thereof engage an upper end of a cylinder 46 which is reciprocally mounted on column 40. Shaft 44 is suitably connected to an operator-controlled accelerator pedal or the like (not shown) to selectively pivot control lever 43 clockwise in FIG. 2 to move cylinder 46 downwardly against the opposed biasing force of spring 42. A passage 47, having an orifice 48 defined at a lower end thereof, is formed in column 40 to communicate engine oil therethrough.

In the illustrated upward or open position of cylinder 46 in FIG. 2, upon startup of the associated engine, engine oil is free to dump into the confines of housing 11 and back to sump, via vent means comprising a pair of flat shoulders 49 formed on the otherwise cylindrical column 40 (FIG. 3). As further shown, governor spring 42 continuously biases an annular valve ring 50 of a valve means 51 against a seat defined on an underside of a flange 52 of cylinder 46.

When cylinder 46 moves downwardly from its FIG. 2 towards its FIG. 1 position when urged by the operator through lever 45, the cylinder will cover shoulders 49 to define an actuating chamber 53 (FIG. 1) which receives pressurized oil from passage 47

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and orifice 48. Such oil is communicated from chamber 53 to an upper side of valve ring 50 via three slots 54 and an annular groove 55 formed in flange 52 of piston 46, as shown in FIG. 4.

5 As described more fully hereinafter, orifice 48 and valve ring 50 comprise control means 51 for continuously controlling oil pressure in actuating chamber 53 in a predetermined ratio relative to the input force of the operator at lever
10 45. When the oil pressure in chamber 53 exceeds a predetermined level, valve ring 50 will unseat to permit dumping of oil therepast, via slots 54 and groove 55, until the oil pressure again falls to such predetermined level. As shown by the "Boosted Governor"
15 curve in FIG. 6, when engine speed exceeds a speed of 1300 rpm, for example, the boost force provided by oil pressure in actuating chamber 53 which acts downwardly on cylinder 46 (FIG. 1) will increase relative to engine speed. As further shown in the
20 shaded area on this curve, the boost force can be varied selectively within the depicted range by suitably varying the design parameters of valve means 51 and related constructions.

 Thus, such boost force will be additive to
25 the force applied to cylinder 46 by operator control means 14 with the latter force being depicted by the vertical distance between the "Standard Governor" (depicting the opposing force of governor spring 42) and "Boosted Governor" curves in Fig. 6. Booster means
30 15 thus automatically aids the operator in his application of pedal effort to change and maintain the setting of governor means 13 at higher engine

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speeds. For example, in FIG. 6 it can be seen that when the engine is running at 2000 rpm, the throttle shaft torque of approximately 1.93 N·m must be overcome by pedal effort without the use of booster means 15. However, the addition of booster means 15 to the system reduces such pedal effort to approximately 1.12 N·m.

It should be noted in FIG. 2 that a snap ring 56, mounted on a lower end of column 40, provides a stop means 57 for setting the maximum downward movement of cylinder 46 relative to column 40.

FIG. 5 illustrates a second booster means embodiment 15a of the present invention wherein corresponding structures are depicted by identical numerals, but wherein numerals depicting modified constructions are accompanied by an "a." Booster means 15a comprises a cylinder 46a slidably mounted on a slightly modified column 40a for upward movement from its extreme downward or engine startup position illustrated in FIG. 5. In this position of cylinder 46a, engine oil may be communicated to an actuating chamber 53a, defined by column 40a and cylinder 46a, via passage 47 and orifice 48. Upon initial running or low idling of the engine, lever 45 is positioned to raise cylinder 46a to an extreme upward position 46a'. Chamber 53a will be thus vented to dump oil into the confines of housing 11, via vent means comprising a shoulder 49a formed on cylinder 46a and one or more slots 58 formed on column 40a. This

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initial dumping phase of operation is depicted between approximately 500 rpm and 1300 rpm on the "Boosted Governor" curve in Fig. 6.

