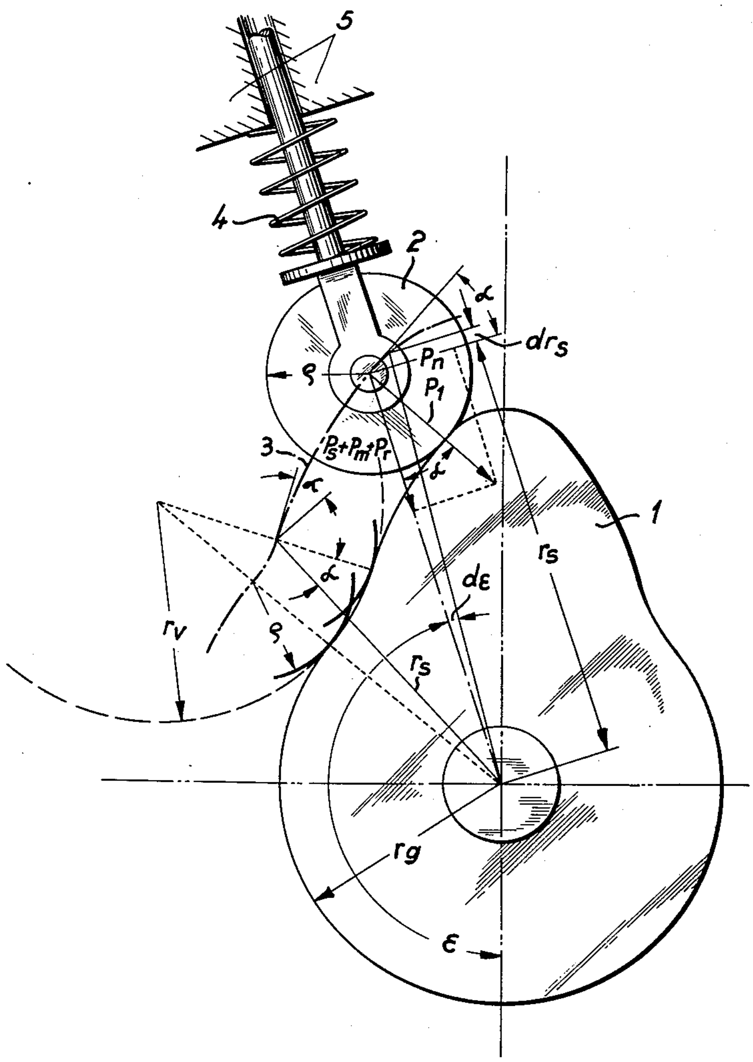
May 15, 1962 W. VOGEL 3,034,363
CAM DRIVE
Filed Dec. 9, 1959



INVENTOR:
WILHELM VOGEL
BY
Michael S. Striker
Attorney
HIS ATTORNEY

United States Patent Office

3,034,363
Patented May 15, 1962

3,034,363
CAM DRIVE

Wilhelm Vogel, Stuttgart-Bad Cannstatt, Germany, assignor to Robert Bosch G.m.b.H., Stuttgart, Germany

Filed Dec. 9, 1959, Ser. No. 858,477
Claims priority, application Germany Dec. 13, 1958
10 Claims. (Cl. 74—55)

The present invention relates to a cam drive consisting of a rotary plate cam whose working outline is constantly tracked by the periphery of a positively guided follower. The cam drive of my invention is particularly useful as a means for reciprocating the piston of a fuel injection pump in internal combustion engines.

Depending upon its intended use, the geometric shape of a plate cam for such cam drives, e.g. a radial cam cooperating with a spring-biased radially reciprocable roller follower, is presently designed with the help of various geometric lines.

(1) If the requirements regarding the load upon the follower and the maximum permissible rotational speed of the plate cam are comparatively low, it is satisfactory to utilize cams with working outlines or cam surfaces consisting of circular arcs and straight zones (tangents) in contact with the arcuate zones. The term "maximum permissible rotational speed" as utilized hereinabove is intended to denote the rotational speed of a plate cam just below that speed which will cause movement of the follower away from contact with the cam surface, i.e. at which the action of a biasing spring is insufficient to maintain the follower in uninterrupted contact with the working outline of a plate cam.

(2) In more rapidly rotating cam drives, the path of the follower at the high point of a plate cam depends upon the angle of rotation of the cam shaft in accordance with a sine law. In such manner, it is possible to attain a most precise, uniform retardation at this critical point of the working outline of a plate cam and, moreover, by using a follower-biasing spring of predetermined strength, a very high maximum permissible rotational speed of the plate cam may be achieved.

(3) A further increase in the maximum permissible rotational speed of a plate cam (i.e. that rotary speed at which the follower still remains in continuous contact with the working outline of the cam) may be brought about by using plate cams with cam surfaces of such configuration as to prevent a vibration of the follower-biasing spring at the latter's own characteristic frequency, because such vibrations invariably affect the flux of forces between the plate cam and its follower. By preventing vibrations of the spring, the maximum permissible rotational speed of the plate cam may be increased by up to 10 percent.

