Feb. 17, 1953 J. F. JONES ET AL 2,628,605

CAM MECHANISM AND METHOD FOR MANUFACTURING THE SAME

Filed Nov. 2, 1950 3 Sheets-Sheet 1

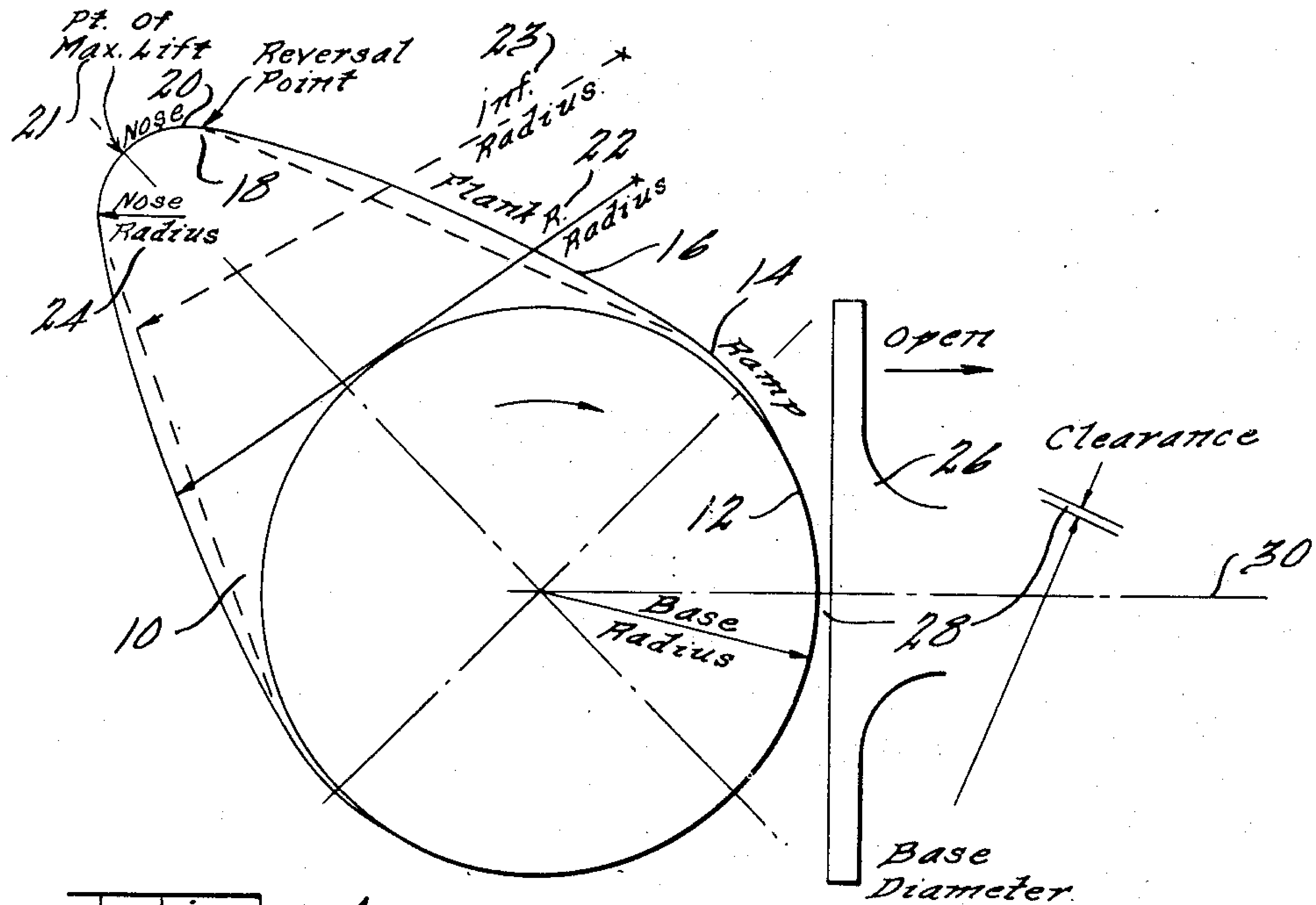

FIG. 1.

