

Description

The invention relates to a reciprocating type fluid delivery pump with two pump heads, comprising:
a driving motor;

a pair of plungers for driving said two pump heads, respectively;

a converting means for converting the rotational motion of said driving motor into reciprocating motion of each plunger, wherein said converting means includes a cam means, engageable with said plungers, for reciprocating said plungers, and wherein said cam means has such a configuration that, when the driving motor is rotated at constant velocity, the delivery flow rate during the delivery starting period of each of the plungers is in excess of that during the other periods of the cycle; and

a means for selectively decelerating said driving motor during said excess delivery period when fluid delivery is not carried out under high pressure;

wherein said cam means is designed such that the delivery speed of each plunger has a first rise while the delivery speed of the other plunger is still constant and a second rise while the delivery speed of the other plunger decreases.

Particularly the present invention relates to a so-called small-volume plunger reciprocating type fluid delivery pump which may be used, for example, to deliver a mobile phase in liquid chromatography.

A typical conventional small-volume plunger reciprocating type fluid delivery pump has a driving motor, plungers for driving two pump heads, respectively, and a converting mechanism for converting the rotational motion of the driving motor into a reciprocating motion of each plunger.

Fig. 4 shows the plunger speed characteristics with respect to an angle θ of a conventional small-volume plunger reciprocating type fluid delivery pump having two pump heads.

The curve 1 shows the plunger speed characteristics of the first pump head, while the curve 2 shows the plunger speed characteristics of the second pump head. A cam is used as a converting mechanism for converting the rotational motion of the driving motor into a reciprocating motion of each plunger. The axis of ordinates, i.e., $dr/d\theta$ (r is the distance from the center of rotation of the cam), represents the plunger speed at the time when the driving motor is rotating at a constant velocity. The upper and lower sides of the axis of abscissas show the plunger speeds at the deliver and suction sides, respectively.

The cam is shaped such that $dr/d\theta$ shows a trapezoidal pattern.

In the case of an ideal fluid whose delivery flow rate is proportional to the delivery-side plunger speed, the flow rate of the fluid being delivered, which is the sum of the respective delivery flow rates of the two pump heads, is constant throughout all angles of the cam when the driving motor is rotated at constant velocity, as shown by the reference numeral 3 in Fig. 4.

In practice, however, it is impossible to deliver a fluid at the beginning of the trapezoidal delivery cycle if the pressure is excessively high due to the compressibility of the fluid, the delay in response of a check valve and other factors, and deficiencies 4 and 5 occur in the delivery flow rate and the pressure as shown in Fig. 5. The result is a periodic pulsing flow which leads to a noise in a detection which must be carried out with a high sensitivity.

In order to reduce pulsations in the delivery of fluids, it is conventional practice to control the speed of rotation of the driving motor such that, during the beginning of delivery where the pressure may be insufficient the motor is rotated faster than in the other periods of the cycle.

More specifically, US-A-Re. 31 608, Magnussen, Jr., discloses a fluid pump mechanism for delivering fluid against a back pressure which comprises a piston movable within a chamber for drawing fluid into the chamber during a chamber filling interval, pressurizing the fluid during a pressurizing interval wherein the fluid pressure attains an effective delivery value prior to delivery from the chamber and delivering the pressurized fluid from the chamber during a delivery interval of piston movement, and means for controlling the rate of piston movement such that the piston moves at a predetermined rate during delivery of the pressurized fluid. The controlling means includes a "pump-up" means for establishing a greater rate of piston movement during the pressurizing of the fluid, for signalling completion of fluid pressurization, and for thereupon establishing the predetermined rate of piston movement for effecting delivery of the pressurized fluid from the chamber, thereby increasing the time during which fluid is delivered to a receiving system and decreasing the time of filling or refilling and pump-up prior to such delivery, and thus enabling fluid to be delivered at a given flow rate and with a greatly reduced pulsation.

