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Sola

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[54] **CONTROL DEVICES**

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[52] **U.S. Cl.** **74/53**
[51] **Int. Cl.** **F16h 25/05**
[58] **Field of Search** .... 74/53, 569; 280/111; 123/139 AK

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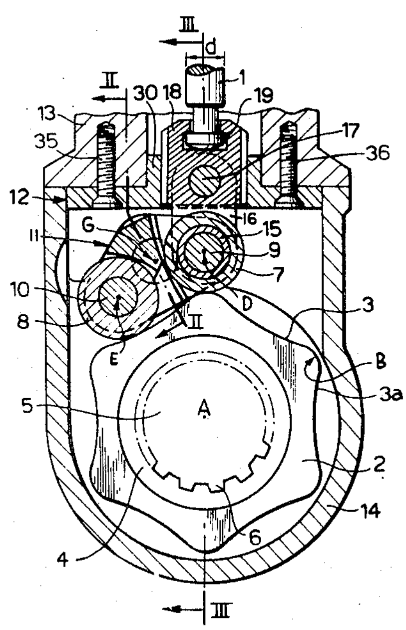
[57] **ABSTRACT**

A control device suitable for the fuel injection pump of an internal combustion engine is shown. The device comprises a rotatable cam having a plurality of symmetrical lobes and a cam follower having two rollers which are both in contact with the cam face. The cam follower is linked to an operating member of an associated pump. The rollers of the cam follower are relatively large to withstand the forces produced at high speed and the pivot for the cam follower is arranged so that it does not obstruct the space between the two rollers. The cam profile is so shaped that the two rollers are both always in contact with the surface of the cam so that there is no play in their movement.

