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(71) Applicant: **HITACHI, LTD.**
6, Kanda Surugadai 4-chome Chiyoda-ku
Tokyo (JP)

(72) Inventor: **Kadomukai, Yuzo**
2625-3 Shinoinayoshi Chiyoda-mura
Niihari-gun Ibaraki-ken (JP)

Nakamura, Yozo
2625-2 Shimoinayoshi Chiyoda-mura
Niihari-gun Ibaraki-ken (JP)

Nagae, Hiromitsu
2585-2 Takaba Katsuta-shi
Ibaraki-ken (JP)

(74) Representative: **Paget, Hugh Charles Edward et al**
MEWBURN ELLIS 2 Cursitor Street
London EC4A 1BQ (GB)

(54) Centrifugal advance mechanism.

(57) A centrifugal advance mechanism has a plurality of rotating weights (5) which travel under centrifugal force outwardly from an axis of a driving shaft (1) against the action of resilient means (11) urging the weights inwardly. The weights carry driving cam members (6) which interact by cam action with a driven cam member (8) which is thereby driven in rotation by the weights. The relative rotational position of the driven cam member (8) and the driving shaft (1) is thus determined by the radial position of the weights relative to the driving shaft (1). To improve performance of the mechanism under rapid rotational speed fluctuations, by means of the geometry of the weights (5) and the interacting cam surfaces of the driving and driven cam members (6,8), the inertial force of (a) the weights (5) and (b) the driven cam member (8) arising upon angular deceleration of the driving shaft (1) is controlled so as to have no outward action on the weights (5) over at least a part of the outward travel of the weights.

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Description

CENTRIFUGAL ADVANCE MECHANISM

This invention relates to a centrifugal advance mechanism, particularly such mechanism for use in the distributor of an ignition system of an internal combustion engine.

Centrifugal advance mechanisms in the control of the ignition timing of internal combustion engines are well known. The mechanism has a plurality of rotating weights, usually two, which travel under centrifugal force outwardly from the axis of a driving shaft against the action of resilient means, usually helical springs, urging the weights inwardly. The weights carry driving cam members which act upon a driven cam member which is thus driven in rotation by the weights. The relative rotational position of the driven cam member and the driving shaft is determined by the radial position of the weights relative to the driving shaft. Thus the rotational position of an output shaft connected to the driven cam member relative to the driving shaft is determined by the rotational speed of the driving shaft. Such a mechanism advances the ignition timing in dependence upon the rotational speed of the engine. In principle, it might be used to retard the rotational position of the output shaft, and the invention is applicable to such an arrangement. However, it will here be described in connection with the advancement of the output shaft, as in ignition timing.

The reason for ignition timing advance in an internal combustion engine is that, when a spark plug fires in the engine, the petrol-air mixture takes some time to achieve complete combustion. Therefore, at high engine speeds, the ignition timing should be advanced to cause the plug to fire before the piston reaches the top dead centre position. The ignition timing will therefore vary with the engine speed, and this is achieved automatically using a centrifugal advance mechanism.

Fig. 11 of the accompanying drawings shows the construction of a conventional centrifugal advance mechanism used in a distributor. Fixed on a driving shaft 1 is a plate 2 carrying pivots 3 for two weights 5. Springs 11 extend between spring pins 4 on the plate 2 and spring pins 7 on the weights 5, and act to pivot the weights inwardly towards the shaft axis 1 about the weight pivots 3. The weights carry driving cam members in the form of pins 6 extending parallel to the axis of the shaft 1 and engaged in cam holes 10 in the driven timing lever 8. The timing lever 8 is fixed to the output shaft 9 which is mounted on but freely rotatable around the driving shaft 1.

When the driving shaft 1 rotates counterclockwise in direction R as seen in Figs. 11 and 12, the weights 5 swing outwardly around the pivots 3 under the action of centrifugal force, against the resilient action of the springs 11. There is thus a balance point at each rotational speed, at the position where the moment of the tension in the springs 11 is equal to the moment of the centrifugal force. The radial position of the weight pins 6 through their cam action in the cam holes 10 determines the rotational position of the output shaft 9 relative to the driving shaft 1. Thus the device establishes a relationship in accordance with the rotational speed of the driving shaft 1 and the spring constant of the springs 11.

The conventional centrifugal advance mechanism described above has been designed without consideration for the effect of rapid fluctuation in the angular velocity of the driving shaft, i.e. angular accelerations and decelerations of the driving shaft. Such accelerations have in recent years greatly increased, with the use of lighter and more powerful petrol engines, and cause very rapid variation of angular velocity even during one rotation. It can be said that ten years ago this angular acceleration was of the order of 2000 to 3000 radians/s^2, whereas in recent engines and engines under development, it may amount to 120000 radians/s^2. This variation of angular acceleration arises from the engine cycle, i.e. compression and combustion in the several cylinders.