US-A-4 045 343, Achener et al., discloses a high pressure liquid chromatography system including a reservoir for a liquid mobile phase, an LC column and detector, a high pressure reciprocating pump for enabling flow from the reservoir through the column, and a positively actuated inlet valve for controlling flow from the reservoir to the pump chamber. The pump is driven by motor means, such as a stepping motor, directly coupled thereto; and the inlet valve is actuated by the power train of the motor and pump, e.g., by

an eccentric carried by the pump crank shaft. The pump piston is similarly driven by an eccentric, the pump and inlet valve eccentrics being angularly displaced in their respective positions at the crank shaft, as to delay opening of the inlet valve for a predetermined period following a pump stroke, in order to enable decompression of the liquid in the pump chamber. The average rotational velocity of the stepping motor is controlled throughout each full crank shaft rotation, so as to enable a precisely selected cycle of pump operation. In particular, the speed of the motor is so regulated in conjunction with the mechanical actuation of the pump piston and inlet valve as to provide (at the low flow rates where such behavior is critical) a very short duration fill period--which implies a rapid withdrawal of the piston or plunger from the pump cylinder. Thereafter, the second portion of the pumping cycle, which corresponds to pumping or displacing the liquid from the pump toward the chromatographic column, is effected under crank shaft rotation (as a function of time) such that the axial displacement of the piston is relatively linear as a function of time.

Fig. 6(A) shows compensation for pulsations in the case where a fluid is delivered under high pressure, while Fig. 6(B) shows pulsation compensation in the case where the delivery of a fluid is effected under low pressure. The reference symbol a represents a pulsing flow in the case where the speed of rotation of the cam is kept constant; b represents the speed of rotation of the cam controlled so as to compensate the pulsing flow; and c represents the pulsing flow thus compensated.

The higher the pressure, the greater the deficiency in pressure; therefore, as the pressure is increased, the rotational speed of the driving motor must be increased correspondingly.

Thus, the above prior art methods wherein the rotational speed of the driving motor is varied to reduce pulsations suffers from the problem that, as the pressure is increased, the rotational speed of the driving motor must be increased correspondingly to make compensation, and therefore the load on the motor increases and a driving motor having a relatively high output is needed.

Furthermore, US-A-4 359 312 discloses a reciprocating type fluid delivery pump of the type mentioned at the beginning, however with two piston pumps instead of plunger pumps. In this pump, the cam means is designed such that

(a) the first rise of the delivery speed of each piston, which occurs while the delivery speed of the other piston is still constant, is an abrupt rise, i.e. the piston speed is increased abruptly; and

(b) between the first rise and the second rise the delivery speed of each piston decreases down to zero delivery speed while the delivery speed of the other piston is yet constant.

Although in the pump according to US-A-4 359 312 it is possible to compensate for pulsations occurring due to the compressibility of a fluid and other factors during the delivery starting period by driving the motor with constant speed or with reduced speed during the excess delivery period, in dependence of the compressibility of the fluid, the general shape of the piston speed curve is unfavorable for the driving motor and the controlling thereof in so far as the first rise of the delivery speed is an abrupt rise and such abrupt rise is followed by a decrease of the delivery speed down to zero.

In view of the above-described circumstances, it is a primary object of the present invention to provide a reciprocating type fluid delivery pump which is so designed that it is possible to more favorably compensate for pulsations occurring due to the compressibility of a fluid and other factors during the delivery starting period in a state wherein no load is applied to the driving motor.

To this end, the present invention provides a reciprocating type fluid delivery pump of the type mentioned at the beginning, which is characterized in that

(a) the first rise is a relatively steep rise; and

(b) the delivery speed of each plunger has a short plateau while the delivery speed of the other plunger is yet constant, said plateau connects said first rise with said second rise.