14 Claims, 9 Drawing Figures



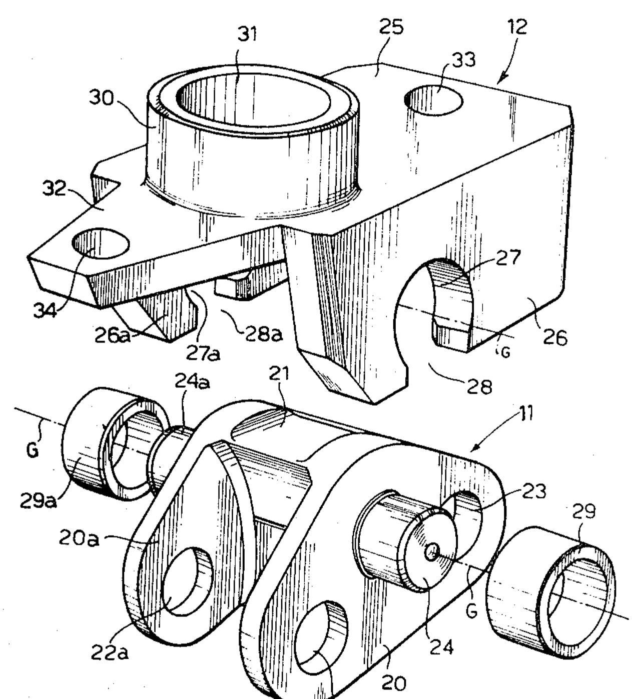


Fig. 1

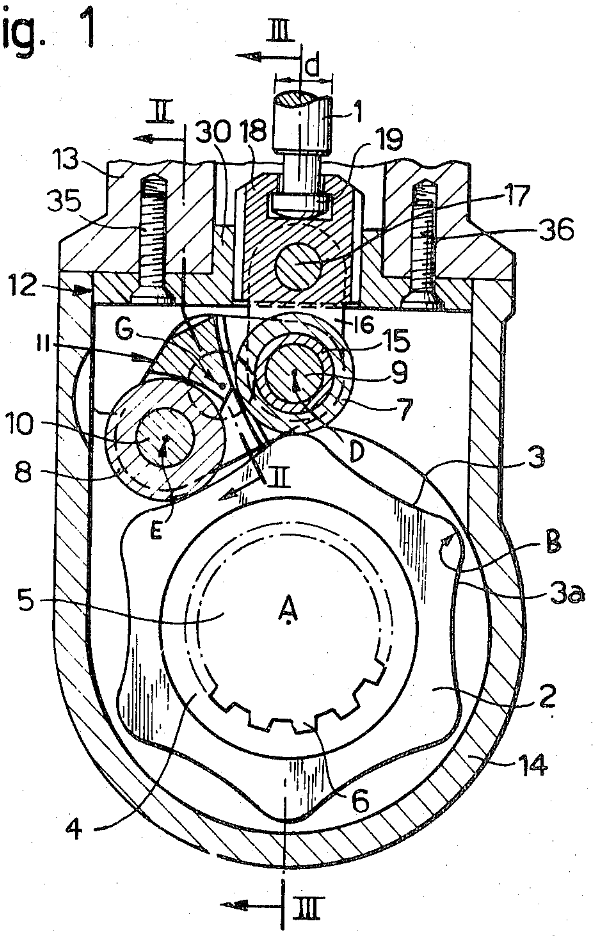


Fig. 2

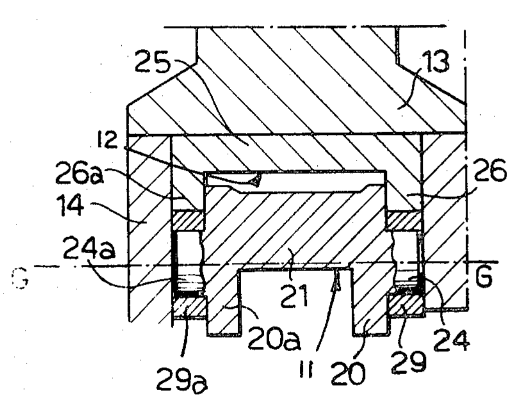


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Fig. 3

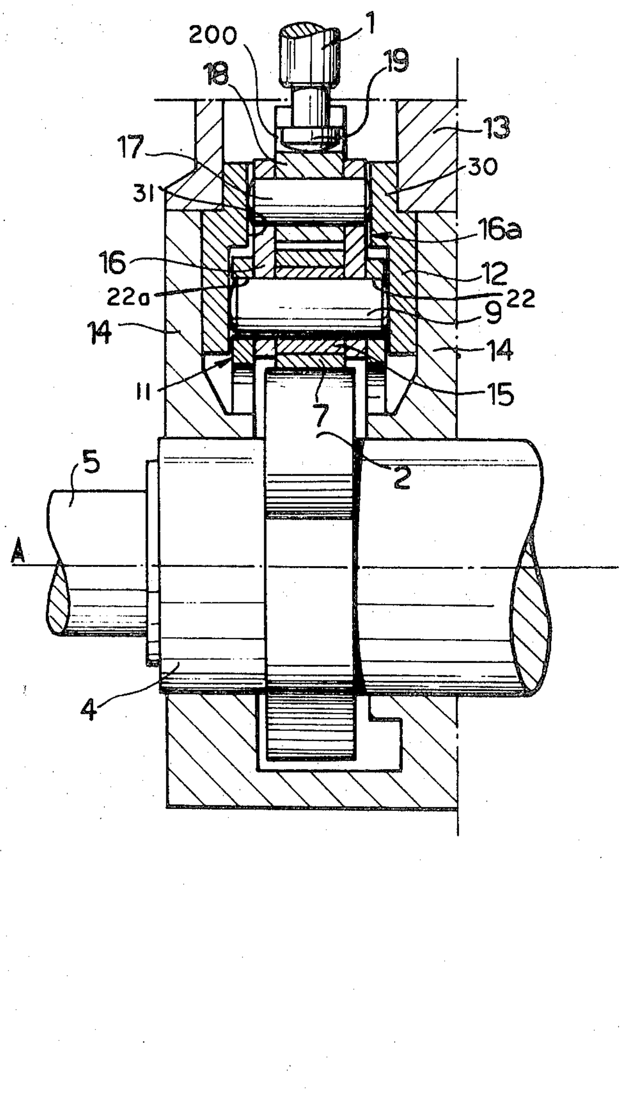

Fig. 4

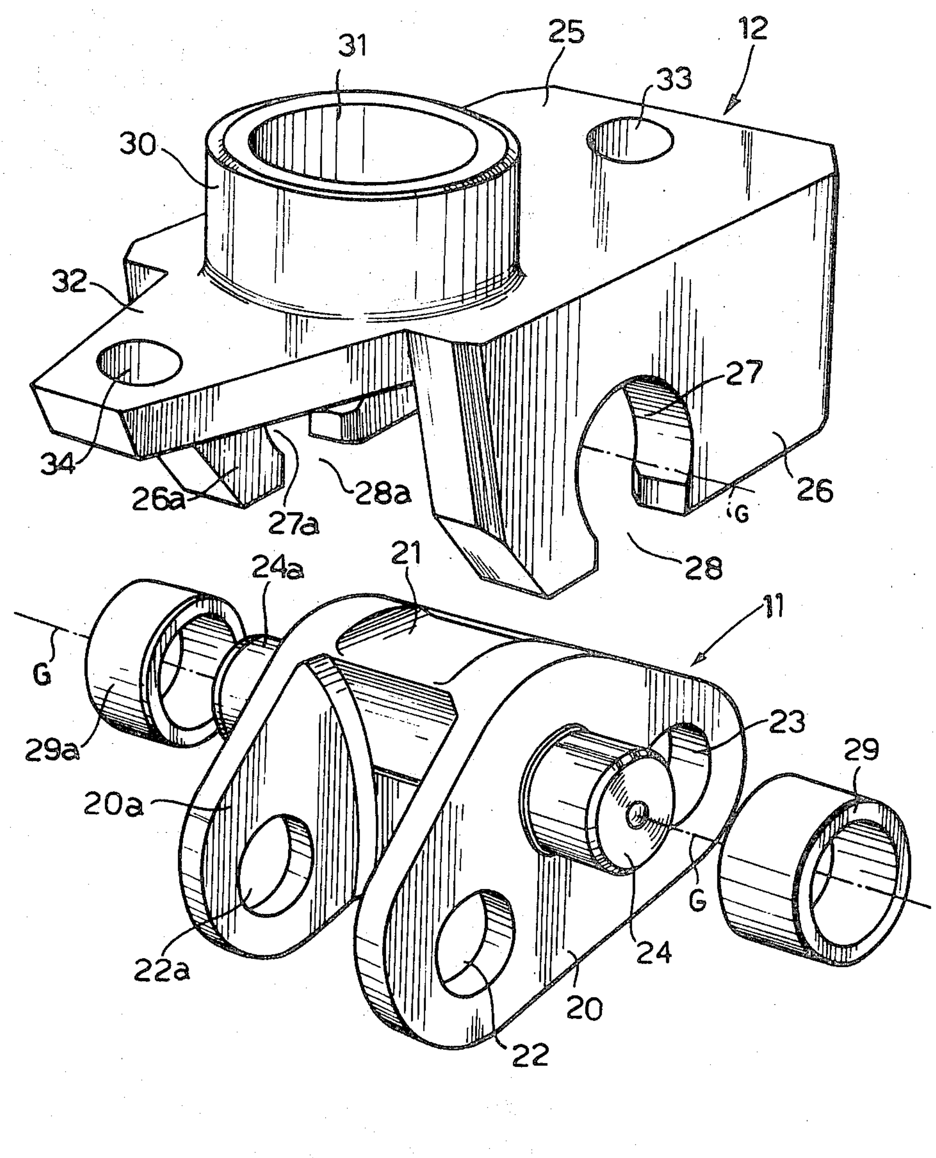
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Fig. 5