It is known that, at a sufficiently high rotational speed of the plate cam, the helices of a coil spring (which is utilized for biasing the follower into continuous contact with the working outline of a plate cam) will be caused to vibrate (for example, owing to higher harmonic frequencies) whereby, when the momentary vibratory movement occurs in a direction away from the plate cam, the pressure at which the spring biases the follower is reduced considerably. The cams in which such weakening of the spring tension can be prevented up to a given comparatively high maximum value of rotational speed are known

as "non-jerking" cams. The calculation or design of such "non-jerking" plate cams involves two basic steps. In the first step, a harmonic analysis is made of the so-called pattern of movement (i.e. the relation of the follower's displacement to the angle of rotation of the plate cam). Such analysis renders it possible to determine the amplitude of all sine functions of which the follower's pattern of movement consists. By dividing the characteristic frequency of the follower-biasing spring with the required maximum rotational speed of the cam shaft, one obtains that lowermost "upper wave" of the follower's pattern of movement which must be avoided in said pattern. In the second step, the desired "non-jerking" outline of the plate cam is determined by a harmonic analysis of the "lower wave" and of such "upper waves" which are below the calculated upper limit.

When the just described method was applied in actual manufacture of plate cams, it was found that the range of permissible loads upon the follower is lower in the "non-jerking" cam drives. In analogous cam drives of known design, and assuming that the load upon the follower remains the same, the maximum permissible rotational speed of the plate cam is increased but the maximum permissible displacement of a follower is reduced.

(4) It is also possible to design a "non-jerking" cam drive in a fully empirical way by constructing the pattern of follower's movement (i.e. the displacement of the follower in dependency on the angle of rotation of the plate cam) of straight lines as well as of sinusoidal and parabolic curves. However, such construction does not take into consideration the permissible load, particularly the maximum permissible Hertzian compression to which the components of the cam drive may be subjected. The term "maximum permissible Hertzian compression" is intended to denote that pressure between the plate cam and its follower which the material of these parts is capable of resisting without destruction.

Heretofore, it was not possible to design or calculate a plate cam for the pistons of fuel injection pumps whose working outline would be of such configuration as to insure larger useful displacement or stroke of the follower and hence larger quantities of fuel delivered per each stroke of the piston without exceeding the maximum permissible Hertzian compression between the follower and the plate cam if the maximum stroke, the material, the diameter of the follower roller, the maximum permissible rotational speed of the plate cam, and the load upon the injection piston must remain unchanged.

An object of the present invention is to provide a plate cam with a working outline or periphery whose geometric form is such as to insure not only an increase in the useful travel of the follower but also an increased rise or lift of the plate cam.

Another object of the invention is to provide a cam of the just outlined characteristics which, when used to operate the injection piston in the fuel injection pump of an internal combustion engine, is capable of increasing the capacity of the pump by up to 100 percent (and on the average by at least 30 to 40 percent) at unchanged loads upon the follower and at unchanged maximum permissible rotational speed of the plate cam.

With the above objects in view, the invention resides in the provision of a plate cam whose working outline if of such curvature—at least in that zone which brings about a cam action upon the follower—that, at a given relation of the load upon the follower and of the angle of cam's rotation to time, a maximum permissible Hertzian compression may be attained at all points of the working outline of the plate cam.

More particularly, and in a more specific cam drive which utilizes a radial cam and a radially reciprocable follower roller which latter travels along the working outline of the cam, the above and certain other objects of my invention may be attained by designing the track of the roller axis, i.e. the pitch curve of the roller follower's center—at least about that zone of the cam surface which produces cam action upon the follower—in such a way as to at least approximately satisfy the following differential equation:

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_1 \cdot \frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2} = \frac{1}{2\pi(1-\nu^2)} \cdot \frac{E}{P^2 l\rho} P_s + \frac{\rho}{r_s} \cdot \frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2} - \frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}} \tag{1}$$

In the above equation,

P_s is the load in kg. in the direction in which the follower roller reciprocates;

p is the maximum permissible Hertzian compression between the roller follower and the surface of the radial cam in kg./mm.^{2};

r_s is the distance between the axis of the radial cam and the pitch curve of the center of the roller follower in mm.;

ϵ is the angle of rotation of the radial cam, in degrees, from the beginning of cam action to the point of computation;

ρ is the radius of the roller follower in mm.;

E is the modulus of elasticity in kg./mm.^{2} (if the radial cam and the follower are made of different materials, the following equation applies: $1/E = \frac{1}{2}(1/E_1 + 1/E_2)$ wherein E_1 is the elasticity modulus of the roller and E_2 is the modulus of the radial cam, or vice versa);

ν is the Poisson ratio (for metals $\nu=0.3$); and

l is the stressed or active width of the roller follower and of the radial cam in mm. (if the width of the cam exceeds the width of the follower, l indicates the width of the follower).

The novel features which are considered as characteristic of the invention are set forth in particular in the appended claims. The invention itself, however, both as to its construction and its method of operation, together with additional objects and advantages thereof, will be best understood from the following detailed description of a specific embodiment when read in connection with the accompanying drawing the single illustration of which is a schematic representation of a radial cam and of a radially reciprocable follower roller therefor.

Referring now in greater detail to the drawing, there is shown a radial cam **1** whose peripheral cam surface is in permanent engagement with a roller follower **2**. The latter's axis of rotation or center travels in a phantom-line arcuate path **3**, hereinafter called pitch curve, and the follower is constantly biased in radial direction of the cam **1** by a resilient element here shown as a helical spring **4**. The bearing means **5** reciprocably guides the follower roller **2** radially of the plate cam **1**.