By virtue of the above-described arrangement, in such an ideal state that the flow rate of a fluid being delivered is proportional to the delivery-side speed of a plunger, if the driving motor is rotated at constant velocity, the delivery flow rate during the delivery starting period of the plunger is in excess of that during the other periods of the cycle. In the case where an ordinary fluid is delivered under high pressure, a deficiency in the delivery flow rate occurs during the delivery starting period due to the compressibility of the fluid and other factors. Therefore, the deficiency is cancelled by the excess that is provided on the basis of the configuration of the cam.

In the case where a fluid is delivered under low pressure, if the driving motor is rotated at a constant velocity, the excess in the delivery flow rate that is provided by virtue of the cam configuration remains, so that the fluid is delivered to an excess. Therefore, the rotational speed of the cam is reduced during the excess delivery period so that the delivery flow rate is maintained at a constant level.

The above and other objects, features and advantages of the present invention will become clear from the following description of the preferred embodiment thereof, taken in conjunction with the accompanying drawings, in which:

Fig. 1 is a chart showing the relationship between the plunger speed characteristics and the delivery flow rate at a time when the driving motor is rotated at constant velocity in accordance with one embodiment of the present invention;

Fig. 2 shows the arrangement of one embodiment of the present invention;

Fig. 3(A), 3(B) and 3(C) show the operations of one embodiment of the present invention under different pressures;

Fig. 4 is a chart showing the relationship between the plunger speed characteristics and the delivery flow rate at the time when the driving motor is rotated at a constant velocity in a conventional fluid delivery pump;

Fig. 5 is a chart showing the relationship between the delivery flow rate and pressure in the conventional fluid delivery pump; and

Fig. 6(A) and 6(B) show the operations of the conventional fluid delivery pump under different pressures.

DESCRIPTION OF THE PREFERRED EMBODIMENT

The present invention will be described hereinafter in detail with reference to the accompanying drawings.

Fig. 2 shows a small-volume plunger reciprocating type fluid delivery pump to which the present invention is applied.

The reference numeral 10 denotes a driving stepping motor, and 11 a cam shaft. The rotation of the stepping motor 10 is transmitted to the cam shaft 11 through a small pulley 12, a belt 13 and a large pulley 14 which is secured to the cam shaft 11.

The cam shaft 11 has two cams 15 and 16 secured thereto for driving two pump heads, respectively. The cam shaft 11 further has a disk 17 secured thereto. The disk 17 is provided with a bore 18, so that the home position relative to the rotational angle of the cams 15 and 16 is detected by detecting the bore 18 by means of a photocoupler 19.

The proximal end of a crosshead 20 is in contact with the cam 15 so that the crosshead 20 performs a reciprocating motion. A plunger 21 of a first pump head 24 is secured to the other end of the crosshead 20. Similarly, the proximal end of a crosshead 22 is in contact with the other cam 16 so that the crosshead 22 performs a reciprocating motion, and a plunger 23 of a second pump head 25 is secured to the crosshead 22.

Cams 15 and 16, which will be described later in detail, are designed to have configurations that provide angle-plunger speed characteristics as shown in Fig. 1.

At the first pump head 24, the reciprocating motion of the plunger 21 causes a fluid 26 to be delivered to a flow path leading to a pressure sensor 28 from a suction filter 29 and an inlet block 27. At the second pump head 25 also, the reciprocating motion of the plunger 23 causes the fluid 26 to be delivered to a flow path leading to the pressure sensor 28 from the suction filter 29 and the inlet block 27. At each of the pump heads 24 and 25, the reference symbol a denotes an inlet check valve, b an outlet check valve, and c seal member.

In the pressure sensor 28, the fluids respectively delivered from the first and second pump heads 24 and 25 are joined together and are then delivered from a pump outlet 30 to a flow path which leads to a column. Reference numeral 31 denotes a drain flow path, and 32 a drain bin.

The reference numeral 33 denotes a CPU. A pressure signal from the pressure sensor 28 and a home position detecting signal from the photocoupler 19 are sent to CPU 33. CPU 33 controls the speed of rotation of the stepping motor 10.