Upon depression of the accelerator pedal by the operator to actuate operator control means 14 (FIG. 1), arms 45 of the governor control lever will move cylinder 46a down towards its FIG. 5 position to thus isolate chamber 53a from slot 58. Booster means 15a and a control means 51a thereof will thereafter function substantially similar to booster means 15 and control means 51 of the FIGS. 1 and 2 embodiment to maintain the oil pressure and control means 51 in chamber 53a substantially constant under control of orifice 48 and a valve ring 50a. Valve ring 50a normally engages a seat defined on an underside of a flange 52a of cylinder 46a, under the biasing force of governor spring 42. During operation, oil pressure is communicated to valve ring 50a via a plurality of ports 54a (one shown) and an annular groove 55a, both defined in cylinder 46a. Thus, should the oil pressure in chamber 53a exceed a predetermined level, valve ring 50a will unseat to dump-out such excess pressure.

25

The above-described booster means 15 and 15a are particularly useful in association with the fuel control system of an internal combustion engine, such as a diesel engine typically run at rated speeds of from 1200 to 2400 rpm. It will be understood by

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those skilled in the arts relating hereto that such booster means are equally adapted for applications whereby a boosting force is desired to supplement an input force for controlling movement of a member
5 which has an opposing force applied thereto.

Referring to FIGS. 1-4, when the engine is at rest and not running, governor spring 42 will expand to pivot flyweights 27 to their inactivated and upright positions illustrated in FIG. 1. Upon
10 startup and idling of the engine between 650 rpm and 1200 rpm, for example, flyweights 27 will pivot to the positions shown in FIG. 2 to control the engine speed by raising riser shaft 23 and attached shaft 19 to pivot bellcrank 16 clockwise in FIG. 1. The
15 standard linkage system connecting bellcrank 16 with the fuel injection fuel pumps of the engine will thus meter the desired quantity of fuel to the cylinders of the engine during idling thereof.

Simultaneously therewith, arms 33 of
20 flyweights 27 will, through spring 42, maintain cylinder 46 in an upper position determined by lever 43 illustrated in FIG. 1, whereby booster means 15 is deactivated by permitting any engine oil communicated thereto via passage 47 and orifice 48 to
25 be dumped within the interior of housing 11. This desired deactivation of booster means 15 will ensure stability of the system at low idle and constant rpm with either hot or cold oil. This condition of engine operation is reflected in FIG. 6 approximately
30 between the 500 to 1300 rpm portion of the "Boosted Governor" curve.

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As further shown in FIG. 6, it can be seen that without the aid of booster means 15 that the pedal effort required to overcome the opposing force of governor spring 42, as reflected by the "Standard Governor" throttle torque curve, increases sharply as engine speed increases. However, with the utilization of booster means 15, depression of the accelerator pedal under a first or operator imposed force will further function to condition the booster means for applying a second, additive force to oppose the opposing force of governor spring 42. As described above, upon depression of the accelerator pedal by the operator, arms 45 of governor control lever 43 will move cylinder 46 downwardly to cover shoulders 49 whereby an oil pressure build-up will occur in booster actuating chamber 53. Such oil pressure is also communicated to valve ring 50, via ports 54 and groove 55. As further shown in FIG. 6, booster means 15 is predesigned to allow oil pressure communicated to groove 55 to apply an opening force F_V in opposition to a force F_S of governor spring 42 whereby oil in chamber 53 is dumped past valve ring 50 in an amount substantially equal to the flow of oil through orifice 48 and to chamber 53.

The effective area of cylinder 46 is subjected to a booster force F_B which is additive to operator input force F_L to counteract the opposing force F_S of governor spring 42. The effective area of cylinder 46 is precalculated and formed to assist the operator by reducing throttle pedal effort at higher speeds, as reflected by the general linear increase of the "Boosted Governor" curve in FIG. 6

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upon increase of engine speed over 1300 rpm. For example, at 2,000 rpm pedal effort would require a throttle torque approximating 1.93 N·m without the aid of booster means 15. However, the pressurization

5. of actuating chamber 53 to simultaneously apply a boost force F_B of approximately 0.78 N·m reduces pedal effort approximately forty-on percent. Thus, the operator is enabled to balance force F_S of governor spring 42 with minimal effort.
10. It should be further noted in Figure 6 that the "Engine Pressure" (oil) curve rises upon starting of the engine and generally levels-off when engine speed exceeds 1200 rpm. Likewise, the "HSG Pressure Before Orifice" curve indicates that the oil pressure in passage 47,
15. on the downstream side of orifice 48, levels-off at approximately 315 KPa when engine speed exceeds 1200 rpm. This relatively stable level of oil pressure can be utilized to properly size orifice 48 for effective operation of booster means 13.
20. Considering forces F_L , F_B , and F_S in the context of a free body diagram, the sum of such forces equals zero to achieve static equilibrium.