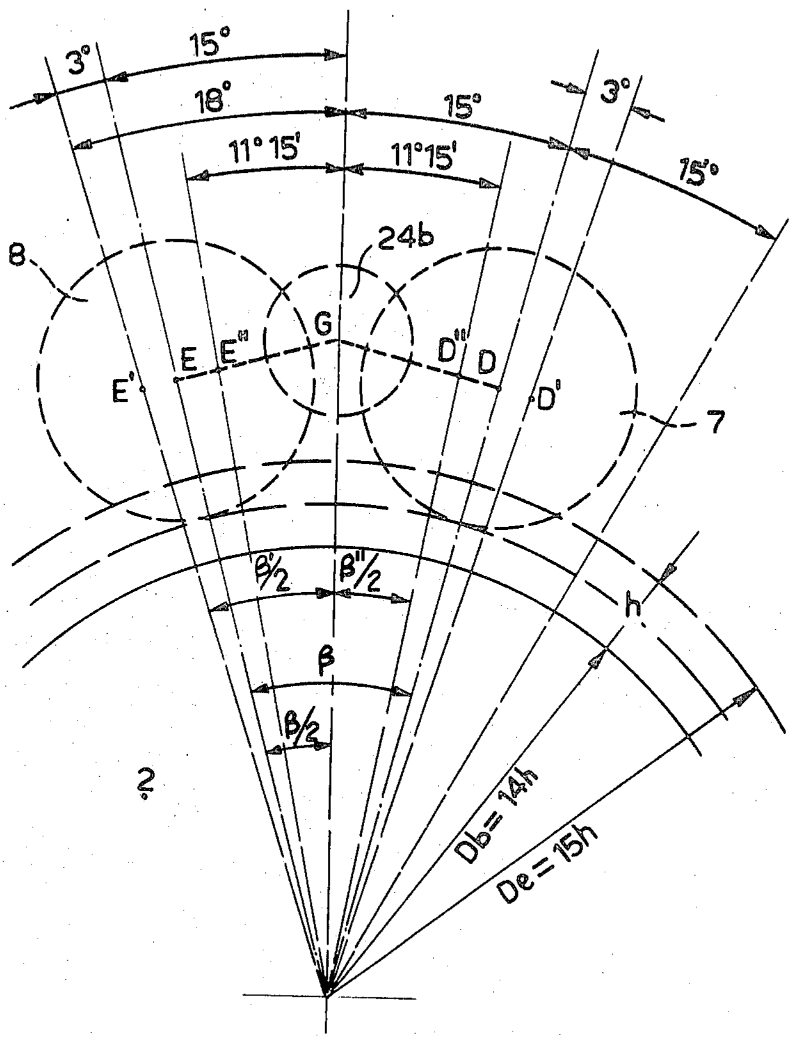


Fig. 6

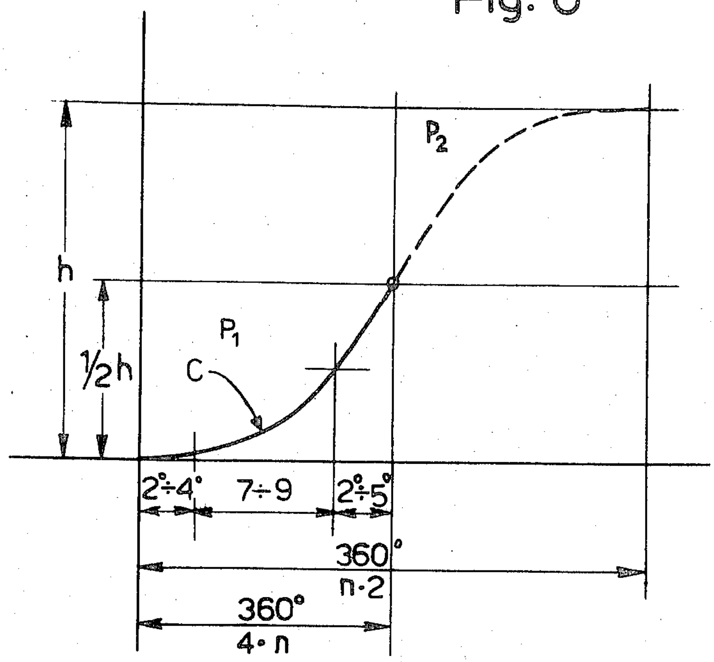
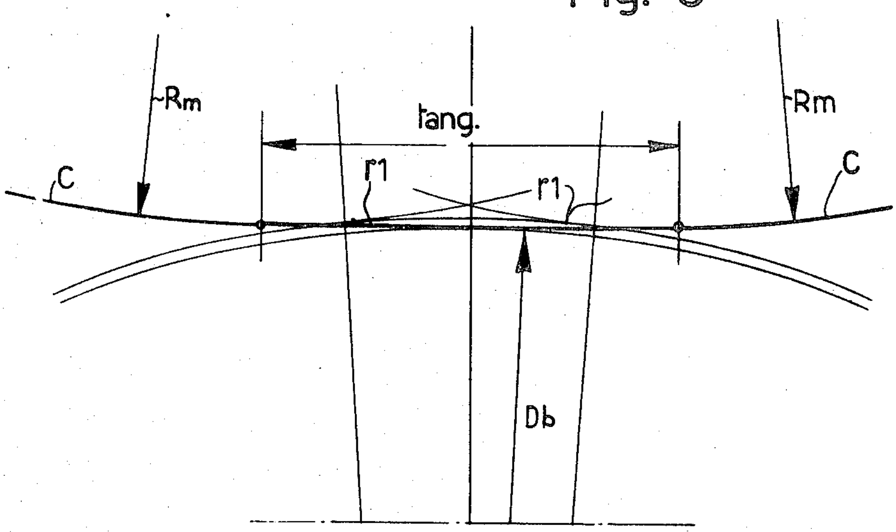