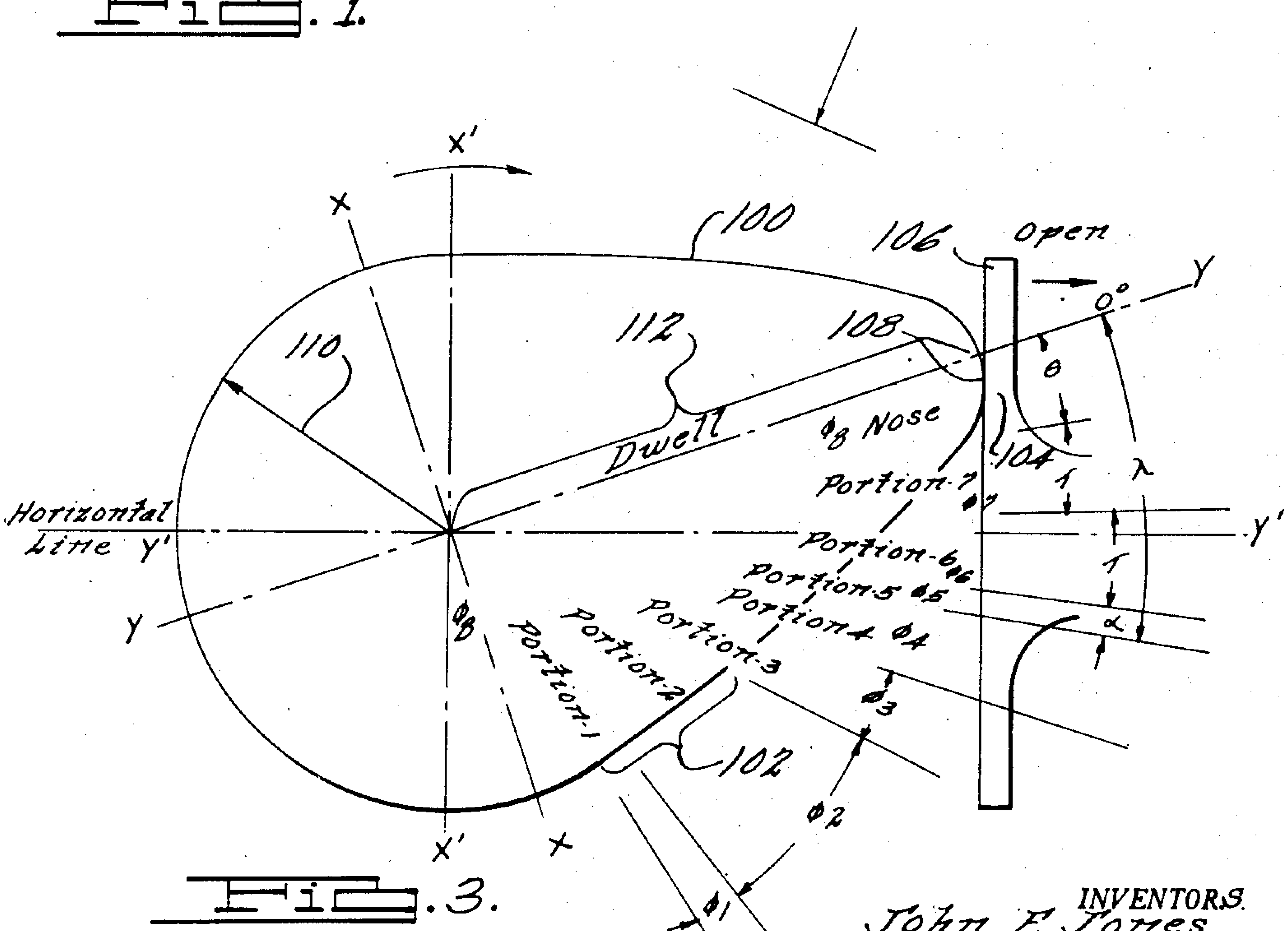

FIG. 3.

INVENTORS.
John F. Jones
Fred F. Timpner
Robert C. Juvinall
BY
Harness and Harris
ATTORNEYS.

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CAM MECHANISM AND METHOD FOR MANUFACTURING THE SAME

Filed Nov. 2, 1950 3 Sheets-Sheet 2

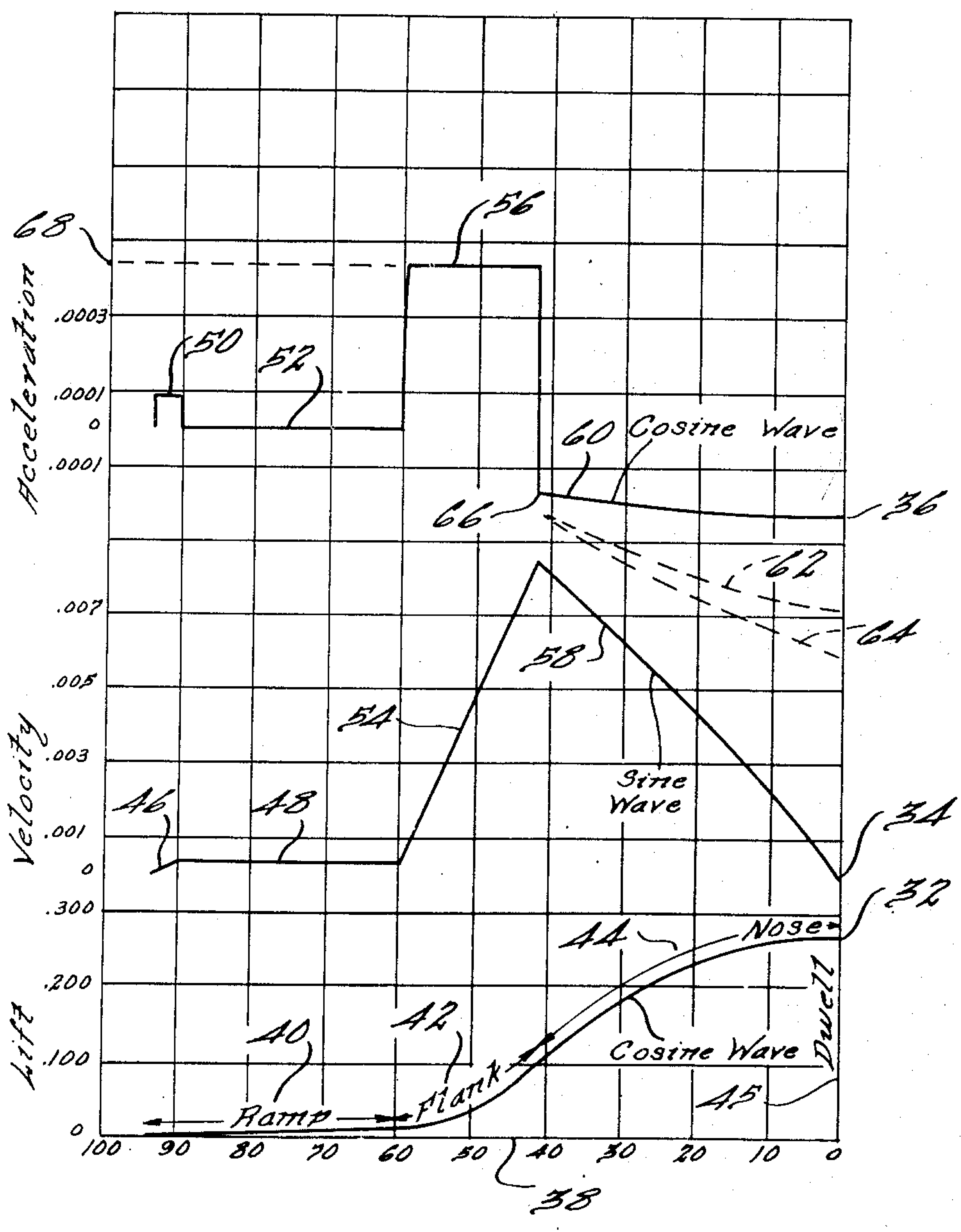

FIG. 2.