Therefore, it can be further seen that

$$F_L + F_B = F_S$$

25. Furthermore,

$$F_V = P_G \times A_V$$

where:-

P_G = the oil pressure in groove 55, and

30. A_V = the effective area of valve ring 50

Additionally,

$$F_B = P_G \times A_B$$

where:-

35. A_B is the effective area of cylinder 46 for the applying force F_B thereon

Also, the summation of forces on the valve ring 50 must equal zero therefore $F_V = F_S$.

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$$\text{Now } P_G = F_V/A_V$$

$$= (F_B + F_L)/A_V$$

$$= (P_G \times A_B + F_L)/A_V$$

5. therefore $P_G (1 - A_B/A_V) = F_L/A_V$

$$\text{therefore } P_G = F_L/(A_V - A_B)$$

so that P_G is proportional to F_L and thus

F_B is also proportional to F_L .

It should be understood that the effective

10. area of grooves 55 and 55a could be increased or decreased in respect to chambers 53 and 53a, respectively, to provide various ratios effecting corresponding changes in boost force F_B . The shaded area in Figure 6 depicts such variations of boost forces which provide
15. for the pre-selection of a boost force to effect a predetermined ratio in respect to the input force of the operator to aid operator comfort.

The Figure 5 booster means embodiment 15a functions substantially similar to above-described booster means 15.

20. As described above, the primary difference between the two embodiments is that upon startup of the engine in Figure 5, flyweights 27 will pivot outwardly to move cylinder 46a upwardly to, in turn, dump oil from chamber 53a to the interior of housing 11, via shoulder
25. 49a and slot 58. This initial dumping of oil from chamber 53a is reflected by the "Boosted Governor" curve in Figure 6, between approximately 500 and 1300 rpm of engine operation, i.e., low idle. When the operator depresses the accelerator pedal to further increase
30. engine speed, the booster force curve will generally assume the operational characteristics shown in Figure 6.