The configurations of the cams 15 and 16 are determined so that the $dr/d\theta$ (the plunger speed at the time of uniform rotation) characteristics with respect to the cam rotational angle θ are set such as those shown in Fig. 1. In Fig. 1, the curves 1a and 2a represent the $dr/d\theta$ characteristics of the first and second pump heads, respectively.

The following is the cam configuration that provides the $dr/d\theta$ characteristics shown in Fig. 1.

$0 < \theta \leq 10$	$r = 15.5 + 0.075\theta^2/294.14062$
$10 < \theta \leq 25$	$r = 15.5 + (1.5\theta - 7.5)/294.14062$
$25 < \theta \leq 70$	$r = 15.5 + (7\theta^2/180 - 4\theta/9 + 605/36)/294.14062$
$70 < \theta \leq 205$	$r = 15.5 + (5\theta - 173.75)/294.14062$
$205 < \theta \leq 220$	$r = 15.5 + (-8\theta^2/90 + 373\theta/9 - 140735/36)/294.14062$
$220 < \theta \leq 250$	$r = 15.5 + (-7\theta^2/180 + 175\theta/9 - 53615/36)/294.14062$
$250 < \theta \leq 300$	$r = 15.5 + (-0.1255\theta^2 + 62.75\theta - 6902.5)/294.14062$
$300 < \theta \leq 340$	$r = 15.5 + (-12.55\theta + 4392.5)/294.14062$
$340 < \theta \leq 360$	$r = 15.5 + (0.31375\theta^2 - 225.9\theta + 4066.2)/294.14062$

As shown in Fig. 1, if the driving motor is rotated at constant velocity, the sum of the respective speeds of the two plungers is larger during the delivery starting period of each plunger than during the other periods of the cycle. Therefore, assuming that the flow of the fluid accurately follows the delivery-side plunger speed, the delivery flow rate 3a goes to excess during the delivery starting period in each cycle, as shown by the reference symbol A.

Figs. 3(A)-3(C) show the relationship between the pressure and the rotational speed of the driving motor in this embodiment.

Fig. 3(A) shows the pulsation compensation that is made in the case where fluid delivery is carried out under substantially no pressure. If the driving motor is rotated at constant velocity, excesses A occur in the delivery flow rate during the delivery starting period in each cycle, as shown by graph a. Therefore, the driving motor is decelerated such that the rotational speed of each cam is reduced during each delivery starting period, as shown by graph b. As a result, the delivery flow rate becomes constant as shown by graph c.

Fig. 3(B) shows the pulsation compensation that is made in the case where the fluid delivery is carried out under low pressure. If the driving motor is rotated at constant velocity, an excess A' in the delivery flow rate appears in the temporally latter portion of each excess delivery period, as shown by graph a. Therefore, the rotational speed of the motor is reduced as shown by graph b. As a result, the delivery flow rate becomes constant as shown by graph c.

Fig. 3(C) shows the pulsation compensation made in the case where the fluid delivery is carried out under high pressure. In this case, deficiencies in the delivery flow rate due to high pressure and excesses in the delivery flow rate due to high pressure and excesses in the delivery flow rate due to the modification of the cams cancel each other, so that the delivery flow rate in graph a at the time when the driving motor is rotated at constant velocity is constant. Accordingly, there is no need to compensate the rotational speed in graph b of the driving motor.

Since the relationship between the pressure and the optimal amount of compensation (i.e., the driving motor deceleration period) depends on the kind of fluid delivered, it may be possible to set the relationship in advance or obtain it automatically while monitoring the level of the pressure.

The compensation that is made by decelerating the driving motor is conducted in a relatively low flow rate region where it is highly essential to reduce pulsations. In flow rate regions where the flow rate is high and fluid delivery is effected at high frequency, pulsations are small, so that it is unnecessary to reduce the rotational speed of the driving motor to compensate. In such a case, the cams may be rotated at constant velocity.