It is assumed in the following description that the radial cam **1** is utilized for reciprocating the non-represented piston of a fuel injection pump for internal combustion engines, and that the piston is connected to, and hence reciprocates with, the follower roller **2**. In a fuel injection pump, the useful or pay load P_s may be calculated—at a given diameter d_s of the piston and at a given pressure p_P in the working chamber of the injection pump—with the help of the following formula:

$$P_s = \frac{\pi}{4} d_s^2 p_P$$

To arrive at the aforementioned differential Equation 1, the following starting equations must be resorted to:

As can be observed in the drawing, the angle which indicates the inclination of the pitch curve **3** may be calculated as follows:

$$\alpha = \arc \tan \frac{dr_s}{r_s d\epsilon}$$

and $P_n = P_s \tan \alpha$, if the inertia forces, the force of spring **4**, and the frictional forces are neglected.

As regards the load P_1 upon the roller follower **2**, the following applies:

$$P_1 = \frac{P_s}{\cos \alpha} = P_s \sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}$$

The Hertzian compression p at the point of contact between the periphery of the roller follower **2** and the peripheral cam surface of the radial cam **1** can be calculated as follows:

$$p^2 = \frac{P_1 E}{2\pi(1-\nu^2)l}\left(\frac{1}{\rho} + \frac{1}{r_{b-\rho}}\right)$$

wherein r_b is the radius of curvature of the pitch curve **3** at the point of contact between the members **1** and **2**. The value of radius r_b may be calculated as follows:

$$r_b = \frac{\left[r_s^2 + \left(\frac{dr_s}{d\epsilon}\right)^2\right]^{3/2}}{r_s^2 + 2\left(\frac{dr_s}{d\epsilon}\right)^2 - r_s \frac{d^2r_s}{d\epsilon^2}}$$

In addition to the useful load P_s, the force of the resilient element **4** must be considered in most instances. When the calculation indicates positive acceleration forces (e.g. when the cam surface is of concave configuration), the load increase upon the roller follower **2** owing to the inertia forces must also be taken into consideration. The aforementioned differential Equation 1 then reads as follows:

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2 \left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2} - \frac{Em\omega^2}{2\pi(1-\nu^2)p^2 l\rho}\right\} = \frac{E}{2\pi(1-\nu^2)p^2 l\rho} \cdot [P_s + P_{fm} - c(r_g + \rho + hm - r_s)] + \frac{\rho}{r_s} \cdot \frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2} - \frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}} \tag{2}$$

The symbols first utilized in the above Equation 2 indicate the following:

P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement h_m;

c is the elasticity constant of the spring **4** in kg./mm.;

m is the mass of moving parts in kg. sec.^{2}/mm.;

r_g is the radius of the base circle of the cam at zero travel in mm.;

h_m is the maximum follower displacement in mm.; and

ω is the velocity of the radial cam **1** in 1/sec.

In the differential Equation 2, it is assumed that

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2 > \left.\frac{d^2r_s}{d\epsilon^2}\right|_1$$

i.e. that

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2$$

is greater than

$$\frac{\frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2l\rho}P_s+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}}{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}}$$

The above arrived at value of

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_1$$

is obtained by a simple rearrangement of Equation 1.

The additional starting equations for the differential Equation 2 are as follows:

The force P_f of spring **4** equals:

$$P_f=P_{fm}-c(r_g+\rho+h_m-r_s)$$

The inertia force P_m at an at least approximately uniform rotation of the radial cam **1** may be determined as follows:

$$P_m=m\omega^2\frac{d^2r_s}{d\epsilon^2}$$

The strong lateral or component forces (side thrust) upon the bearings **5** of the roller follower **2**, which are particularly felt in hollow cams, render it necessary to consider the frictional forces developing between the bearings **5** and the roller follower **2** in the direction in which the follower reciprocates. The frictional force $P_r=\mu P_n$ (wherein μ is the coefficient of friction). The side thrust or later force P_n can be determined by the following equation:

$$P_n=(P_s+P_f+P_m+P_r)\tan\alpha$$

Thus, the total load P_1 upon the roller follower **2** may be calculated as follows:

$$P_1=\frac{P_s+P_f+P_m+P_r}{\cos\alpha}=(P_s+P_f+P_m)\frac{\sqrt{1+\tan^2\alpha}}{1-\mu\tan\alpha}$$

The final differential equation is then as follows:

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_{3,4}\left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-v^2)p^2l\rho\left(1-\mu\frac{dr_s}{r_sd\epsilon}\right)}\right\}$$
$$=\frac{E}{2\pi(1-v^2)p^2l\rho\left(1-\mu\frac{dr_s}{r_sd\epsilon}\right)}[P_s+P_{fm}-c(r_g+\rho+hm-r_s)]$$
$$+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}} \quad (3)$$

Thus, if the frictional force between the follower and its bearings must be taken into consideration, the differential Equations 1 and 2 are modified by replacing the ratio

$$\frac{E}{2\pi(1-\nu^2)l\rho}$$

with the ratio

$$\frac{E}{2\pi(1-v^2)l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}$$

The modified Equation 1 then takes into consideration the useful load P_s and the frictional forces P_r, and the modified Equation 2 considers the useful load P_s, the spring force P_f, the inertia forces P_m and the frictional force P_r.