INVENTORS.
John F. Jones.
Fred F. Timpner.
Robert C. Juvinall.
BY
Harness and Harris
ATTORNEYS.

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Filed Nov. 2, 1950 3 Sheets-Sheet 3

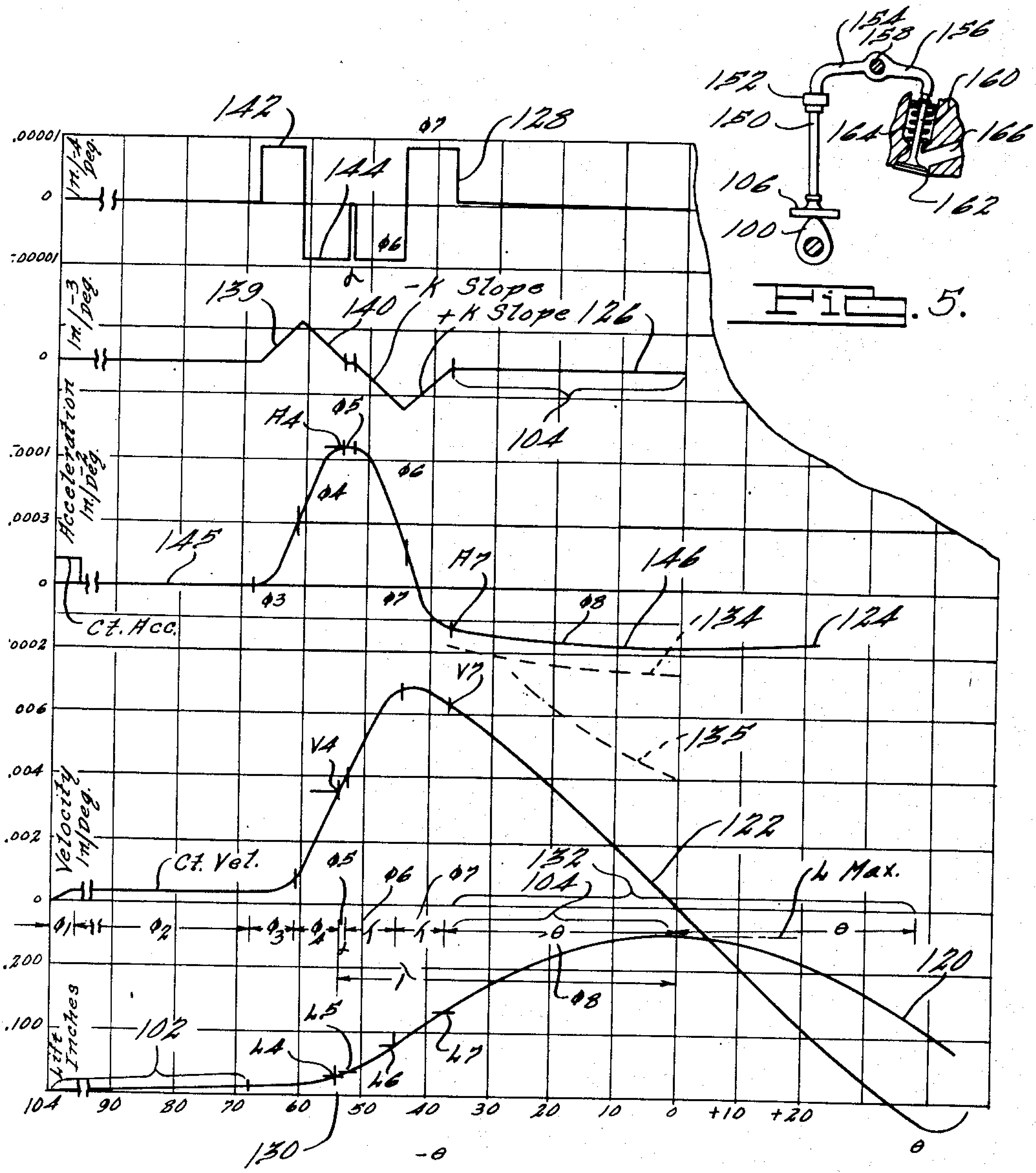

FIG. 5.

FIG. 4.

INVENTORS.
John F. Jones
Fred F. Timpner
Robert C. Juvinall
BY
Harness and Harris
ATTORNEYS.

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2,628,605

CAM MECHANISM AND METHOD FOR MANUFACTURING THE SAME

John F. Jones, Berkley, and Fred F. Timpner, Birmingham, Mich., and Robert C. Juvinall, Chicago, Ill., assignors to Chrysler Corporation, Highland Park, Mich., a corporation of Delaware

Application November 2, 1950, Serial No. 193,632

5 Claims. (Cl. 123—90)

1

This invention relates to cam mechanism and more particularly to cam surface contour and the dimensions to which the contour is formed for providing smooth and durable performance in service. The instant improvement is most needed in the valve train of high speed automotive engines which employ overhead valves, but is not necessarily limited to automotive engines and high speed cam trains of extended length having control over the operation of overhead valves.

Cams are traditionally cut to contours developed on the basis of circular arcs and straight line tangents in combinations such that the arcs are tangent at each end either to another circular arc or to a straight line tangent. True to be sure, the geometry and calculations are simple but the resulting behavior of the ordinary cam at high speeds is far from satisfactory where long trains are involved. High rate return springs for the cam follower are, of course, necessary to prevent bounce under such circumstances and even then other dynamic disturbances are likely to show up. Moreover, the excessively high rate of spring, to which resort must be made, is likely to cause pitting and galling at the cam nose owing to the maximum spring force being exerted over what turns out to be the minimum contact area exposed by the cam during its cycling.