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CLAIMS

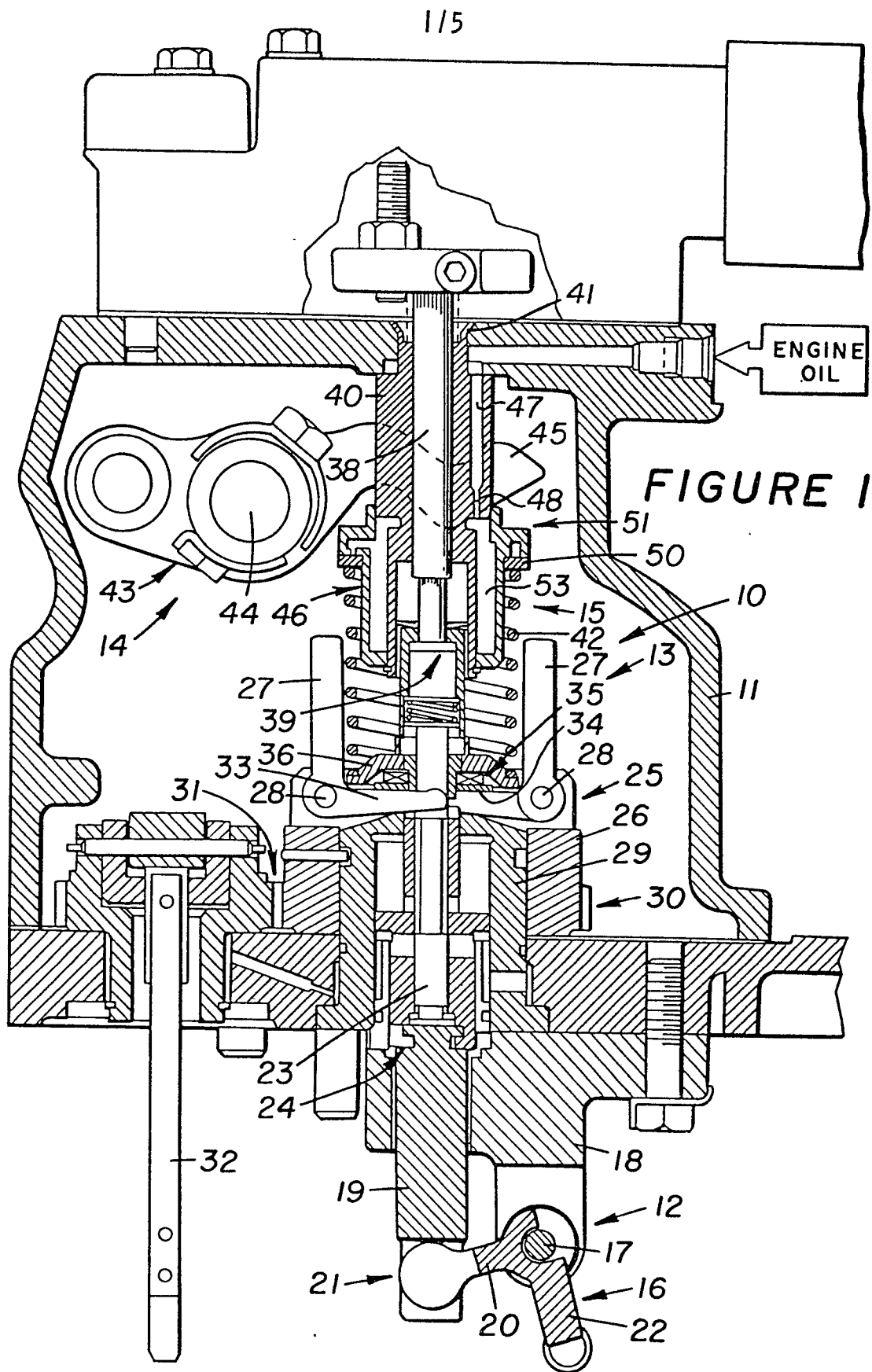
1. A booster apparatus having a control member (23), first means (13) for applying a first force to the control member (23) to move it in a first direction, second means (14) for applying a second force to the control member (23) in opposition to the first force, and third means (15) for applying a third force to the control member (23) additive to the second force to assist the second means (14) in urging the control member (23) in a second direction opposite to the first direction, characterized in that the third means (15), includes control means (51) for controlling the third force in a predetermined ratio with the second force.
2. Apparatus according to claim 1, wherein the third means (15) includes an actuating chamber (53), the control means (51) maintaining a fluid pressure in the chamber (53) proportional to the value of the second force.
3. Apparatus according to claim 2, wherein the control means (51) includes orifice means (48) for communicating a predetermined fluid flow to the actuating chamber (53).
4. Apparatus according to claim 2 and claim 3, wherein the control means (51) further includes valve means (50) for venting fluid from said actuating chamber (53) to maintain the fluid pressure therein substantially constant for a given value of the second force.