Fig. 9



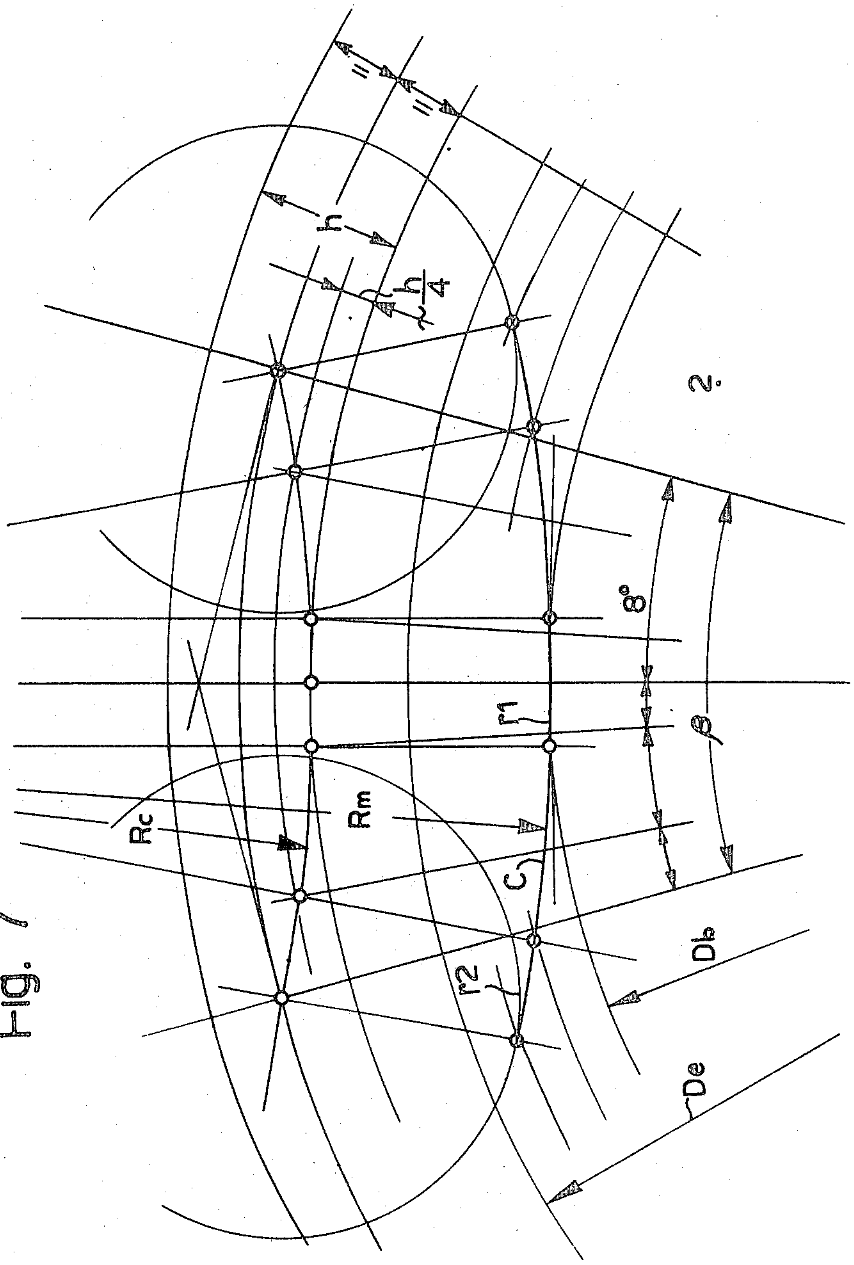
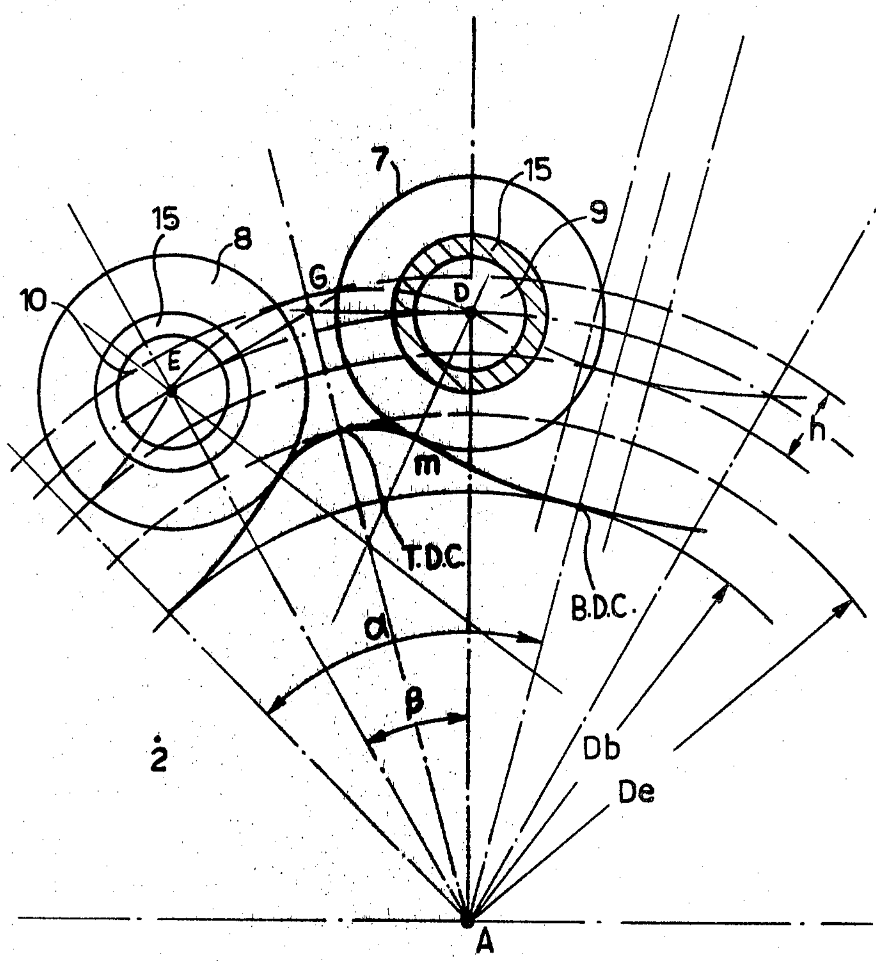


Fig. 7

Fig. 8



CONTROL DEVICES

BACKGROUND OF THE INVENTION

The present invention relates to a control device for a pump, such as a fuel injection pump for a multi-cylinder internal combustion engine. In particular, the invention relates to a control device of the type comprising a cam profile having a plurality of substantially symmetrical lobes, extending radially outwardly in a single plane perpendicular to the axis of rotation of the cam, and a cam follower having a pair of rollers or sliding blocks supported by a rocker arm, both rollers being engaged on respective opposed lobe faces of the cam. One roller or sliding block is directly associated with an operative element of the associated pump and is moved by the cam to produce a pumping stroke of this operative element and the other roller or sliding block operates as a return element to effect the return stroke of the operative element of the pump.

This general type of control device is already known in itself and forms the subject of other patents, such as Italian Pat. No. 759,541 and Italian Pat. No. 818,422, both in the name of the present Applicants.

It has been found that in cases where the control cam has to have five or more lobes, that is for the control of pumps to supply engines of five or more cylinders, there are constructional difficulties which must be overcome in order to achieve satisfactory functioning of these devices. With cams of five or more lobes the angle subtended at the center of the cam by each lobe is relatively small, with the result that the distance between the rollers which are arranged to engage with the flanks of the cam lobes also becomes rather small. In these circumstances there is not sufficient space between the two rollers to locate the pivot pin on which the rocker arm which supports the rollers is mounted.

A reduction in the diameter of the pivot pin on which the rocker arm is mounted is not possible because this pin has to be sufficiently robust to resist the stresses during operation of the pump. The forces borne by this pin are very great due to the fact that the rate of injection necessary for direct-injection engines of high specific power and high rate of rotation, generates maximum injection pressures which increase in proportion to the rate of rotation, such pressures, in the neighborhood of the pumping chamber may frequently be as much as 800 kg/cm_2.

Moreover, it is similarly not possible to reduce the diameter of the rollers carried by the rocker arm to make more space because these have to closely follow the profiles of the cam lobes, and the load on these rollers is also high. This load can reach, for example, 700 kgs on the main roller connected to the pumping element. From this it follows that the diameter, especially the diameter of the main roller, cannot be reduced, partly because it is necessary to keep the specific contact pressures within certain limits on the flanks of the cam lobes, and partly because, with the heavy load and with a high rate of rotation of the rollers, it is necessary to interpose a floating bush between the roller and its mounting pin.