According to a feature of the present invention, a cam is provided in which the maximum spring force and the maximum unit pressure on the cam face may be held to allowable limits even though the cam train is relatively long and operating at high speeds.

According to another feature, a cam is provided in which the acceleration forces applied to the cam follower are applied gradually.

According to still another feature of the invention is the provision of a cam mechanism in which the force necessary to decelerate the inertia of the train without bounce varies according to the spring force available. That is to say, as contrasted with the traditional practice of designing the cam and then selecting the spring believed desirable for the cam, the improved practice makes provision whereby the cam is contoured to behave according to the operating characteristics of the spring.

Further features, objects, and advantages will either be specificaly pointed out or become apparent when for a better understanding of the invention, reference is made to the following detailed description taken in conjunction with the accompanying drawings in which:

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Figures 1 and 3 each show respectively, a conventional cam and the improved cam;

Figures 2 and 4 are charts representing a graphic analysis of the behavior of the respective conventional cam and improved cam; and

Figure 5 is a fragmentary view of the improved cam applied to an overhead valve train.

As particularly respects Figures 1 and 2, the conventional cam **10** is shown in order that by way of contrast due appreciation may be made of the merits of the improved cam later to be described in detail. Conventional cam **10** is provided with a base circle **12** from the surface of which there arises a ramp **14** which blends into a flank **16**. It is the traditional function of the ramp **14** to take up the clearance between the cam and its follower and to cause the follower to commence movement at a relatively low constant velocity. Flank **16** may be indicated, as shown in dotted lines, to have an infinite radius **23** such that the cam is in effect straight-sided in the flank portion. What is preferred though as the usual variation in this practice, is for the flank to use a finite radius R such as indicated at **22** for the flank notwithstanding the fact that the end results approach one another as to the cam behavior. Flank **16** is tangent to a circular arc nose **20** at the reversal point indicated at **18**. In the center of the nose at point **21** is indicated the point of maximum lift which may also be called the point of dwell. Circular arc nose **20** is provided with a constant nose radius **24**. Point **18**, the point of reversal, so called, represents that point at which the motion of the follower abruptly undergoes the transition from acceleration to deceleration and is characterized by the fact that the follower tends to leave the surface of the cam but for the opposition of the follower spring. Follower **26** which cooperates with cam **10** is diagrammatically represented as adapted for rectilinear motion along the axis **30** and to be separated from the base circle of the cam by a clearance dimension indicated at **28**. Curves **32**, **34**, and **36** in Figure 2 show the cam behavior from the standpoint of lift, velocity, and acceleration respectively.

The values just enumerated are plotted against degrees of cam opening indicated at **38**. In lift curve **32** the ramp portion **40** merges with the flank portion **42** and flank portion **42** in turn merges with nose portion **44**. The lift behavior for the nose is interrupted in Figure 2 along the dwell axis **45** and the lift curve may be symmetrical on the other side or not as desirable. The nose portion **44** and, if desirable, the flank portion **42** of curve **32**, may comprise cosine waves. In velocity curve **34** the nose portion **58** of curve **34** will be seen to follow accordingly a sine wave. Flank portion **54** of velocity wave **44** may be either straight or conform to a relatively unpronounced sine wave. It will be noted that the ramp portions **46** and **48** in velocity curve **34** are formed such that the cam brings the follower up to the velocity indicated along the constant velocity portion **48** at a constant rate of change. In acceleration curve **36**, the ramp portions **50** and **52** correspond to the ramp portions **46** and **48** to velocity curve **34**. It will be seen in the acceleration curve **36** that for a part of the ramp travel acceleration is constant as indicated at **50**, and then reduces back to zero at **52** throughout the period of constant velocity of the cam mechanism. By constant velocity of the mechanism is meant constant velocity of travel of the cam follower inasmuch as the calculations here involved are based on the theory of constant cam rotation. In acceleration curve **36** a flank portion **16** of the cam will be seen to produce a constant acceleration along portion **56** to yield a constant rate of acceleration of value indicated at **68**. Since for a given inertia the force necessary is proportional to the acceleration required, it will be seen that the cam train by conventional design is subjected almost immediately from no force to an appreciable accelerating force corresponding value of acceleration indicated at **68**. The reversal point **18** of Figure 1 corresponds to the line of reversal **66** on the acceleration curve **36**. The acceleration in this instance drops instantaneously from a value indicated at **68** to the decelerative value of **66**. The cam opening forces then are no longer effective and it is the decelerative force of the cam train spring which is active to bring about the deceleration throughout travel across the cam nose. The decelerating portion **60** of acceleration curve **36** conventionally follows a cosine wave of rather flat proportions. As noted previously, since acceleration is in effect a measure of force involved, it has been convenient to superimpose curves of spring resistance in units of force as indicated at **62** and **64**. With proportional coordinates, it may be observed that at all points the curves **62** for one high rate spring performance and **64** for another high rate spring performance must at all times be spaced from the decelerating curve **60** in order to prevent valve bounce. It has been observed that as the spring rate is increased either by virtue of the relatively high inertia of a long valve train or because of the relatively high speed to which a long valve train must be operated, the spring force curves become more and more steep relative to the comparatively flat cosine wave **60**. The graphical analysis of Figure 2 then will bring out the fact that the excessive force available with the relatively steep curve as shown at **64** attains its maximum value at the tip of the nose of cam **10**, notably at point **21** in Figure 1. Yet the relatively smallest surface area of the cam is active when the tip of the nose is contacting the cam and with maximum spring force being there exerted, the unit pressures may be expected to reach values such as to cause failure and galling in the vicinity of the cam nose. Moreover, even though the spring force and the decelerating force necessary may be near enough to be compatible, still a broad consideration will show that the accelerating and the decelerating forces between curve portions **52** and **56** and curve portions **56** and **60** respectively will tend to cause dynamic disturbances later to show up in phases of the valve operation particularly in long value train arrangements.