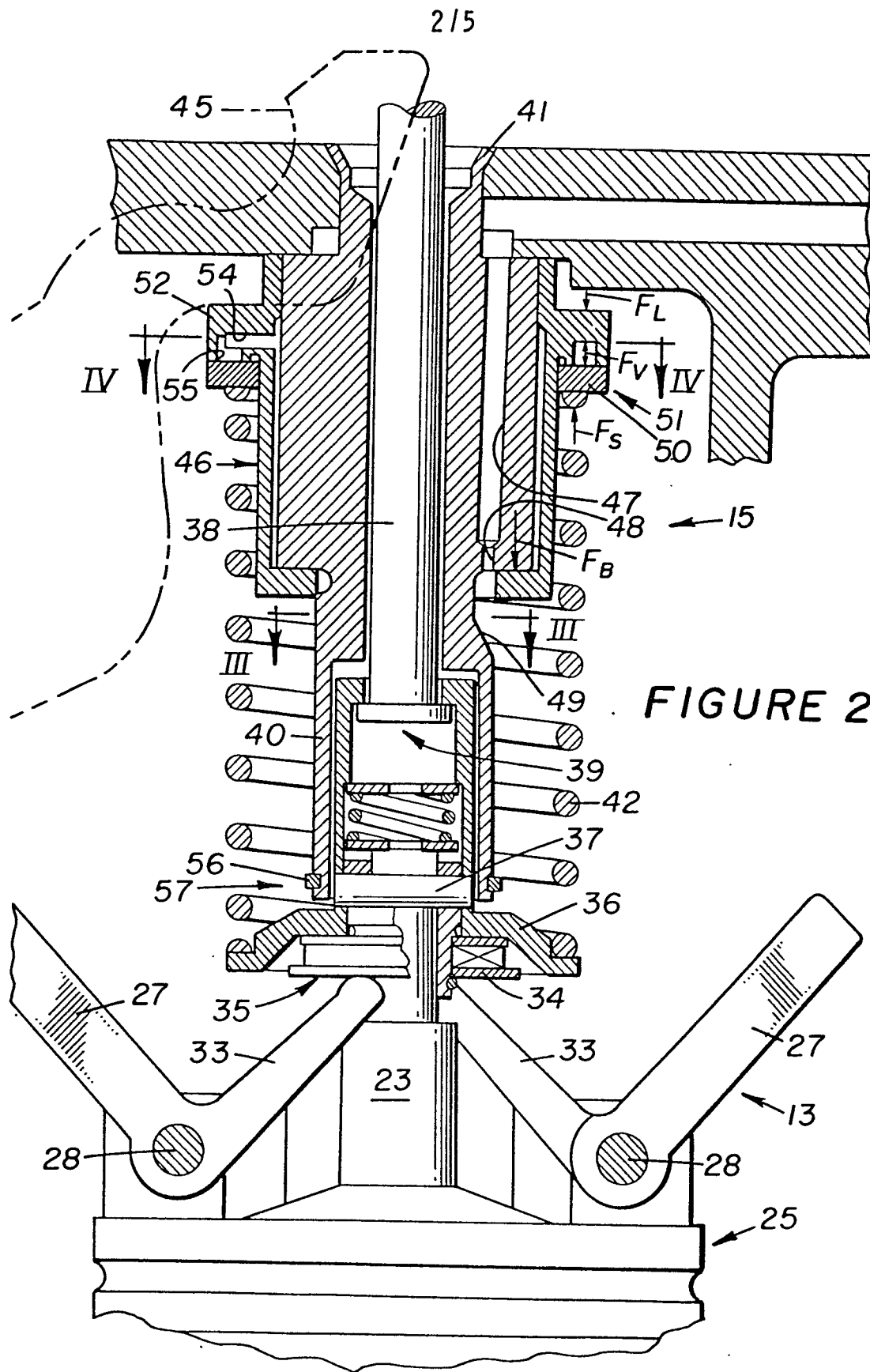
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5. Apparatus according to any of claims 2 to 4, wherein the third means (15) further includes a cylinder (46).
6. Apparatus according to claim 5, wherein the cylinder (46) is reciprocally mounted on a column (40) to define
5. the actuating chamber (53) therewith in, the second means (14) engaging the cylinder (46) to move it in the second direction, and the first means (13) including spring means (42) for urging the cylinder (46), in the first direction.
10. 7. Apparatus according to claim 5 or claim 6, wherein the valve means includes a ring (50) normally engaging a seat defined on the cylinder (46), and further including means (54, 55) for communicating fluid pressure from the actuating chamber (53) to the ring (50).
15. 8. Apparatus according to any of claims 6 or 7, further including stop means (57) for limiting reciprocal movement of the cylinder (46).
9. Apparatus according to any of claims 2 to 8, further including vent means (49) for opening the chamber (53)
20. to vent fluid pressure from the actuating chamber (53).
10. Apparatus according to claims 5, 6 and 9, wherein the vent means (49) is defined between the cylinder (46) and said column when the cylinder (46) is moved to a predetermined position on the column (40).
25. 11. Apparatus according to claim 10, wherein the vent means (49) includes at least one angled shoulder formed externally on the column (40).
12. Apparatus according to claim 9 wherein the vent means includes at least one shoulder (49a) formed
30. internally on the cylinder (46a) and at least one slot (58) formed externally on the column (40a).
13. Apparatus according to claim 1, wherein the third force applied to the control member (23) conforms substantially to the "Boost Force" curve shown in
35. Figure 6 of the accompanying drawings.