In the present invention, the configurations of cams for reciprocating respective plungers are improved so that, when the driving motor is rotated at constant velocity, the delivery flow rate during the delivery starting period of each of the plungers is in excess of that during the other periods of the cycle. Therefore, the speed of rotation of the driving motor is controlled in such a manner that, when the fluid is delivered under low pressure, the rotational speed is reduced, whereas, when the fluid delivery is carried out under high pressure, the motor is rotated at constant velocity. Accordingly, the load on the driving motor is reduced and it is possible to employ a driving motor having a relatively low output in comparison with the conventional compensation wherein the rotational speed of the driving motor is increased under high pressure.

Although the present invention has been described above through specific terms, it should be noted here that the described embodiment is not necessarily exclusive and various changes and modifications may, of course, be imparted thereto without departing from the scope of the present invention which is limited solely by the appended claim.

Claims

0<θ≤10	r = 15.5 + 0.075θ ² /294.14062
10<θ≤25	r = 15.5 + (1.5θ - 7.5)/294.14062
25<θ≤70	r = 15.5 + (7θ ² /180 - 4θ/9 + 605/36)/294.14062
70<θ≤205	r = 15.5 + (5θ - 173.75)/294.14062
205<θ≤220	r = 15.5 + (-8θ ² /90 + 373θ/9 - 140735/36)/294.14062
220<θ≤250	r = 15.5 + (-7θ ² /180 + 175θ/9 - 53615/36)/294.14062
250<θ≤300	r = 15.5 + (-0.1255θ ² + 62.75θ - 6902.5)/294.14062
300<θ≤340	r = 15.5 + (-12.55θ + 4392.5)/294.14062
340<θ≤360	r = 15.5 + (0.31375θ ² - 225.9θ + 4066.2)/294.14062

As shown in Fig. 1, if the driving motor is rotated at constant velocity, the sum of the respective speeds of the two plungers is larger during the delivery starting period of each plunger than during the other periods of the cycle. Therefore, assuming that the flow of the fluid accurately follows the delivery-side plunger speed, the delivery flow rate 3a goes to excess during the delivery starting period in each cycle, as shown by the reference symbol A.

Figs. 3(A)-3(C) show the relationship between the pressure and the rotational speed of the driving motor in this embodiment.

Fig. 3(A) shows the pulsation compensation that is made in the case where fluid delivery is carried out under substantially no pressure. If the driving motor is rotated at constant velocity, excesses A occur in the delivery flow rate during the delivery starting period in each cycle, as shown by graph a. Therefore, the driving motor is decelerated such that the rotational speed of each cam is reduced during each delivery starting period, as shown by graph b. As a result, the delivery flow rate becomes constant as shown by graph c.

Fig. 3(B) shows the pulsation compensation that is made in the case where the fluid delivery is carried out under low pressure. If the driving motor is rotated at constant velocity, an excess A' in the delivery flow rate appears in the temporally latter portion of each excess delivery period, as shown by graph a. Therefore, the rotational speed of the motor is reduced as shown by graph b. As a result, the delivery flow rate becomes constant as shown by graph c.

Fig. 3(C) shows the pulsation compensation made in the case where the fluid delivery is carried out under high pressure. In this case, deficiencies in the delivery flow rate due to high pressure and excesses in the delivery flow rate due to the modification of the cams cancel each other, so that the delivery flow rate in graph a at the time when the driving motor is rotated at constant velocity is constant. Accordingly, there is no need to compensate the rotational speed in graph b of the driving motor.

Since the relationship between the pressure and the optimal amount of compensation (i.e., the driving motor deceleration period) depends on the kind of fluid delivered, it may be possible to set the relationship in advance or obtain it automatically while monitoring the level of the pressure.