As respects Figures 3 and 4, the improved cam **100** is provided with portions **1** and **2**, which together constitute a ramp **102**. The flank is constituted by portions **3**, **4**, **5**, **6**, and **7** and blends with a nose portion **104**. The intermediate angles of rotation corresponding to portions **1**, **2**, **3**, etc. correspond generally with the intermediate angles ϕ_1, ϕ_2, ϕ_3. The follower of the flat face type is indicated at **106** cooperating with the cam along the instantaneous point of contact **108**. Cam **100** is provided with a base circle radius **110** and a dimension of maximum lift **112** indicated along the dwell YY for the cam. The other principal axis for the cam XX is seen to be disposed at an angle ϕ_8 with X'X' axis for follower **106**. The Y'Y' axis for follower **106** is the axis along which cam **106** is adapted for rectilinear movement by action of cam **100**. Angle λ represents portions **5**, **6**, **7**, and the nose collectively; the intermediate angles throughout the entire angle λ are indicated as ϕ_5, ϕ_6, ϕ_7, and ϕ_8 which range between the limits respectively set at α,T,T, and θ. In Figure 4, curve **120** corresponds to the lift curve for cam **100**. Curve **122** corresponds to the velocity curve for cam **100**. Curve **124** corresponds to the acceleration curve for the cam **100**. Curve **126** corresponds to the rate of change of acceleration of the cam **100**, and curve **128** brings out those points at which the rate of change of acceleration of the cam is constant. As particularly regards acceleration curve **124** in Figure 4, a spring force curve has been superposed adjacent to it at **134**. Both spring force curve **134** and the acceleration curve portion **124** are shown in a relatively flat disposition but such showing is purely for convenience and the same principle would apply to the spring force curve V of the steeper slope **135**. One primary object of the instant invention being to correlate the cam behavoir to the physical behavior of the spring, it is desirable that the force required to decelerate the valve train be equal to or less than the force actually exerted by the spring. In equation form, for critical speed this statement may be represented as follows:

$$F_{\text{REQ.}}=F_{\text{SPR.}} \tag{1}$$

As is well known in the art the force required may be readily expressed in equation form thusly:

$$F_{\text{REQ.}}=\frac{WK}{g}\cdot\frac{d^2f(t)}{dt^2} \tag{2}$$

where:

W=equivalent weight at valve.
K=spring design factor.
g=386 in./sec.^{2}.
$f(t)$=lift curve at valve, or maximum lift M less instantaneous lift L_v.

Since the spring is presumed to be an elastic body such that its resistance is proportional to the displacement of the end thereof, the spring force is:

$$F_{\text{SPR.}}=B-Rf(t) \tag{3}$$

where:

B=spring load at maximum lift.
R=spring rate lb./in.

From equations 1, 2, and 3 the value $f(t)$ may be solved for by differential equations such as to yield:

$$(4)\qquad L_t=M-\frac{B}{R}\left[1-\cos\left(\sqrt{\frac{Rg}{KW}}\cdot t\right)\right]$$

or

$$(5)\qquad L=\frac{1}{c}\left[M-\frac{B}{R}\left(1-\cos\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{\pi N}\right)\right]$$

where:

C=ratio of valve lift to cam lift.
L=lift at cam rather than at valve.
N=engine R. P. M.
ϕ_8=degrees of cam rotation during t sec.

Velocity then equals:

(6)

$$V_{rad.}=\frac{dL}{d\phi_8}=-\frac{1}{C}\left[\frac{B}{R}\sqrt{\frac{Rg}{KW}}\,\frac{60}{\pi N}\sin\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{N\pi}\right]$$

where:

$$1\text{ radian}=\frac{180}{\pi}\text{ degrees}$$

In Figure 3 the sliding point of contact **108** has a Y′ coordinate equal to the instantaneous value of lift L. Therefore, y' may be equated to the value for L in equation 5:

$$(7)\qquad y'=L=\frac{M}{C}-\frac{B}{CR}\left(1-\cos\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{N\pi}\right)$$

Since tappet velocity numerically equals the distance of the sliding contact point **108** from the Y′Y′ axis the other coordinate for point **108** of value x' may be expressed as follows:

$$(8)\qquad x'=V_{rad.}=-\frac{B}{CR}\sqrt{\frac{Rg}{KW}}\,\frac{60}{\pi N}\sin\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{\pi N}$$

$$(9)\qquad \text{Let }\mu=\sqrt{\frac{Rg}{KW}}\,\frac{60}{\pi N}$$

The coordinates of the point of sliding contact **108** with reference to the cam axis XX, YY at the angle ϕ_8 to axes X′X′, Y′Y′ may be formulated as follows:

$$(10)\qquad x=x'\cos\phi_8+y'\sin\phi_8$$

$$(11)\qquad y=y'\sin\phi_8+y'\cos\phi_8$$

Sutiable differentiations will yield:

(12)

$$(12)\qquad \frac{dy}{dx}=\frac{\frac{dy}{d\phi_8}}{\frac{dx}{d\phi_8}}=-\tan\phi_8$$

(13)

$$\frac{d^2y}{dx^2}=\frac{d^2y}{dxd\phi_8}\cdot\frac{d\phi_8}{dx}=\frac{-1}{\left[\frac{B}{CR}(1-\mu^2)\cos\mu\phi_8+\left(\frac{M}{C}-\frac{B}{CR}\right)\right]\cos^3\phi_8}$$

Appropriate substitution of the values of the first and second derivatives Equations 12 and 13 into the general equation for the radius of curvature of a curve $y=f(x)$ will yield:

(14)

$$\text{Radius}=-\left[\frac{M}{C}-\frac{B}{CR}(1-\cos\mu\phi_8+\mu^2\cos\mu\phi_8)\right]$$