Even the solution proposed in the above mentioned Italian Pat. No. 818,422 by the same Applicant, of eliminating one roller, that is the pump return roller, and of substituting for it a sliding block, has not proved satisfactory in practice in the case of a cam having five or more than five lobes. This is because it is necessary

to eliminate the wear on the cam caused by the sliding friction between such a block and the flanks of the lobes during the periods of the reciprocating action of the rocker in which the negative inertia forces of the whole system which cause the return action of the pump are opposed by the sliding block and the cam lobe with which it is in contact. Because of these forces the end of the rocker arm, opposite that which is associated with the operative element of the pump, is pressed against the flanks of the cam lobes with such loads as to make it absolutely necessary to minimize the friction on the lobe. This can only be achieved by the use of a roller rather than a sliding block in this position. In brief, it is essential to have rolling contact rather than sliding contact even between the two elements which are subject to the least loads because these loads are nevertheless great enough to cause an undesirable amount of wear on the cam if sliding contact is used.

Moreover, in practice, such control devices for injection pumps designed to control engines of six or more cylinders, which have been designed according to known criteria, have had disadvantages in regard to the profile of the cam lobes, the number of which is equal to the number of cylinders of the engine for which the pump is intended. These disadvantages arise mainly because, in order to achieve a satisfactory rate of fuel injection the radii of the cam profiles for each lobe are extremely small and difficult to manufacture with normal mechanical machining techniques. These profiles also allow some slackness of the operative element of the pump in relation to the control and return rollers, particularly at the acceleration reversal points of the pumping element and hence at the reversal points of the inertia forces.

OBJECTS OF THE INVENTION

It is an object of this invention to provide a solution to the problem of mounting the rocker arm member which will allow it to support rollers, both for control and return of the pumping element, of a sufficiently large diameter to bear the specific contact pressures upon the profile of the cam.

It is another object of this invention to provide a cam profile suitable for guiding rollers without any slack points and which can be readily manufactured using normal mechanical machining techniques.

SUMMARY OF THE INVENTION

According to the present invention there is provided a control device for a fuel injection pump of a multi-cylinder internal combustion engine, of the kind having a cam with a plurality of lobes each lobe being symmetric about a radial line through the top dead center of the lobe, a cam follower having two rollers both in contact with the profile of the cam, the rollers being mounted at respective ends of a rocker arm which is supported for pivotal movement about an intermediate point midway between the roller axes to a support member attachable to the casing of an associated pump, one of the two rollers being connectable by means of a connecting rod, to an operating member of an associated pump, in which the pivotal support for the said rocker arm does not extend into the region between the two rollers, and in which the part of the profile of each lobe of the cam between the top dead center position and the half lift position of the cam follower is determined to be the appropriate envelope of the successive positions of the one roller when the other roller moves over the part of the profile of the cam between the bottom dead center position and the half lift position of the said cam follower.

It will be appreciated that devices constructed in accordance with this invention should find particular utility in operating the pumping elements of injection pumps having six or more lobes, that is for engines of six or more cylinders.

Various further features and advantages of the invention will become apparent from a consideration of the following description, with reference to the accompanying drawings, which is presented by way of non-restrictive example.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a diagrammatic side view, partially cut away, of part of a pump having a control device constructed according to the invention;

FIG. **2** is a section taken along the line II-II of FIG. **1**, illustrating particularly the assembly of the rocker cam pivot pins;

FIG. 3 is a section taken along the line III—III of FIG. 1;

FIG. **4** is an enlarged perspective view of a rocker arm and support suitable for use in embodiments of the invention;

FIG. **5** is a diagram illustrating the position of the rollers and of the rocker arm pivot pins in relation to a cam with five lobes, and to a cam with six lobes;

FIG. **6** is a qualitative diagram of the rate of motion of a cam follower with respect to the cam profile according to the invention;

FIG. **7** is an enlarged diagrammatic view of part of the cam profile showing the main and reversal rollers of the pumping element in a first position corresponding to the half lift position of the cam follower along the stroke of the pumping element;

FIG. **8** is a diagrammatic view of the cam profile with the control and reverse rollers of the pumping element in a second position, again corresponding to the half lift position of the cam follower along the stroke of the pumping element; and

FIG. **9** is an enlarged diagrammatic view of the cam profile showing the position corresponding to the bottom dead center position of the cam follower.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

In all the Figures referred to above, the elements which correspond with each other are indicated with the same reference numbers. Referring now to FIG. **1** there is shown the lower part **1** of a single pumping element of an injection pump controlled by a cam **2** with six equal lobes, each lobe having respective flanks **3** and **3***a* on either side of a lobe peak B, the profiles **3** and **3***a* being symmetrical about a radial line AB from the axis A of rotation of the cam to the peak B.

The cam **2** is borne on a sleeve **4** which is driven by a shaft **5** via a claw clutch **6**, which may, for example, be helicoidal, to allow for adjustment of the angular position of the cam **2** in relation to the control shaft **5.** Upon the periphery of the cam **2** there rest two rollers, **7** and **8** respectively, mounted for rotation upon axles **9** and **10** of axes, D and E respectively, of which are parallel to the axis A of the shaft **5** and supported by a rocker arm **11** which in its turn is pivotally mounted upon a support. The rocker arm **11** is mounted for pivoting movement about an axis **9.** The support **12,** together with the casing **13,** forms the upper closure of the casing of the pump **14.** Between the roller **7** and its axle **8** there is inserted a floating bush **15.**

The connection between the roller **7** and the pumping element **1,** as can also be seen in FIG. **3,** is achieved by means of connecting rods **16** and **16***a* which are joined, at one end, to the axle **9** of the roller **7** and at the other end to a pin **17** which joins them to a cursor **18,** which in its turn, is connected to a mushroom head **19** of the pumping element **1.** The mushroom head **19** is located in a groove **200** of the cursor **18** so as to allow for possible rotation of the pumping element **1** in relation to the cursor **18,** and so as to maintain a rigid axial coupling with the cursor **18.**

With particular reference to FIG. **4,** the structure of the rocker arm **11** and of the related support member **12** by means of which it is attached to the casing of the pump14 will be described.