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14. A fuel control system having a booster apparatus according to any one of claims 1 to 14, a fuel control means (40) for controlling the supply of fuel to an engine, the first means (13) of the apparatus
5. comprising a governor (13) for automatically controlling supply of fuel to the engine by the fuel control means (12) in response to the speed of the engine, the second means (14) comprising operator input means (14) selectively applying the first force to the control
10. member (23) of the governor (13) to oppose a counter-acting force thereof, the third means (15) applying the second force to the governor (13), additive to the first force, to aid the operator input means (14) in controlling the governor (13).
- 15.





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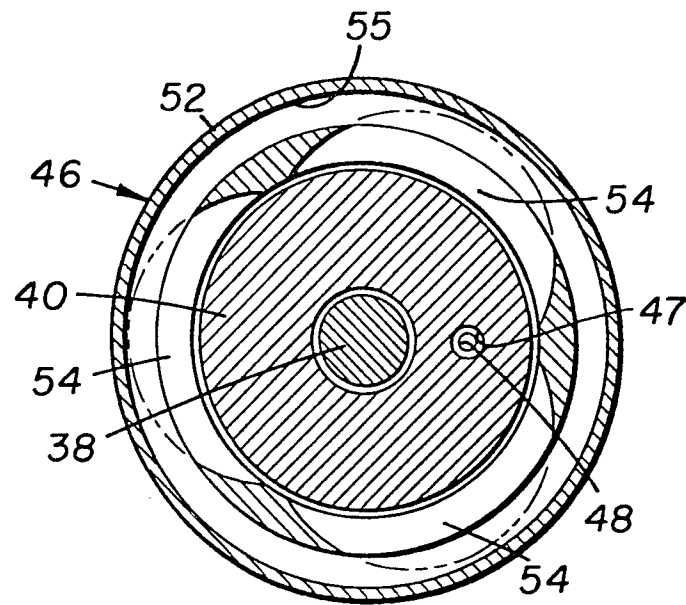