The compensation that is made by decelerating the driving motor is conducted in a relatively low flow rate region where it is highly essential to reduce pulsations. In flow rate regions where the flow rate is high and fluid delivery is effected at high frequency, pulsations are small, so that it is unnecessary to reduce the rotational speed of the driving motor to compensate. In such a case, the cams may be rotated at constant velocity.

In the present invention, the configurations of cams for reciprocating respective plungers are improved so that, when the driving motor is rotated at constant velocity, the delivery flow rate during the delivery starting period of each of the plungers is in excess of that during the other periods of the cycle. Therefore, the speed of rotation of the driving motor is controlled in such a manner that, when the fluid is delivered under low pressure, the rotational speed is reduced, whereas, when the fluid delivery is carried out under high pressure, the motor is rotated at constant velocity. Accordingly, the load on the driving motor is reduced and it is possible to employ a driving motor having a relatively low output in comparison with the conventional compensation wherein the rotational speed of the driving motor is increased under high pressure.

Although the present invention has been described above through specific terms, it should be noted here that the described embodiment is not necessarily exclusive and various changes and modifications may, of course, be imparted thereto without departing from the scope of the present invention which is limited solely by the appended claim.

Claims



1. A reciprocating type fluid delivery pump with two pump heads (24, 25), comprising:
a driving motor (10);
a pair of plungers (21, 23) for driving said two pump heads (24, 25), respectively;
a converting means (11, 13 - 16, 20, 22) for converting the rotational motion of said driving motor (10) into reciprocating motion of each plunger (21, 23), wherein said converting means (11, 13 - 16, 20, 22) includes a cam means (15, 16), engageable with said plungers (21, 23), for reciprocating said plungers (21, 23), and wherein said cam means (15, 16) has such a configuration that, when the driving motor (10) is rotated at constant velocity, the delivery flow rate during the delivery starting period of each of the plungers (21, 23) is in excess of that during the other periods of the cycle; and
a means (33) for selectively decelerating said driving motor (10) during said excess delivery period when fluid delivery is not carried out under high pressure;
wherein said cam means (15, 16) is designed such that the delivery speed of each plunger (21, 23) has a first rise while the delivery speed of the other plunger (23, 21) is still constant, and a second rise while the delivery speed of the other plunger (21, 23) decreases;
characterized in that
(a) the first rise is a relatively steep rise; and
(b) the delivery speed of each plunger (21, 23) has a short plateau while the delivery speed of the other plunger (23, 21) is yet constant, said plateau connects said first rise with said second rise.
2. The pump of claim 1, characterized in that said cam means (15, 16) includes a pair of cams, each cam engaging a corresponding one of said plungers (21, 23).
3. The pump of claim 1, characterized in that said decelerating means (33) decelerates said driving motor (10) during said excess delivery period when said fluid delivery is carried out under low pressure.