As can be seen from FIG. **4** the rocker arm **11** comprises two generally plate-like elements **20** and **20***a*, spaced from each other by a distance which is equal at least to the length of the rollers **7** and **8**; these elements **20** and **20***a*, are shaped in the form of an isosceles triangle with rounded vertices, and are rigidly connected by a bridge **21** which leaves sufficient space free between the elements **20** and **20***a* for the rollers **7** and **8** to be housed. The rollers **7** and **8** are mounted on axles **9** and **10** which are inserted into the holes **22–22***a* and **23–23***a* (the latter not visible in FIG. **4**).

On the outer facing wall of each plate **20** and **20***a* there is a stub shaft or pin **24** and **24***a*, aligned with each other and with the axis G about which the rocker arm **11** is pivoted. As can be seen from the structural arrangement described above, the rocker arm **11** can carry two rollers **7** and **8** of a diameter which is large enough to bear the high pressures on the profile of the cam, and at the same time allows the use of pivot pins **24** and **24***a* which, while not interfering with the rollers **7** and **8,** can nevertheless be of dimensions sufficient to increase to a satisfactory degree their resistance to the fatigue strains of the rocker arm during operation under heavy loads (more than 700 kgs), which loads are transmitted from the pumping element chamber on to the main or control roller **7.**

The rocker arm **11,** as can be seen from both FIG. **2** and FIG. **4,** is supported for pivoting motion about the axis G, by a support member **12** which comprises a substantially U shaped element with a base **25,** and limbs **26** and **26***a*. On the limbs **26** and **26***a* there are provided apertures **27** and **27***a* through the faces thereof and each aperture has an opening **28** and **28***a*. The diameter of the apertures **27** and **27***a* is greater than the diameter of the stub shafts or pins **24** and **24***a*, and the port of the openings **28** and **28***a* is only very slightly greater than the diameter of the said stub shafts or pins. This allows the rocker arm to be assembled by inserting the pin stubs **24** and **24***a* radially through openings **28** and **28***a*, and subsequently sliding the bushes **29** and **29***a* over the stub shafts or pins **24** and **24***a* and into the apertures **27** and **27***a* to hold the stub shafts and hence the rocker arm **11** in position. These bushes **29** and **29***a* whose outer diameter matches the inner diameter of the apertures **27** and **27***a*, prevent the rocker arm from slipping out of its supports.

The center portion, or base **25** is provided with a collar **30** having a central bore **31** through which passes the sliding block **18** connected to the pumping element **1**. The base **25** is moreover provided with a projecting tongue **32**, to assist in locating and securing the support member **12** in the upper part of the casing of the pump **14**. This can be effected because the base **25** and the tongue **32** are provided with holes **33** and **34** respectively for receiving screws **35** and **36** (see FIG. **1**) by means of which the support **12** with the rocker arm **11** mounted on to it, can be attached to the casing **13** of the pump **14**.

It will be understood, from the description above, that the structure also allows simple assembly and disassembly of the rocker arm and of the rollers so that it is possible to replace the latter by others of greater or smaller diameter to compensate for possible differences due to working and/or to wear, without having to touch the cam **2**. This has the advantage of not changing the phasing of the cam **2** during maintenance operations on the pump when it is assembled on an engine.

With reference now to FIG. **5** it will be possible to appreciate the importance of the technical problem resolved by the structure of the rocker arm described above. The base diameter of the cam **2**, is indicated *Db* and its outer diameter is indicated *De*. The lift of the cam, that is the stroke which the pumping element **1** has to make is indicated *h*. The greater the number of cylinders of the engine to be supplied, the greater the number of lobes on the cam **2**, each of which controls the pumping element **1**. Since, as has already been stated, the pressure exerted by the roller **7** upon the profile of the cam is very great, the diameter of the roller **7** and hence also of the roller **8** must be above a certain minimum value and, since it is necessary to have two rollers (as has been shown above) with a cam with six lobes, it is impossible to assemble the rocker arm **11** on a pivot pin such as **24***b*, which is sufficiently strong if this has to pass into the zone between the rollers **7** and **8**, as in known control devices of this type.

As is seen in FIG. **5** in fact, in the case of a six lobe cam, the angle α, subtended at the center of the cam by each lobe, is only 60°. Since, when a roller is in a position corresponding to the bottom dead center of the cam profile, the other roller will have to be in a position corresponding to the upper dead center, the angle β, subtended at the center A of the cam between the axes D and E of the roller will be equal to $\alpha/2$. The distance ED between the centers of the two rollers will be given by:

$$ED = 2\ AD. \sin \beta/2$$

and since the diameters of the two rollers **7** and **8** must be substantially equal, the distance *ED* can only be slightly greater than the diameter of the roller **7** or **8**.

Therefore, as has been said already, there would be so little space free between the two rollers **7** and **8** of the rocker arm **11** as to make it impossible to arrange for a pivot pin such as 24*b* to pass through the rocker **11** in the known way and still have sufficient strength in its center portion to withstand the stresses caused by the heavy loads discussed above.

It works out moreover, still with reference to FIG. **5**, that with a six lobe cam, since the stroke h of the pumping element is equal to $(D_e - D_b)$, even if relatively large base diameters *Db* as much as **14**.h are used and hence external diameters $De = 15.h$, the distance *ED* between the centers of the two rollers remains so small that it prevents the use of a pivot pin passing through the rocker arm **11**.

In the case of a rocker arm for a **5** lobe cam where the angle $\beta'/2$ would be greater by 3° than $\beta/2$ used in the calculation of *ED*, (see FIG. **5**), especially if rollers of relatively large diameter in relation to the diameter of the cam are used, the same problem of the impossibility of fitting in a strong pivot pin is found. The problem remains, notwithstanding the apparent greater separation of the axes of the rollers (from *E* to *E'* and from *D* to *D'*) with only five lobes, because of the larger angular sector available for the development of each lobe (72° instead of 60°) because it is preferable, if possible, to reduce the base diameter *Db* of the cam and consequently the distance *E'D'* remains small.

If now a cam with eight lobes is considered, still with reference to FIG. **5**, the centers of the rollers **7**, **8** will be at *E''* and D'' and the angle $\beta/2$ is reduced to $\beta''/2 = 11°.15'$. The distance E''D'' between the rollers **7** and **8** is also reduced and the problem of the pivot pin **24***b* is even more serious. On the other hand, if rollers **7** and **8** of a greatly reduced diameter were used, then because of the small difference in length between AD and AG, it would be the passage of the peak A of the lobes (of diameter *De*)close to the pin **24***b*, which would impose a restriction on the diameter of the said pin **24***b*.