FIGURE 4

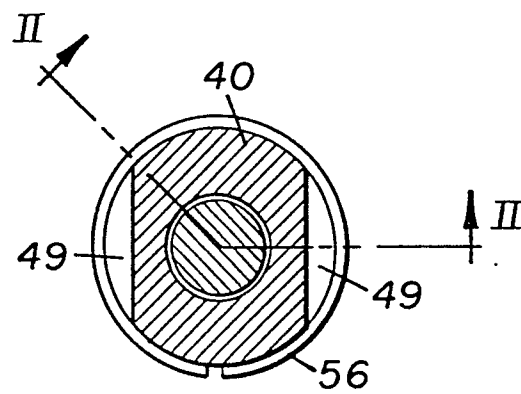
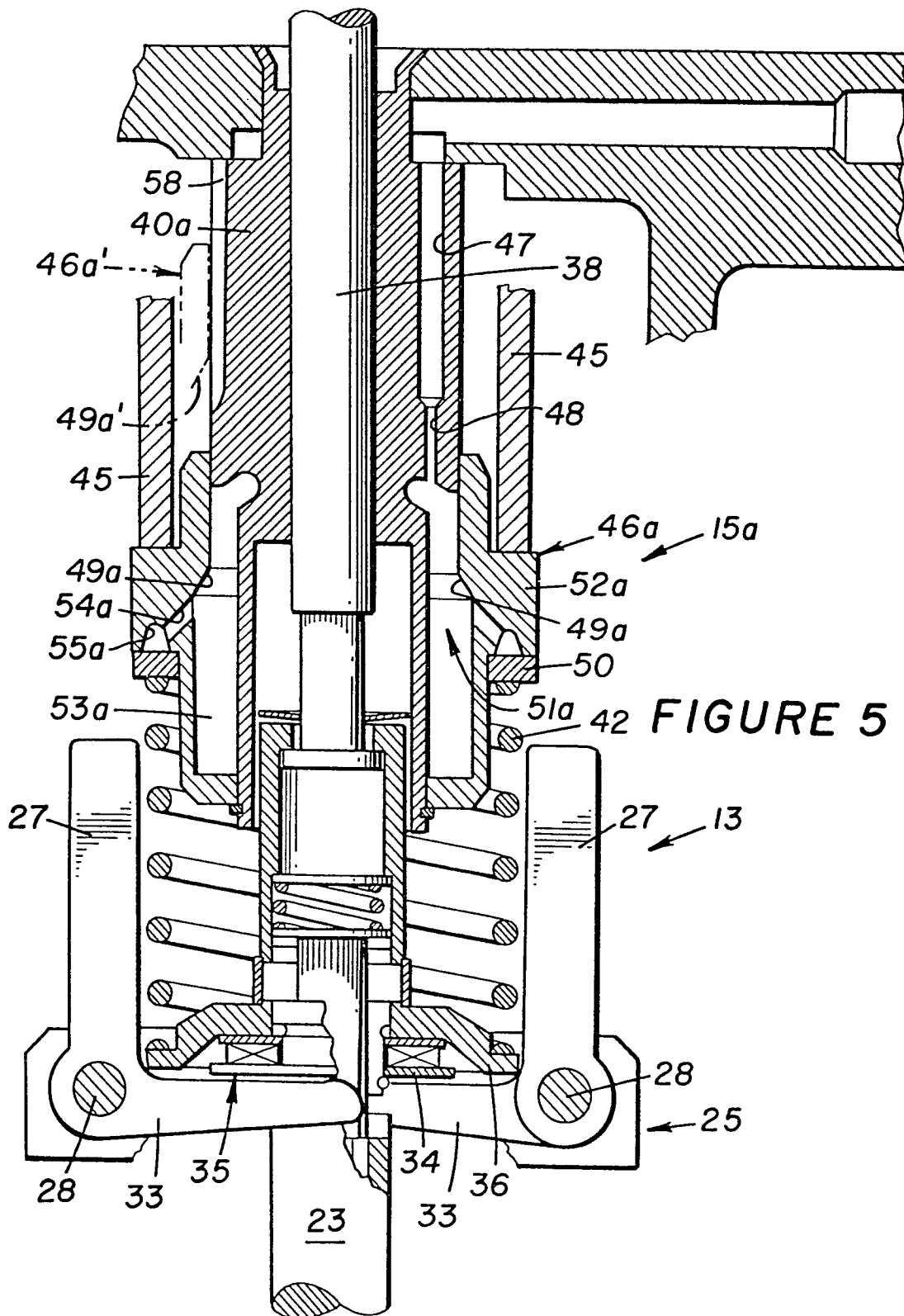