Since μ generally works out greater than one for high speed engine work the minimum radius of curvature occurs when ϕ_8 equals zero and may be represented therefor as:

$$(15)\qquad \text{Radius}_{\min}=-\left[\frac{M}{C}-\frac{B}{CR}\mu^2\right]$$

The velocity on the nose may be rewritten in terms of degrees rather than radians:

$$(16)\qquad V=-\frac{1}{C}\left[\frac{B}{R}\sqrt{\frac{Rg}{KW}}\,\frac{1}{3N}\sin\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{N\pi}\right]$$

Acceleration on the nose then is:

$$(17)\qquad A=\frac{dv}{d\phi_8}=\frac{1}{C}\left[\frac{Bg}{KW}\left(\frac{1}{3N}\right)^2\cos\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{N\pi}\right]$$

where ϕ_8 is the instantaneous angularity of rotation of the cam on the nose. Since inertia forces vary with the square of the speed of a moving object such as a valve train, inertia problems become increasingly important in high speed engine work; acceleration being a direct measure of the instantly active inertia forces, it is of advantage to have the acceleration increase gradually rather than to be applied and withdrawn suddenly. It is the function of the flank portion of the improved cam of Figures 3 and 4 to accomplish this acceleration gradually and at the same time to blend smoothly with the nose portion just discussed. The ramp **102** is conventional in that, as is a fact that curves **120**, **122**, and **124** will bring out, the cam mechanism undergoes constant acceleration A_1 for an instant, then zero acceleration and constant velocity V_2. At some point during the theoretically constant velocity V_2 period the clearance is taken up. As depends on the point at which this clearance is taken up then, the value for lift at the end of portion **1** is:

$$(18)\qquad L_1=\frac{1}{2}A_1\phi_1^2$$

At the end of portion 2 at constant velocity, the lift is:

$$(19)\qquad L_2=V_2\phi_2+L_1=V_2\phi_2+\frac{1}{2}A_1\phi_1^2$$

Portions **3** and **4** bring the cam train to maximum acceleration in a gradual fashion such that the rate of change of acceleration on portions **3** and **4** of the cam is constantly increasing at the rate K as is indicated at **139** and is constantly decreasing at the rate K as is indicated at **140** on curve **126**. Hence for portion **3**:

$$(20)\qquad A_3=\frac{1}{2}K\phi_3^2$$

$$(21)\qquad \frac{d}{d\phi_3}A_3=K\phi_3$$

$$(22)\qquad V_3=\frac{1}{6}K\phi_3^3+V_2$$

(23)

$$L_3=\frac{1}{24}K\phi_3^4+V_2\phi_3+L_2=\frac{1}{24}K\phi_3^4+V_2\phi_3+V_2\phi_2+\frac{1}{2}A_1\phi_1^2$$

At the end of portion **4**:

$$(24)\qquad A_4=\frac{1}{2}(-K)\phi_4^2+\frac{d}{d\phi_3}A_3\phi_4+A_3$$

$$(25)\qquad A_4=\frac{1}{2}(-K)\phi_4^2+K\phi_3\phi_4+\frac{1}{2}K\phi_3^2$$

$$(26)\qquad V_4=\frac{1}{6}(-K)\phi_4^3+\frac{1}{2}\left(\frac{d}{d\phi_3}A_3\right)(\phi_4^2)+A_3\phi_4+V_3$$

(27)

$$V_4=\frac{1}{6}(-K)\phi_4^3+\frac{1}{2}K\phi_3\phi_4^2+\frac{1}{2}K\phi_3^2\phi_4+\frac{1}{6}K\phi_3^3+V_2$$

(28)

$$L_4=\frac{1}{24}(-K)\phi_4^4+\frac{1}{6}\left(\frac{d}{d\phi_3}A_3\right)(\phi_4^3)+\frac{1}{2}A_3\phi_4^2+V_3\phi_4+L_3$$

(29)

$$L_4=\frac{1}{24}(-K)\phi_4^4+\frac{1}{6}K\phi_3\phi_4^3+\frac{1}{2}\cdot\frac{1}{2}K\phi_3^2\phi_4^2+\left(\frac{1}{6}K\phi_3^3+V_2\right)\phi_4+\frac{1}{24}K\phi_3^4+V_2\phi_3+V_2\phi_2+\frac{1}{2}A_1\phi_1^2$$

The magnitude of the acceleration A_4, velocity V_4, and lift L_4 at the end of portion **4** are then readily determinable by assigning real values in Equations 25, 27, and 29 for K, ϕ_4, ϕ_3, V_2, and A_1 as appropriate. In accordance with the plan of the invention, on portion **5** the angle ϕ_5 varies from zero value to value α; the acceleration A_5 is constant and equals acceleration A_4 at the end of portion **4** previously discussed. Further let the rate of change of acceleration decrease constantly at rate K for T degrees on portion **6**, and on portion **7** increase constantly at rate K for T degrees.

In order then that the acceleration and velocity curves at the end of portion **7** intersect the nose contour, then acceleration A_7 must equal the acceleration yielded by Equation 17 for the nose and the velocity V_7 must equal the velocity yielded by Equation 16 for the velocity at the nose. Let the final value for the acceleration in Equation 17 be represented as A_7 and the final value for the velocity in Equation 16 be indicated as V_7. Let the end value for the nose angle equal θ degrees:

(30) $$\lambda=\alpha+2T+\theta$$

(31) $$A_7=A_4+(-K)T^2$$

(32) $$V_7=-KT^3+A_4(\lambda-\theta)-V_4$$

A value of T is to satisfy the simultaneous Equations 31 and 32 is found to be:

(33) $$T=\frac{V_7-A_4(\lambda-\theta)-V_4}{A_7-A_4}$$

From Equations 30 and 33:

(34) $$\alpha=\lambda-\theta-2\left[\frac{V_2-A_1\{\lambda+(-\theta)\}-V_1}{A_7-A_4}\right]$$

From Equation 31:

(35) $$K_6=\frac{-A_7-A_4}{T^2}$$

By substituting assumed values for λ and θ in Equation 34 such that α yields a positive or zero value and by substituting the appropriate values of λ and θ in formulas 34 and 35 degrees arrive at the length for the various cam segments can be determined. As to portion **5** where the acceleration is constant ($A_5=A_4$): By appropriate integration:

(36) $$V_5=A_5\phi_5+V_4=A_4\phi_5+V_4$$

By appropriate integration:

(37) $$L_5=\frac{1}{2}A_4\phi_5^2+V_4\phi_5+L_4$$

The angle θ_5 increasing from zero value to value α. As to portion **6**:

(38) $$\frac{d}{d\phi_6}A_6=-K\phi_6$$

By integration:

(39) $$A_6=-\frac{1}{2}K\phi_6^2+A_5=-\frac{1}{2}K_6\phi_6^2+A_4$$

(40) $$V_6=-\frac{1}{6}K\phi_6^3+A_4\phi_6+A_4\phi_5+V_4$$

(41) $$L_6=-\frac{1}{24}K\phi_6^4+\frac{1}{2}A_4\phi_6^2+(A_4\alpha+V_4)\phi_6+L_5$$

The angle θ_6 increasing from zero value to value T as to portion **7**:

(42) $$A_7=\frac{1}{2}K\phi_7^2+\left(\frac{d}{d\phi_6}A_6\right)\phi_7+A_6$$

(43) $$A_7=\frac{1}{2}K\phi_7^2-K\phi_6\phi_7-\frac{1}{2}K\phi_6^2+A_4$$

By integration:

(44)

$$V_7=\frac{1}{6}K\phi_7^3+\frac{1}{2}(-K_6\phi_6)\phi_7^2+\left(-\frac{1}{2}K\phi_5^2+A_4\right)\phi_7-\frac{1}{6}K\phi_6^3+A_4\phi_6+A_4\alpha V_4$$

(45)

$$L_7=\frac{1}{24}K\phi_7^4+\frac{1}{6}(-KT)\phi_7^3+\frac{1}{2}\left(-\frac{1}{2}KT^2+A_4\right)\phi_7^2+\left(-\frac{1}{6}KT^3+A_4T+A_4\alpha+V_4\right)\phi_7+L_6$$

The angle ϕ_7 increasing from zero value to value T. It will be appreciated that when the assumed values for λ and θ satisfy the conditions of Equation 34 for the value α the radius of curvature of the nose portion may be easily computed from Equation 14 inasmuch as the angle ϕ_8 in Equation 14 varies from the value zero to the value θ. The lift curves for the ramp and flank blending into the nose may be readily computed according to the lift curves of the respective portions into which the ramp and flank are divided. Equations 18 and 19 yield the lift curves for the appropriate values of ϕ_1 and A_1 assumed and for the appropriate values of ϕ_2 and V_2 assumed. The lift curves for the first portions of the flank may be determined in accordance with Equations 23 and 29 once the values for angles ϕ_3, K, and ϕ_4 are established. Equations 33 and 35 yield respectively the values of T and K_6 by means of which the lift curves for portions **5**, **6**, and **7** of the flank may be computed through appropriate substitution in Equations 37, 41, and 45. A cam could then, according to the foregoing limits, behave in the manner desired. In particular regard to Figure 5 a cam, according to the instant invention, is shown in an appropriate setting. Cam **100** and follower **106** of the flat face type coact to operate a push rod **150** on the end of which is an adjustable cam tappet **152**. Tappet **152** cooperates with a rocker arm pivoted about pivot **158** and having arms **154** and **156** of relative lengths corresponding to the rocker arm ratio C. Arm **156** contacts the end of an overhead valve **160** seating at **162** over the end of a fluid passage **166**. The cam mechanism causes opening of valve **160** and valve seating as occasioned by the action of a valve spring **164**. Cam **100** is cut to the contours according to the chart of Figure 4 and will operate as follows. On the opening side of the cam the behavior is divided broadly into three categories as best brought out in curve **124**. That is, after the initial acceleration on the ramp, the valve **160** is opened first at the constant acceleration indicated at **145**, which is zero, secondly, at the constant acceleration indicated at ϕ_5 which is a constant maximum, and thirdly, along the portion of curve **124** indicated at **146** on the nose, which is of negative acceleration or deceleration. During the latter named portion **146**, it will be noted that the decelerative value or force generally parallels the spring force **134** which is available inasmuch as according to Equation 1 foregoing, the cam is cut so as to have such behavior as to the nose portion. In order to prevent dynamic disturbances from showing up in the cam train as, for instance, would be occasioned by push rod **150** compressing as a spring instantaneously and later expanding such that the action of valve **160** does not follow exactly the operation of actuating cam **100**, the transition from zero acceleration portions **145** on curve **124** to the maximum constant acceleration A_4 will be observed to be a gradually acting contour. As may be gathered from curve **126** the portion **139** corresponding to angle ϕ_3 is such that the rate of change of acceleration is constant in a positive fashion. Hence the second derivative of the acceleration, as indicated at **142** on curve **128** is constant. In order that for the acceleration surve ϕ_4 to act as a companion portion to portion ϕ_3 the rate of change of acceleration is shown at curve **126** in constant but of a negative character as at **140**.