Patentansprüche

1. Fluidabgabepumpe vom hin- und hergehenden Typ mit zwei Pumpenköpfen (24, 25), umfassend:
einen Antriebsmotor (10);
ein Paar Plunger (21, 23) zum Antreiben der beiden Pumpenköpfe (24, 25), jeweils;
eine Umwandlungseinrichtung (11, 13 - 16, 20, 22) zum Umwandeln der Drehbewegung des Antriebsmotors (10) in hin- und hergehende Bewegung von jedem Plunger (21, 23), wobei die Umwandlungseinrichtung (11, 13 - 16, 20, 22) eine mit den Plungern (21, 23) in Eingriff bringbare Nocken- bzw. Nockenscheibeneinrichtung (15, 16) zum Hin- und Herbewegen der Plunger (21, 23) aufweist, und worin die Nocken- bzw. Nockenscheibeneinrichtung (15, 16) eine derartige Konfiguration hat, daß dann, wenn der Antriebsmotor (10) mit konstanter Geschwindigkeit gedreht wird, die Abgabeströmungsrate während der Abgabebeginnperiode von jedem der Plunger (21, 23) im Übermaß gegenüber derjenigen während der anderen Perioden des Zyklus ist; und
eine Einrichtung (33) zum wahlweisen Verzögern des Antriebsmotors (10) während der Übermaßabgabeperiode, wenn die Fluidabgabe nicht unter hohem Druck ausgeführt wird;
worin die Nocken- bzw. Nockenscheibeneinrichtung (15, 16) derart ausgebildet ist, daß die Abgabegeschwindigkeit von jedem Plunger (21, 23) einen ersten Anstieg hat, während die Abgabegeschwindigkeit des anderen Plungers (23, 21) noch konstant ist, und einen zweiten Anstieg, während die Abgabegeschwindigkeit des anderen Plungers (21, 23) abnimmt;
dadurch **gekennzeichnet**, daß
(a) der erste Anstieg ein relativ steiler Anstieg ist; und
(b) die Abgabegeschwindigkeit von jedem Plunger (21, 23) ein kurzes Plateau hat, während die Abgabegeschwindigkeit des anderen Plungers (23, 21) noch konstant ist, wobei das Plateau den ersten Anstieg mit dem zweiten Anstieg verbindet.
2. Pumpe nach Anspruch 1, dadurch **gekennzeichnet**, daß die Nocken- bzw. Nockenscheibeneinrichtung (15, 16) ein Paar Nocken bzw. Nockenscheiben aufweist, wobei jede Nocke bzw. Nockenscheibe mit einem entsprechenden einen der Plunger (21, 23) in Eingriff tritt.
3. Pumpe nach Anspruch 1, dadurch **gekennzeichnet**, daß die Verzögerungseinrichtung (33) den Antriebsmotor (10) während der Übermaßabgabeperiode, wenn die Fluidabgabe unter niedrigem Druck ausgeführt wird, verzögert.



Revendications

1. Pompe alternative de délivrance d'un fluide équipée de deux têtes de pompe (24,25), comprenant:
 - un moteur d'entraînement (10);
 - un couple de pistons (21,23) servant à entraîner respectivement lesdites deux têtes de pompe (24,25);
 - des moyens de conversion (11,13-16,20,22) pour convertir le mouvement de rotation dudit moteur d'entraînement (10) en un déplacement alternatif de chaque piston (21,23), lesdits moyens de conversion (11,13,16,20,22) comprenant des moyens formant cames (15,16) possédant une configuration telle que, lorsque le moteur d'entraînement (10) tourne à une vitesse constante, le débit de refoulement pendant la période de démarrage du refoulement de chacun des pistons (21,23) dépasse le débit présent pendant les autres périodes du cycle; et
 - des moyens (33) pour décélérer de façon sélective ledit moteur d'entraînement (10) pendant ladite période de refoulement à débit excessif lorsque le refoulement du fluide n'est pas effectué sous haute pression;
 - dans laquelle lesdits moyens formant cames (15,16) sont agencés de telle sorte que la vitesse de refoulement de chaque piston (21,23) présente une première montée, alors que la vitesse de refoulement de l'autre piston (23,21) est encore constante, et une seconde montée, alors que la vitesse de refoulement de l'autre piston (21,23) diminue;
 - caractérisée en ce que
 - (a) la première montée est une montée relativement pentue; et
 - (b) la vitesse de refoulement de chaque piston (21,23) présente un palier court, alors que la vitesse de refoulement de l'autre piston (23,21) est encore constante, ledit palier raccordant ladite première montée à ladite seconde montée.
2. Pompe selon la revendication 1, caractérisée en ce que lesdits moyens formant cames (15,16) comprennent un couple de cames dont chacune engrène avec l'un correspondant desdits pistons (21,23).
3. Pompe selon la revendication 1, caractérisée en ce que lesdits moyens de décélération (33) décèlent ledit moteur d'entraînement (10) pendant ladite période de refoulement avec un débit excessif, lorsque ladite délivrance du fluide est exécutée à basse pression.