FIGURE 3

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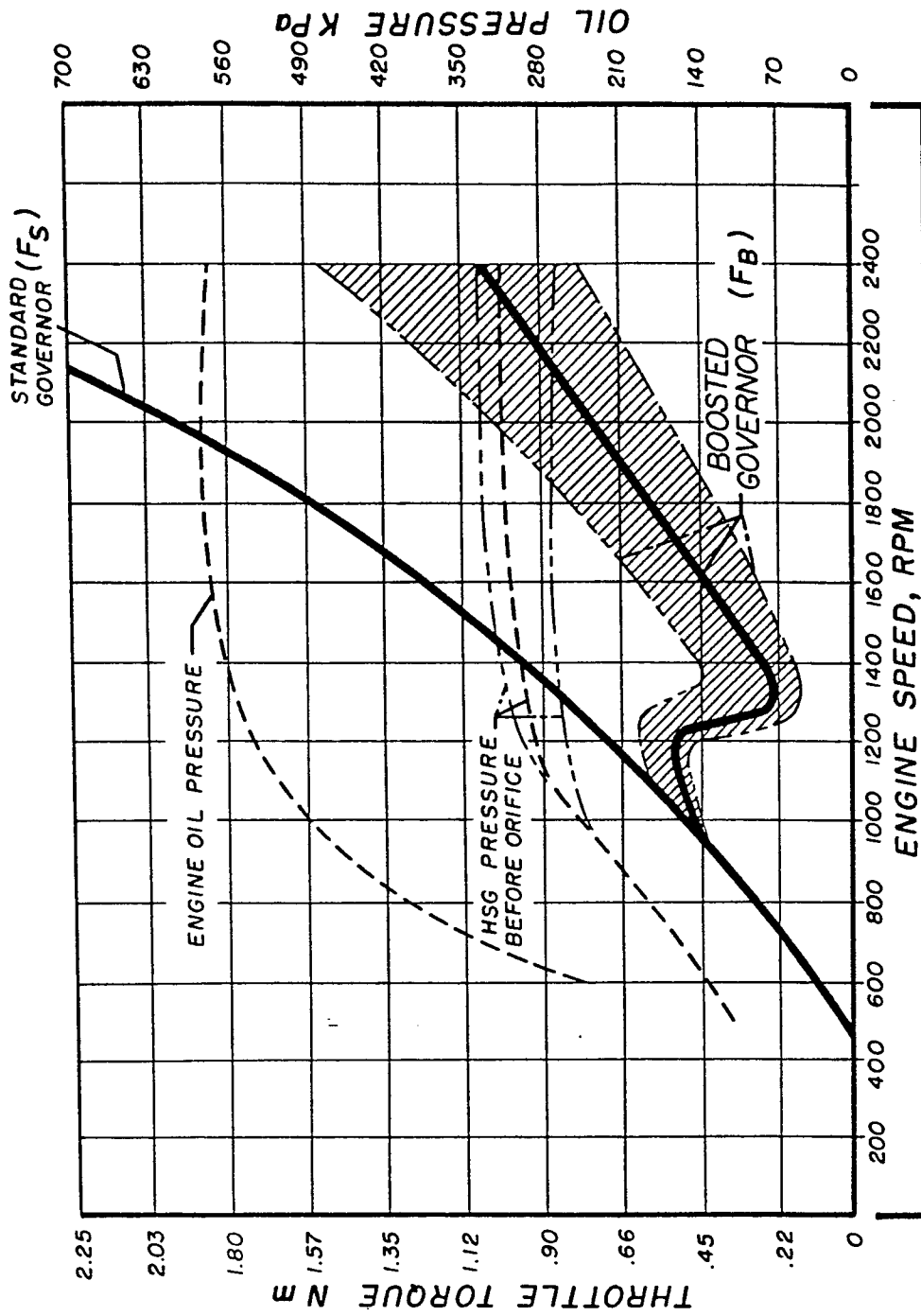


FIGURE 6



DOCUMENTS CONSIDERED TO BE RELEVANT			CLASSIFICATION OF THE APPLICATION (Int. Cl. 4)
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	
X	<u>US - A - 2 648 533 (MAXWELL)</u> * Column 1, line 55 to column 4, line 17; figures 1,2 *	1-6, 8,9, 13,14	F 02 D 1/08 1/04
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X	<u>US - A - 2 631 837 (ROBINSON et al.)</u> * Column 1, line 21 to column 4, line 44; figure 1 *	1-6,8, 9,13, 14	
	--		TECHNICAL FIELDS SEARCHED (Int. Cl. 3)
A	<u>US - A - 2 332 925 (MARTIN)</u>		
A	<u>US - A - 2 215 406 (PAXMAN)</u>		F 02 D F 02 M

			CATEGORY OF CITED DOCUMENTS
			X: particularly relevant A: technological background O: non-written disclosure P: intermediate document T: theory or principle underlying the invention E: conflicting application D: document cited in the application L: citation for other reasons
			&: member of the same patent family. corresponding document
The present search report has been drawn up for all claims			
Place of search	Date of completion of the search	Examiner	
The Hague	07-07-1981	BICHI	