It is to be observed that rather than have the acceleration vary instantaneously from zero to maximum value as in a conventional cam, the improved cam **124** manifests the behavior of a gradual transition to maximum acceleration. This gradual transition is of prime importance. On the decelerating side of the accelerating curve **124**, another gradual transition is effected from the portion of constant maximum acceleration indicated at ϕ_5 to the end of portion **7**, indicated at A_7 where the flank blends into the nose portion **146**. The rate of change of the acceleration throughout the portion ϕ_6 will be observed on curve **126** to be constant and of a negative character that is minus K slope throughout the portion ϕ_7 on curve **124** the rate of change of slope will be observed to be constant and of a positive character plus K. Hence in curve **128** the second derivative of the acceleration at ϕ_6 will be observed to be constant and the second derivative of the acceleration at ϕ_7 will be observed to be constant. Since the lift formulas were set up on the premise of the acceleration and velocity curves of portion **7** blending in with the nose, a smooth contour will result at the respective points L_7, V_7, and A_7 on curves **120**, **122** and **124** respectively. The duration of these respective portions as best seen in curve **122**, is accurately determined by formula such that the angle λ is always equal to the sum of the respective duration angles α, T, T, and θ. As particularly regards curve **124**, since the companion portions ϕ_3 and ϕ_4 and also the companion portions ϕ_6 and ϕ_7 are complementary in the respect that where one is of constant rate of change of acceleration of one character, the other is of constant rate of change of acceleration of the opposite character. The build-up then from zero acceleration to maximum acceleration and from maximum acceleration to deceleration as indicated by nose portion **146** is gradual and hence the forces applied will be gradual as contrasted with the theoretical instantaneous application of these same forces in the conventional cam. Since the lift curves and the curve for the radius of curvature of the improved cam are explicit as set forth in the foregoing, the cam contour may be accurately cut to produce a cam acting in accordance with the behavior graphically shown by the chart of Figure 4. Such approach to the problem is of advantage in ironing out dynamic disturbances, bounce, and coil oscillation as may be manifested when conventional cams are attempted to be used in the environment of a long overhead valve train operating at high speeds.

Variations within the spirit and scope of the invention described are equally comprehended by the foregoing description.

What is claimed is:

1. Mechanism including a curved cam and follower comprising a sliding pair in which the lift is in accordance with the following formula:

$$L=\frac{1}{C}\left[M-\frac{B}{R}\left(1-\cos\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{N\pi}\right)\right]$$

where:

L=Lift at cam.
M=Maximum lift at valve.
B=Total spring load at maximum valve lift.
R=Spring rate.
K=Spring factor.
W=Valve gear equivalent weight.
N=Engine R. P. M.
ϕ_8=Cam degrees from maximum lift.
C=Ratio of valve lift to cam lift.

2. A sliding pair comprising a cam and follower of which the lift of the latter varies during a cycle of the former as corresponds to the rotative position thereof, in which each cycle comprises three lift periods viz., a constantly increasing rate of change of acceleration, a constantly decreasing rate of change of acceleration, and a relatively high constant acceleration, in which the said three periods are obtained by employing a curved cam surface and flat faced follower cooperating to produce an acceleration whereof the second derivative yielded is constant during each said period.

3. A sliding pair comprising a cam and follower of which the lift of the latter varies during a cycle of the former as corresponds to the rotative position thereof, in which each cycle comprises a rising phase comprising three lift periods viz., a lift period of constant zero acceleration, a lift period of constant relatively high acceleration, and a lift period of negative acceleration, and transition periods therebetween, in which the aforesaid periods are obtained by employing a flat face on the cam follower in contact with working arcuate surfaces on the cam whereof the transition portions of the latter tangentially merge into the lift period portions cut for the aforesaid lift periods to produce consecutively adjacent segments of equal duration and amount of effect, one segment of each pair affording a change of rate of change of acceleration at constant increase and the other segment of each pair affording a change of rate of change of acceleration at constant decrease.

4. In the method of cam lifting a cam follower included in a train of valve gear comprising a normally closed valve connected to the cam follower and a valve spring opposing movement of opening of the valve caused by the lifting of the cam follower, the improved single step compris-

ing the lifting of the cam follower according to the following formula:

$$L=\frac{1}{C}\left[M-\frac{B}{R}\left(1-\cos\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{N\pi}\right)\right]$$

where:

L=lift at cam.
C=ratio of valve lift to cam lift.
M=Maximum lift.
B=spring load at maximum lift.
R=spring rate lb./in.
g=386 in./sec^2.
K=spring design factor of safety.
W=equivalent weight of valve gear at valve.
ϕ_8=cam degrees on cam nose from maximum lift increasing in the direction of rotation (negative values on opening side of cam), and
N=engine R. P. M.

so as to maintain a constant spring factor at all points of the cam follower on the cam nose between point of reversal and point of full lift and thereby maintain the spring resistance proportionately constant at all points.

5. In the method of cam lifting a cam follower included in a train of value gear comprising a normally closed valve connected to the cam follower and a valve spring opposing movement to open the valve caused by the lifting of the cam follower, the improvement comprising the steps of providing a cam having a radius of curvature on the nose in accordance with the general case of the expression:

$$\text{Radius}=-\left[\frac{M}{C}-\frac{B}{CR}(1-\cos\mu\phi_8+\mu^2\cos\mu\phi_8)\right]$$

wherein:

L=Base circle radius plus maximum lift.
B=Total spring load at maximum valve lift.
C=Spring rate.
ϕ_8=Cam degrees from maximum lift.

$$\mu=\sqrt{\frac{(R)(g)}{\frac{(\text{Spring Factor})}{(\text{Valve Gear Equiv. Wt.})}}}\;\frac{60}{(\pi)(\text{R.P.M.})}$$

and thereby lifting the cam follower by means of rotating the cam nose thereagainst according to the expression:

$$L=\frac{1}{C}\left[M-\frac{B}{R}\left(1-\cos\sqrt{\frac{Rg}{KW}}\,\frac{60\phi_8}{N\pi}\right)\right]$$

wherein:

L=lift at cam.
C=ratio of valve lift to cam lift.
M=maximum lift.
B=spring load at maximum lift.
R=spring rate lb./in.
g=386 in./sec.^{2}
K=spring design factor of safety.
W=equivalent weight of valve gear at valve.
ϕ_8=cam degrees on cam nose from maximum lift increasing in the direction of rotation (negative values on opening side of cam), and
N=engine R. P. M.

so as to maintain a constant spring factor and thereby maintaining the spring resistance exerted directly proportional to the valve gear deceleration.

JOHN F. JONES.
FRED F. TIMPNER.
ROBERT C. JUVINALL.

REFERENCES CITED

The following references are of record in the file of this patent:

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