With the technical solution offered by this invention which allows elimination of the pivot pin in the center portion of the rocker arm, as described above, the use of two rollers of large diameter is possible.

In particular, with the technical solution offered by this invention, the outer diameter of the roller **7** which controls the working stroke of the pumping element **1**, being independent of the pivot pin **24***b* of the rocker arm, may be large enough to be able to sustain the heavy loads which are caused by high injection pressures, and to be able to attain a high rotational speed.

With reference now to the FIG. **6**, **7**, **8** and **9**, there will be described the new cam profile which allows effective utilization of the rollers **7** and **8** of the rocker arm with a relatively large diameter in relation to the base diameter *Db* of the cam **2**. As is known, the working stroke of the pumping element **1** (in the direction of the arrow F in FIG. **1**) is effected up to the half stroke position, by the profile of the lobes. Thus starting with the roller **7** at the position corresponding to bottom dead center of the element **1** at the base diameter *Db* of the cam, and moving up to the point *m* (see particularly FIG. **8**) of the flank of the lobe causes the element **1** to effect a half stroke ($h/2$) after a rotation of the cam **2** equal to $360/4n$, where n is the number of lobes of the cam.

It is considered preferable to determine the profile of the lobes by initially defining the first portion of the flank which, together with the roller **7**, generates the first half stroke $h/2$ and subsequently to form the second part of the profile of the lobes up to the top dead center (p.m.s.) by envelopement, that is by defining the second part of the profile to be the appropriate envelope of the successive positions of one roller when the other roller moves over the first part of the profile. Since the profiles **3** and **3***a* of the cam **2** are symmetrical in relation to the radial line AB (see FIG. **1**) the cam having to serve both for right hand and for the left hand rotation, once the first section of profile up to the point m of FIG. **8**, representing half the stroke h of the pumping element, has been defined, the remaining portion of the profile up to the top dead center can be made by envelopement of the successive positions of the roller 7 with respect to the corresponding positions of the roller 6 during the movement of the roller 7 over the first section of the profile.

In this way, at the half stroke $h/2$ of the pumping element 1 the play between the two rollers 7 and 8 on the opposite flanks of the cam lobes works out theoretically at nil. This condition is very important in that it is precisely in this position that accelerations and hence the forces of inertia invert, passing over from positive to negative on each roller.

Moreover in order to obtain a good combustion in engines which require deliveries of fuel at up to 200 mm^3 and more for each cycle and with theoretical gradients of fuel delivery of the pumping element up to 45 mm^3 per degree of rotation of the shaft of the pump, it is necessary to stabilize the maximum stroke h of the pumping element **1**, if possible, to keep it below the ratio $d/c = 2$ where d is the diameter of the pumping element and c is its stroke.

If pumping elements of a relatively large diameter d are avoided, then it will be possible to contain the specific contact pressures between the roller 7 and the flanks of the cam lobes within values comensurate with a relatively long working life. Hence it is important, in determining the profile of the cam lobes, to ensure the desired rate of injection without having to have recourse to the use of pumping elements having an excessively large diameter d.

In order to ensure a good fuel injection rate whatever the velocity gradient transmitted by the cam 2 to the pumping element 1 it is, moreover, important that the maximum speed of the element 1 should occur about the half stroke $h/2$ of the said pumping element, this being the position which also corresponds to half of the angle β subtended by the lobe at the center of the cam, that is the position of the rollers shown in **FIG. 8.**

The profile P_2 of the upper part of the flanks of the lobes, determined by envelopement from the contact of the main roller **7** upon the profile of the lower part P_1, through the angular movement of the rocker arm **11** and of the secondary roller **8**, is thus suitable to sustain the heavy loads due to the pumping element **1.**

The technical solution of the present invention also takes into account the kinematic determining condition according to which it is only after the half stroke $h/2$ of the pumping element 1 has been passed that deceleration and hence a reduction in the linear velocity of the pumping element commences. The dynamic conditions due to the inversion of the direction of acceleration and hence of the forces of inertia, impose the necessity that in proximity to the position m of the rollers on the flanks of the lobes which corresponds to the half stroke position of the pumping element there shall be the least possible play between the rollers **7, 8** and the flanks of the cam lobes over which they move.

In order to take into account all these conditions, on the first part P_1 of the profile of the lobe the characteristics of this part of the profile are arranged, so that the second part P_2 of the profile of the lobe, made by envelopement from the first part P_1 is of convex curvature, of variable radius, and has from the half stroke to about three-fourths of the whole stroke a curve of radii about half of the diameter of the roller 7 and around five-sixths of the stroke, has a curve of radii about one-third of the diameter of the roller. At approximately nine-tenths of the total stroke the curve may be reduced to radii equal to or slightly less than one-fourth of the diameter of the rollers. At the peaks of the lobes the curve does not go down to radii less than one-fifth of the diameter of the rollers.

Thus the stroke h of the pumping element can be used up to 80 – 85 percent without altering the rate of injection and without other disadvantages.

This condition, moreover, together with the high initial velocity given to the pumping element 1 at the beginning of the delivery phase (corresponding to the lower part of each lobe P_1) and together with the high average velocity maintained during the whole injection phase, makes possible the use of pumping element diameters relatively small in relation to the whole delivery of the pump. This is a considerable advantage, particularly with respect to the rate of injection which exists at the outlet of the holes of the injector (not shown).

With particular reference to FIGS. 7 and 9 it will be noted that the initial profile P_1 of the lobes, has a concave curve with a radius Rm between 35 and 45 mm depending on the diameter chosen for the main roller and on the desired initial velocity of the pump element. More particularly the initial profile P_1, from the bottom dead center up to the half stroke position $h/2$, comprises a first substantially rectilinear section r_1 coinciding with the said curve c of radius Rm terminating at the point corresponding to approximately ¼ h, and a third section r_2, also rectilinear and tangential to the intermediate section curve c and extending to the point m corresponding to the half stroke position $h/2$. The remainder of the profile, that is the final profile P_2 of the lobe, from the half stroke position $h/2$, up to the top dead center at the peak of the lobe is formed by envelopement of the initial profile P_1, as has been discussed above.