When the differential Equation 2 or (3) is applied. it must be taken into consideration that, at higher or lower rotational speeds, the ratio

$$\frac{d^2r_s}{d\epsilon^2}$$

might assume different values. If no other conditions must be taken into consideration, the final shape of the working outline on the radial cam **1** (i.e. of that portion of the cam surface which causes a cam action upon the follower) must be based on such solutions of the differential equation which indicate a smaller convex- or a larger concave curvature.

Thus, the following applies:

$$\text{If } \left.\frac{d^2r_s}{d\epsilon^2}\right|_{\omega=0}>\left.\frac{dr_s}{d\epsilon^2}\right|_{\omega=\omega_{max}}, \text{ then } \left.\frac{dr_s}{d\epsilon^2}\right|_{\omega=0}$$

must be used in the differential equation. However, if

$$\left.\frac{dr_s}{d\epsilon^2}\right|_{\omega=\omega_{max}}>\left.\frac{dr_s}{d\epsilon^2}\right|_{\omega=0}, \text{ then } \left.\frac{d^2r_s}{d\epsilon^2}\right|_{\omega=\omega_{max}}$$

must be used in the equation.

In order to take even greater advantage of the cam drive, i.e. to increase the efficiency of the cam drive beyond that when the above differential equation is used only for the calculation of such portions of the cam surface which cause a useful stroke (rise or fall) of the roller follower **2**, the present invention also provides for application of the differential equation for that portion of the cam surface which is located at the high point of the cam disc **1**, i.e. at the point on the limit circle. If the cam drive is used to operate the piston of an injection pump for internal combustion engines, the pressure in the working chamber of the injection pump at the time when the roller follower engages with the high point of the cam surface on the member **1** must be considered zero, i.e. only the tension of the spring, the inertia forces and the frictional forces must be taken into consideration.

It can occur, particularly in immediate proximity of the high point of the cam surface on member **1**, that the calculated curvature is so pronounced as to cause a movement of the follower roller away from the cam disc when the latter reaches a given rotational speed.

For the above reason, it is proposed in accordance with the present invention to limit the application of the differential equation to calculation of such zones or portions of the cam surface in which the spring tension is greater than, or at least equal to, the inertia forces, so that the roller follower always rests on and is thus in permanent contact with the cam surface. For the remaining zone of the cam surface, the following formula will apply:

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_m=-\frac{\frac{\pi}{4}d_s^2P_p+P_{fm}-c(r_g+\rho+h_m-r_s)}{m\omega^2_{max}}$$

Also, in order to consider the Hertzian compression in the differential equation, it is necessary that

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_m>\left.\frac{d^2r_s}{d\epsilon^2}\right|_{\omega=0} \quad (3, 4)$$

With the help of the above explained formulae, it is possible to calculate a cam surface—starting from the high point of the cam—in such a way as to consider the entire zone of useful travel, i.e. that arcuate portion of the cam surface which causes a reciprocatory movement of the follower roller. It is, however, a further condition that the "useful" portion of the cam surface (i.e. that zone which causes a reciprocation of the follower member) should comprise no area where the value of $tg\alpha$ is less than zero. The value of $tg\alpha$ is considered here with reference to the pitch curve **3**. According to the present-day knowledge, such cams would be senseless.

The two arcuate zones of transition from that portion of the cam surface which is calculated with the help of my novel differential equation into the base circle (radius r_g) can be achieved in a number of ways.

If r_v represents the radius of the smallest grinding disc which, for manufacturing reasons, may be utilized in the production of the cam disc 1, and if the radius r_v is known, then the following equation must be satisfied at the zones of transition of the arcuate curve obtained with the differential equation into a circular path:

$$r_s = -(r_v-\rho)\cos\alpha + \sqrt{(r_g+r_v)^2-(r_v-\rho)^2\sin^2\alpha}$$

It may occur that the points of transition fall into the zone of required useful displacement of the follower, i.e. into that arcuate portion of the cam surface which causes a reciprocation of the follower. In such manner, the duration of the stroke of a piston connected with the follower roller 2 (as expressed in degrees of the cam angle) may be increased. It is, therefore, advisable in these instances to reduce the radius to such an extent that the point of transition coincides with the lowermost point of the "useful" displacement of the follower roller.

In such cases, and since the values of r_s and α are considered to be known upon the solution of the differential equation, the following applies:

$$r_{v_{max}} = \frac{r_s^2+\rho^2-2r_s\rho\cos\alpha - r_g^2}{2(r_g+\rho-r_s\cos\alpha)}$$

The following applies for the condition that, by suitable selection of instrumentalities used in the manufacture of the cam disc, r_v equals ρ and the overall displacement of the roller follower must be brought about in shortest time possible. During the transition, the magnitude of acceleration at the lowermost point of the cam's lobe depends solely upon the strength of the antifriction bearings in which the parts are mounted. Such maximum strength P_{1_m} must be determined either by calculation or empirically and, upon determination thereof, the following formula applies:

$$m\omega^2\frac{d^2r_s}{d\epsilon^2}\bigg|_1 = P_{1_m}\frac{1-\mu\dfrac{dr_s}{r_s d\epsilon_1}}{\sqrt{1+\left(\dfrac{dr_s}{r_s d\epsilon_1}\right)^2}} - \frac{\pi}{4}d_s^2 p_P - P_{fm} + c(r_g+\rho+h_m-r_s)$$

The solution of this differential equation preferably begins at the point $r_s=r_g+\rho$. By shifting in the direction of (i.e. of the angle of cam's rotation), it is possible to find that transition point (also known as point of reversal) at which the curves obtained by the solutions of the differential equations contact (tangentially) each other.