The characteristics mentioned are illustrated also in FIG. 6 with reference to the rate of motion in the case of cams of five to eight lobes. Upon the abscissa are recorded the degrees of rotation of the cam from the bottom dead center and upon the ordinate the corresponding values of the fraction of a stroke h moved by the pumping element. As can be seen, in the region of the first half stroke the section of rectilinear profile of $r1$ occupies an angle of cam rotation between 2° and 4°, the concave curved section c of radius Rm which extends to the position $h/4$, occupies an angle between 7° and 9°, whilst the third section $r2$, which is also rectilinear and which extends to the position $h/2$, occupies the remaining 3°–4°. The gradient of third section r2, between the positions corresponding to ¼h and ½h, may have a slope of between 0.5 and 0.30 mm per degree of rotation of the cam. Alternatively, it can be defined in such a way that, with the pumping element **1** having its diameter d no greater than twice the stroke h, a theoretical gradient of fuel delivery of the pumping element of between 30 and 45 mm^3 per degree of rotation of the cam is obtained.

From the above data it will be seen that for a **5** lobe cam there could be selected, for example, the angles 5° + 8°.30′ + 4°.30′ = 18° (corresponding to ¼.360/5) whilst for the **8** lobe cam there could be selected respectively the angles 2° + 7°.15′ + 2° = 11°.15′.

In certain cases, according to one variant of the invention, which is especially useful for an eight lobe cam, in order to augment as much as possible the stroke of the pumping element and to make it possible to reach high linear velocities in the first half of the stroke, there can be eliminated the first rectilinear section $r1$ from the bottom dead center, whilst still retaining the said second and third section, c and $r2$ respectively.

What is claimed is:

1. A control device for a fuel injection pump of a multi-cylinder internal combustion engine, said control device having a casing, a cam rotatably mounted in the casing,
   - said cam having a surface including a plurality of lobes, each said lobe being symmetric about a radial line passing through the top dead center of said lobe,
   - cam follower means comprising a rocker arm having at each end thereof a roller, said rollers both being in contact with the surface of said cam,
   - a support member attached to the casing
   - support means on said rocker arm cooperating with said support member to support said rocker arm for pivotal movement about an axis intermediate said rollers, said support means being outside the region between said rollers and
   - a connecting rod having one end operatively connected to one of said rollers and the other end connected to a cursor slidable in said support members, said cam surface being shaped such that the part of the profile of each lobe of said cam between the top dead center position and the half lift position of said cam follower is determined to be the appropriate envelope of the successive positions of said roller when the other said roller moves over the part of the profile of said cam between the bottom dead center position and said half lift position of said cam follower.

2. The device of claim 1, wherein said rocker arm comprises two plate-like elements, each said plate-like element having substantially the form of an isosceles triangle with rounded vertices and with a base longer than the sides, a bridge element rigidly connecting said plate-like elements side by side at the vortex opposite said base, said support means including a stub shaft projecting outwardly from each of said plate-like elements, means forming axially aligned apertures in each pair of the other rounded vertices of the plate-like elements an axle means for said rollers disposed in said apertures.

3. The device of claim 2 wherein said support member for said rocker arm comprises a substantially U-shaped element having in each limb thereof a circular aperture of diameter larger than the diameter of each said stub shaft,
   - a slot in each limb of the U-shaped element, each said aperture communicating laterally with a respective said slot so as to provide a passage of width at least equal to the diameter of each said stub shaft to allow the radial introduction thereof into said aperture,
   - a bush in each said aperture having an external diameter equal to the diameter of the respective aperture, said bush arranged for insertion into said aperture over said stub shaft, said bush serving to lock each said stub shaft in position in said aperture.

4. The device of claim 3 wherein the base of said U-shaped element has an aperture surrounded by a collar through which said cursor passes.

5. The device of claim 2 wherein there is provided a bush between each axle for said rollers and the respective apertures in the ends of said rocker arm.

6. The device of claim 1 wherein said cam has five or more lobes, and wherein a first part of said profile of each lobe, from the bottom dead center position up to the point corresponding with the half lift position of said cam follower means, comprises a concave curve having a radius between 35 and 45 mm, and at least one rectilinear section between said point corresponding to said half lift position of said cam follower and said concave curved section to which it is tangential, said rectilinear section extending to a point which corresponds substantially to a quarter of the lift of said cam follower means.

7. The device of claim 6, wherein said first portion of said profile of each said lobe comprises a first rectilinear section, said concave curved section with radius between 35 and 45 mm, and a second rectilinear section formed by said rectilinear section extending to said half lift position of said cam follower means, both of said rectilinear sections being tangential to said curved section.

8. The device of claim 7 wherein, for cams of five to eight lobes, said first rectilinear section subtends an angle between 2° and 5° at the center of said cam, said curved section subtends an angle of between 7° and 9° at said center of said cam and said second rectilinear section subtends an angle between 2° and 5° at said center of said cam.

9. The device of claim 6 wherein said second rectilinear section of said first part of said profile of said cam lobe which extends at least between said point corresponding substantially to the quarter lift position of said cam follower means and said point corresponding to said half lift position of said cam follower means, has a gradient of between 0.5 and 0.3 mm per degree of rotation of said cam.

10. The device of claim 6 wherein said second rectilinear section of said first part of said cam lobe, extending at least between said point corresponding substantially to the quarter lift position of said cam follower means and said point corresponding to said half lift position of said cam follower means, has a gradient arranged so that when said device is used with a pump having a diameter no greater than twice the stroke, will provide a theoretical gradient of delivery from said pump of between 30 mm^3 and 45 mm^3 per degree of rotation of said cam.

11. The device of claim 1 wherein said two rollers of said cam follower means are a main roller and a subsidiary roller, and a second part of said profile of each said lobe, extending from the top dead center of said half lift position of said cam follower means, has a convex curve in which the section adjacent said second rectilinear section of said first part of said profile, from said half lift position of said cam follower means to about a three-fourths lift position of said cam follower, has a radius not less than half the diameter of said main roller of said cam follower means.

12. The device of claim 1 wherein said two rollers of said cam follower means are a main roller and a subsidiary roller, and a second part of said profile of each said lobe, between top dead center and said half lift position

of said cam follower means has a radius of not less than one-third of said diameter of said main roller of said cam follower means at the section extending about the position corresponding to the five-sixths lift position of said cam follower means.

13. The device of claim **1** wherein said two rollers of said cam follower means are a main roller and a subsidiary roller, and a second section of said profile of each said lobe, between top dead center and said half lift position of said cam follower means has a radius of not less than one-fourth of said diameter of said main roller of said cam follower means at the section extending about the position corresponding to nine-tenths lift position of said cam follower means.

14. The device of claim **1** where said two rollers of said cam follower means are a main roller and a subsidiary roller, and the profile of said peak of each said lobe of said cam, about the top dead center position thereof, has a radius not less than one-fifth of the diameter of said main roller of said cam follower means.

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