In the event that a point of transition is located externally of the "useful" portion of the cam surface (i.e. externally of that zone of the cam surface which produces displacement of the roller follower), a second curve can be calculated in continuation of the curve corresponding to the useful zone of the cam surface (i.e. to that zone where the piston connected to the follower roller performs a stroke). For the second curve, the pressure in the working chamber of the fuel injection pump must be considered zero in the same way as in the calculation of curvature at the high point of the cam surface.

Without further analysis, the foregoing will so fully reveal the gist of the present invention that others can, by applying current knowledge, readily adapt it for various applications without omitting features that, from the standpoint of prior art, fairly constitute essential characteristics of the generic and specific aspects of this invention and, therefore, such adaptations should and are intended to be comprehended within the meaning and range of equivalence of the following claims.

What is claimed as new and desired to be secured by Letters Patent is:

1. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising, in combination: a rotary radial cam having a peripheral cam surface, a roller follower having a center, means for constantly biasing the follower into contact with the cam surface, and means for reciprocably guiding the follower in the radial directions of said cam when the latter rotates, the curvature of at least that portion of the cam surface which brings about a cam action upon the follower being such that, at a given relation of load upon the follower and of the angle of rotation of said cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, and the pitch curve of the follower's center about said cam surface being determined by the differential equation

$$\frac{d^2r_s}{d\epsilon^2}\bigg|_1\cdot\frac{\rho}{r_s^2\left[1+\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2\right]^2} = \frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2 l\rho}P_s + \frac{\rho}{r_s}\cdot\frac{1+2\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2\right]^2} - \frac{1}{\sqrt{1+\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2}}$$

in which

P_s is the load in kg. in the direction in which the follower reciprocates and acting upon said portion of the cam surface,

p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},

r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,

ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,

ρ is the radius of the follower in mm.,

E is the modulus of elasticity in kg./mm.^{2},

ν is the Poisson ratio, and

l is the active width of the follower and of the cam.

2. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising, in combination: a rotary radial cam having a peripheral cam surface, a roller follower having a center, means for constantly biasing the follower into contact with the cam surface, and means for reciprocably guiding the follower in the radial directions of said cam when the latter rotates, said follower consisting of a material different from the material of said cam, the curvature of at least that portion of the cam surface which brings about a cam action upon the follower being such that, at a given relation of load upon the follower and of the angle of rotation of said cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, and the pitch curve of the follower's center about said cam surface being determined by the differential equation

$$\frac{d^2r_s}{d\epsilon^2}\bigg|_1\cdot\frac{\rho}{r_s\left[1+\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2\right]^2} = \frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2 l\rho}P_s + \frac{\rho}{r_s}\cdot\frac{1+2\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2\right]^2} - \frac{1}{\sqrt{1+\left(\dfrac{dr_s}{r_s d\epsilon}\right)^2}}$$

in which

P_s is the load in kg. in the direction in which the follower reciprocates and acting upon said portion of the cam surface,

p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},

r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,

ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,

ρ is the radius of the follower in mm., E is the modulus of elasticity in kg./mm.^{2} determined by the equation $1/E=\frac{1}{2}(1/E_1+1/E_2)$ wherein E_1 is the modulus of elasticity of the follower and E_2 is the modulus of elasticity of the cam,

ν is the Poisson ratio, and

l is the active width of the follower and of the cam.

3. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a radial cam having a peripheral cam surface and rotated at substantially uniform rotational speed, a roller follower having a center, means for reciprocably guiding the follower in the radial directions of said cam when the latter rotates, and spring means for constantly biasing the follower into contact with said cam surface, the curvature of at least that portion of the cam surface which brings about a cam action upon the follower being such that, at a given relation of total load upon the follower—including the useful load, inertia forces and the force of said spring means in the directcion in which the follower reciprocates—and of the angle of rotation of the cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, and the pitch curve of the follower's center about said cam surface being determined by the differential equation

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2\left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-\nu^2)p^2l\rho}\right\}$$
$$=\frac{E}{2\pi(1-\nu^2)p^2l\rho}[P_s+P_{fm}-c(r_g+\rho+h_m-r_s)]$$
$$+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}$$

wherein

$\left.\frac{dr_s}{d\epsilon^2}\right|_2$ is greater than

$$\frac{\frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2l\rho}P_s+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}}{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}}$$

P_s is the useful load in kg. in th direction in which the follower reciprocates and acting upon said portion of the cam surface,

p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},

r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,

ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,

ρ is the radius of the follower in mm.,

E is the modulus of elasticity in kg./mm.^{2},

ν is the Poisson ratio,

l is the active width of the follower and of the cam,

P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement,

c is the elasticity constant of the spring means in kg./mm.,

m is the mass of moving parts in kg. sec.^{2}/mm.,

r_g is the radius of the base circle of the cam at zero travel in mm.,

h_m is the maximum follower displacement in mm., and

ω is the rotational speed of the cam in 1/sec.

4. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a rotary radial cam having a peripheral cam surface, a roller follower having a center, means for constantly biasing the follower into contact with the cam surface, said follower reciprocable in the radial directions of said cam when the latter rotates, and bearing means for reciprocably guiding the follower, the curvature of at least that portion of the cam surface which brings about a cam action upon the follower being such that, at a given relation of total load upon the follower—including the useful load and the frictional forces between the follower and said bearing means—and of the angle of rotation of said cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, and the pitch curve of the follower's center about said cam surface being determined by the differential equation

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_1\cdot\frac{\rho}{r_s\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}=\frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}P_s$$
$$+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}$$

in which

P_s is the useful load in kg. in the direction in which the follower reciprocates and acting upon said portion of the cam surface,

p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},

r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,

ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,

ρ is the radius of the follower in mm.,

E is the modulus of elasticity in kg./mm.^{2},

ν is the Poisson ratio,

l is the active width of the follower and of the cam, and

μ is the coefficient of friction between the follower and the bearing means.

5. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a radial cam having a peripheral cam surface and rotated at substantially uniform rotational speed, a roller follower having a center and reciprocable in the radial directions of said cam when the latter rotates, spring means for constantly biasing the follower into contact with said cam surface, and bearing means for reciprocably guiding the follower, the curvature of at least that portion of the cam surface which brings about a cam action upon the follower being such that, at a given relation of total load upon the follower—consisting of useful load, of frictional forces between the follower and said bearing means, of pressure of said spring means, and of inertia force of the follower—and of the angle of rotation of the cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, and the pitch curve of the follower's center about said cam surface being determined by the differential equation

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2\left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-\nu^2)p^2l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}\right\}$$
$$=\frac{E}{2\pi(1-\nu^2)p^2l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}[P_s+P_{fm}-c(r_g+\rho+h_m-r_s)]$$
$$+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}$$

wherein

$$\left.\frac{d^2 r_s}{d\epsilon^2}\right|_2 \text{ is greater than}$$

$$\frac{\frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2 l\rho}P_s+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}}{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}}$$

P_s is the useful load in kg. in the direction in which the follower reciprocates and acting upon said portion of the cam surface,
p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},
r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,
ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation.
ρ is the radius of the follower in mm.,
E is the modulus of elasticity in kg./mm.^{2},
ν is the Poisson ratio.
l is the active width of the follower and of the cam,
P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement,
c is the elasticity constant of the spring means in kg./mm.,
m is the mass of moving parts in kg. sec.^{2}/mm.,
r_g is the radius of the base circle of the cam at zero travel in mm.,
h_m is the maximum follower displacement in mm.,
ω is the rotational speed of the cam in 1/sec., and
μ is the coefficient of friction between the follower and the bearing means.

6. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a radial cam rotated at substantially uniform rotational speed and having a peripheral cam surface consisting of a first arcuate portion whose curvature equals the curvature of the base circle of said cam and a second arcuate portion adapted to produce a cam action upon a follower, said second arcuate portion having a high point, a roller follower having a center and reciprocable in the radial directions of said cam by the second portion of said cam surface when the cam rotates, means for guiding said follower, and spring means for constantly biasing the follower into contact with said cam surface, the curvature of the second portion of said cam surface being such that, at a given relation of total load upon the follower—including the useful load, inertia forces and the force of said spring means in the direction of the follower's reciprocation—and of the angle of rotation of said cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, the pitch curve of the follower's center about the second portion of said cam surface save for the high point thereof being determined by the differential equation

$$\left.\frac{d^2 r_s}{d\epsilon^2}\right|_2\left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-\nu^2)p^2 l\rho}\right\}$$
$$=\frac{E}{2\pi(1-\nu^2)p^2 l\rho}\left[P_s+P_{fm}-c(r_g+\rho+h_m-r_s)\right]$$
$$+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}$$

and the pitch curve of the follower's center about said high point being determined by the differential equation

$$\left.\frac{d^2 r_s}{d\epsilon^2}\right|_3=-\frac{P_{fm}-c(r_g+\rho+h_m-r_s)}{m\omega^2_{max}}$$

wherein

$$\left.\frac{d^2 r_s}{d\epsilon^2}\right|_2 \text{ is greater than}$$

$$\frac{\frac{1}{2\pi(1+\nu^2)}\cdot\frac{E}{p^2 l\rho}P_s+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}}{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}}$$

wherein

$$\left.\frac{d^2 r_s}{d\epsilon^2}\right|_3 \text{ is greater than } \left.\frac{d^2 r_s}{d\epsilon^2}\right|_2$$

and wherein

P_s is the useful load in kg. in the direction in which the follower reciprocates and acting upon said second portion of the cam surface,
p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},
r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,
ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,
ρ is the radius of the follower in mm.,
E is the modulus of elasticity in kg./mm.^{2},
ν is the Poisson ratio,
l is the active width of the follower and of the cam,
P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement,
c is the elasticity constant of the spring means in kg./mm.,
m is the mass of moving parts in kg. sec.^{2}/mm.,
r_g is the radius of the base circle of the cam at zero travel in mm.,
h_m is the maximum follower displacement in mm.,
ω is the rotational speed of the cam in 1/sec., and
ω_{max} is the maximum desired rotational speed of the cam in 1/sec.

7. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a radial cam having a peripheral cam surface consisting of a first arcuate portion whose curvature equals the curvature of the base circle of said cam and a second arcuate portion adapted to produce a cam action upon a follower and having a high point, said cam being rotated at substantially uniform rotational speed, a roller follower having a center and reciprocable by the second portion of said cam surface in the radial directions of said cam when the latter rotates, spring means for constantly biasing the follower into contact with said cam surface, and bearing means for reciprocably guiding the follower, the curvature of the second portion of the cam surface being such that, at a given relation of total load upon the follower—including the useful load, the frictional forces between the follower and said bearing means, the pressure of said spring means, and the inertia force of the follower—and of the angle of rotation of said cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, the pitch curve of the follower's center about the second portion of said cam surface

save for the high point thereof being determined by the differential equation

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2\left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-\nu^2)p^2l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}\right\}$$

$$=\frac{E}{2\pi(1-\nu^2)p^2l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}[P_s+P_{fm}-c(r_g+\rho+h_m-r_s)]$$

$$+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}$$

and the pitch curve of the follower's center about said high point being determined by the differential equation

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_3=-\frac{P_{fm}-c(r_g+\rho+h_m-r_s)}{m\omega^2_{max}}$$

wherein

$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2$ is greater than

$$\frac{\frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2l\rho}P_s+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}}{\frac{\rho}{r_s\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}}$$

$\left.\frac{d^2r_s}{d\epsilon^2}\right|_3$ is greater than $\left.\frac{d^2r_s}{d\epsilon^2}\right|_2$

P_s is the useful load in kg. in the direction in which the follower reciprocates and acting upon said second portion of the cam surface,
p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},
r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,
ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,
ρ is the radius of the follower in mm.,
E is the modulus of elasticity in kg./mm.^{2},
ν is the Poisson ratio,
l is the active width of the follower and of the cam,
P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement,
c is the elasticity constant of the spring means in kg./mm.,
m is the mass of moving parts in kg. sec.^{2}/mm.,
r_g is the radius of the base circle of the cam at zero travel in mm.,
h_m is the maximum follower displacement in mm.,
μ is the coefficient of friction between the follower and the bearing means,
ω is the rotational speed of the cam in 1/sec., and
ω_{max} is the maximum desired rotational speed of the cam in 1/sec.

8. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a radial cam rotated at substantially uniform rotational speed and having a peripheral cam surface consisting of a first arcuate portion whose curvature equals the curvature of the base circle of said cam, a second arcuate portion adapted to reciprocate a follower when the cam rotates, and two third arcuate portions constituting the transitions from said first arcuate portion into said second arcuate portion, a roller follower having a center and reciprocable by said second arcuate portion and said third arcuate portions in the radial directions of the cam when the latter rotates, spring means for constantly biasing the follower into contact with the cam surface, and bearing means for reciprocably guiding the follower, the curvature of said second arcuate portion and of said third arcuate portions being such that, at a given relation of total load upon the follower—consisting of useful load, of frictional forces between the follower and said bearing means, of pressure of said spring means, and of inertia force of the follower—and of the angle of rotation of said cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and said cam, the pitch curve of the follower's center about said cam surface being determined by the differential equation

$$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2\left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-\nu^2)p^2l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}\right\}$$

$$=\frac{E}{2\pi(1-\mu^2)p^2l\rho}\cdot\frac{1}{1-\mu\frac{dr_s}{r_sd\epsilon}}[P_s+P_{fm}-c(r_g+\rho+h_m-r_s)]$$

$$+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}$$

and the curvature of said third arcuate portions being determined by the differential equation

$$m\omega^2\left.\frac{d^2r_s}{d\epsilon^2}\right|_4=P_{lm}\frac{1-\mu\frac{dr_s}{r_sd\epsilon}}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}-P_s-P_{fm}+c(r_g+\rho+h_m-r_s)$$

wherein

$\left.\frac{d^2r_s}{d\epsilon^2}\right|_2$ is greater than

$$\frac{\frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2l\rho}P_s+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_sd\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2}}}{\frac{\rho}{r_s\left[1+\left(\frac{dr_s}{r_sd\epsilon}\right)^2\right]^2}}$$

$$\left.\frac{dr_s}{d\epsilon}\right|_4=\left.\frac{dr_s}{d\epsilon}\right|_2$$

$$r_s|_4=r_s|_2$$

P_{lm} is the maximum permissible load upon the follower and said bearing means in kg.,
P_s is the useful load in kg. in the direction in which the follower reciprocates and acting upon said second arcuate portion and said third arcuate portions of the cam surface,
p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},
r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,
ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,
ρ is the radius of the follower in mm.,
E is the modulus of elasticity in kg./mm.^{2},
ν is the Poisson ratio,
l is the active width of the follower and of the cam,
P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement,
c is the elasticity constant of the spring means in kg./mm.,
m is the mass of moving parts in kg. sec.^{2}/mm.,
r_g is the radius of the base circle of the cam at zero travel in mm.,
h_m is the maximum follower displacement in mm.,
ω is the rotational speed of the cam in 1/sec., and
μ is the coefficient of friction between the follower and the bearing means.

9. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a radial cam rotated at substantially uniform rotational speed and having a peripheral cam surface consisting of a first arcuate portion whose curvature equals the curvature of the base circle of said cam, a second arcuate portion adapted to reciprocate a follower when the cam rotates, and two third arcuate portions constituting the transitions from the first arcuate portion into said second arcuate portion, a roller follower having a center and reciprocable by said second arcuate portion and by said third arcuate portions in the radial directions of the cam when the latter rotates, means for guiding said follower, and spring means for constantly biasing the follower into contact with said cam surface, the curvature of said second arcuate portion and of said third arcuate portions being such that, at a given relation of total load upon the follower—including the useful load, inertia force of the follower and the force of said spring means in the direction in which the follower reciprocates—and of the angle of rotation of the cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, the pitch curve of the follower's center about said second arcuate portion and said third arcuate portions being determined by the differential equation

$$\frac{dr_s}{d\epsilon^2}\bigg|_2\left\{\frac{\rho}{r_s\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-\nu^2)p^2l\rho}\right\}$$

$$\frac{E}{2\pi(1-\nu^2)p^2l\rho}[P_s+P_{fm}-c(r_g+\rho+h_m-r_s)]+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}$$

and the curvature of said third arcuate portions being determined by the equation

$$r_s=-(r_v-\rho)\cdot\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}+\sqrt{(r_g+r_v)^2-(r_v-\rho)^2\frac{\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}$$

P_s is the useful load in kg. in the direction in which the follower reciprocates and acting upon the second arcuate portion and the third arcuate portions of said cam surface,
p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},
r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,
ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,
ρ is the radius of the follower in mm.,
E is the modulus of elasticity in kg./mm.^{2},
ν is the Poisson ratio,
l is the active width of the follower and of the cam,
P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement,
c is the elasticity constant of the spring means in kg./mm.,
m is the mass of moving parts in kg. sec.^{2}/mm.,
r_g is the radius of the base circle of the cam at zero travel in mm.,
r_v is the smallest radius of concave curvature of said third arcuate portions in mm.,
h_m is the maximum follower displacement in mm., and
ω is the rotational speed of the cam in 1/sec.

10. A cam drive, particularly for the pistons of fuel injection pumps in internal combustion engines, comprising a radial cam rotated at substantially uniform rotational speed and having a peripheral cam surface consisting of a first arcuate portion whose curvature equals the curvature of the base circle of said cam, a second arcuate portion adapted to reciprocate a follower when the cam rotates, and two third arcuate portions constituting the transitions from the first arcuate portion into said second arcuate portion, a roller follower having a center and reciprocable by said second arcuate portion and by said third arcuate portions in the radial directions of the cam when the latter rotates, means for guiding said follower, and spring means for constantly biasing the follower into contact with said cam surface, the curvature of said second arcuate portion and of said third arcuate portions being such that, at a given relation of total load upon the follower—including the useful load, inertia force of the follower and the force of said spring means in the direction in which the follower reciprocates—and of the angle of rotation of the cam to time, a maximum permissible Hertzian compression prevails at all points of contact between the follower and the cam, the pitch curve of the follower's center about said second arcuate portion and said third arcuate portions being determined by the differential equation

$$\frac{d^2r_s}{d\epsilon^2}\bigg|_2\left\{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{Em\omega^2}{2\pi(1-\nu^2)p^2l\rho}\right\}$$

$$=\frac{E}{2\pi(1-\nu^2)p^2l\rho}[P_s+P_{fm}-c(r_g+\rho+h_m-r_s)]+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}$$

and the curvature of said third arcuate portions being determined by the equation

$$r_{vm}=\frac{r_s^2+\rho^2-r_g^2-2r_s\rho\frac{1}{\sqrt{1+(dr_s/r_s d\epsilon)^2}}}{2\left[r_g+\rho-r_s\frac{1}{\sqrt{1+(dr_s/r_s d\epsilon)^2}}\right]}$$

wherein

$\frac{d^2r_s}{d\epsilon^2}\Big|_2$ is greater than

$$\frac{\frac{1}{2\pi(1-\nu^2)}\cdot\frac{E}{p^2l\rho}P_s+\frac{\rho}{r_s}\cdot\frac{1+2\left(\frac{dr_s}{r_s d\epsilon}\right)^2}{\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}-\frac{1}{\sqrt{1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2}}}{\frac{\rho}{r_s^2\left[1+\left(\frac{dr_s}{r_s d\epsilon}\right)^2\right]^2}}$$

P_s is the useful load in kg. in the direction in which the follower reciprocates and acting upon the second arcuate portion and the third arcuate portions of said cam surface,
p is the maximum permissible Hertzian compression between the follower and the cam in kg./mm.^{2},
r_s is the distance between the axis of the cam and the pitch curve of the follower's center in mm.,
ϵ is the angle of rotation of the cam, in degrees, from the beginning of cam action to the point of computation,
ρ is the radius of the follower in mm.,
E is the modulus of elasticity in kg./mm.^{2},
ν is the Poisson ratio,
l is the active width of the follower and of the cam,

P_{fm} is the maximum spring pressure in kg. at a maximum follower displacement,

c is the elasticity constant of the spring means in kg./mm.,

m is the mass of moving parts in kg. sec.^{2}/mm.,

r_g is the radius of the base circle of the cam at zero travel in mm.,

r_{vm} is the radius of curvature at the points of transition from said first arcuate portion into said third arcuate portions,

h_m is the maximum follower displacement in mm., and

ω is the rotational speed of the cam in 1/sec.

References Cited in the file of this patent

UNITED STATES PATENTS

2,567,689	Bishop	Sept. 11, 1951
2,628,605	Jones et al.	Feb. 17, 1953

OTHER REFERENCES

"Cams, Elementary and Advanced," F. D. Furman, First Ed., 1921, published by Wiley and Sons, page 76, figure 64 relied on.

"Cams," H. A. Rothbart, 1956, page 23, published by Wiley and Sons.