In a conventional centrifugal advance mechanism, the rapid flucuation in angular velocity leads to unstable advance characteristics. This is explained by Fig. 12 of the accompanying drawings, and results from the inertial force of the weights 5 and the timing lever 8 under this angular acceleration. The acceleration produces a force Fw at the centre of gravity 12 of the weight 5 which, when the angular acceleration of the driving shaft 1 is negative, creates a moment Fw.e around the pivot 3 tending to swing the weight outwardly around the pivot 3. Likewise, a moment -Fw.e is generated in the weight 5 tending to close it inwardly from the constant velocity balancing point, when the angular acceleration is positive.

On the other hand, if the moment of inertia around the axis of the driving shaft 1 of the timing lever 8 is It, a moment It.α is generated as shown in Fig. 12, when the angular acceleration of the shaft 1 is negative (-α) and this moment is transmitted to the weights 5 through the contact point of the cam holes 10 and the weight pin 6. The moment tends to swing the weights more outwardly from the balancing point. Conversely, when the angular acceleration of the shaft 1 is positive (α), a moment -It.α is generated in the timing lever 8, and through the cam contact this tends to swing the weights inwardly from the balancing point.

In general, ignition takes place in the compressive stroke of the engine, under the influence of the centrifugal advance mechanism. During this compressive stroke, the angular acceleration of the driving shaft 1 is negative and consequently, for the reason given above, the weights 5 at this instant are at a radial position further from the shaft 1 than is the case without angular acceleration. The solid line a in Fig. 13 illustrates the case without angular acceleration, whereas the increased advance angle caused by the deceleration of the shaft 1 is indicated by broken line b. This increased advance causes deterioration in the combustion in the engine, resulting in the phenomonon of knocking. Fig. 13 indicates that the difference of the advance angle from the design characteristic is larger when the rotational speed (angular velocity) of the drive shaft 1 is smaller, i.e. the angular acceleration variation has particularly bad effect on the advance angle at low rotational speeds.

In an attempt to deal with this problem, an additional element has been added to the centrifugal advance mechanism. This is shown in Japanese Laid-Open Utility Model Specification 61-41865. The additional element is a "stabiliser ring" which is an extra plate mounted on the shaft axis 1 and free to rotate, and having curved slots receiving the weight pins 6. These slots intersect the slots 10 in the timing plate at an angle, with the result that the inertial forces of the stabiliser ring and the timing plate clamp the weight pins during acceleration of the shaft 1, preventing movement of the weights under the influence of such acceleration. However, this device suffers from the defect that the weight pins which are contacted by the timing lever and by the stabiliser ring suffer rapid and uneven wear, and become unsatisfactorily deformed.

The object of this invention is to overcome, at least partly, the problem just described and to provide a centrifugal advance mechanism that has highly stable advance characteristics regardless of the variation of angular acceleration of the driving shaft.

In its broadest aspect, the invention is set out in claim 1. The stabiliser ring device described above uses an extra element to control the effect of the acceleration on the weights, while the present invention achieves this through the geometry of the weights and the interacting cam surfaces.

One way of carrying out the invention is set out in claim 2. Here, the geometry of the weights is selected so that the effect of the angular deceleration of the driving shaft is opposite to that in the conventional advance mechanism described above. Preferably the effect of the inertial forces arising upon angular deceleration urges the weights inwardly over at least 50%, more preferably 80%, of the radial travel of the weights. It is even possible to arrange the weights so that this effect of the inertial forces urges the weights inwardly over the whole of the travel of the weights. For the reason explained above, the influence of the angular deceleration is particularly great at low engine speeds, and accordingly it is preferred that the portion of the travel of the weights at which the inertial forces urge the weights inwardly includes the radially inner end portion of the travel of the weights. The effect of the inertial forcxe of the weights at the weight pivots is of course determined by the shape of the weights. It is particularly preferred in the present invention that the weights have concave surfaces facing radially outwardly from the driving shaft axis.

Another way of carrying out the invention is set out in claim 10. In this case the geometry of the cam surfaces is chosen to achieve the desired result of controlling the effect of the inertial force arising upon acceleration or decelreation of the driving shaft. The portion of the travel of the weights at which the inertial force of the driven cam member does not act to cause movement of the weights is preferably at the radially inner end of the travel of the weights, where the effect of the acceleration upon the timing advance characteristics is greatest, but there may be two such portions of the travel of the weights, one at the radially inner end of the travel and the other at a radially outer region of the travel of the weights.

Embodiments of the invention will now be described by way of non-limitative example with reference to the accompanying drawings, in which:-

Fig. 1 is an overall plan view of a centrifugal advance mechanism embodying the invention.

Fig. 2 is a sectional view at section A-A of Fig. 1.

Figs 3,4,5,6 and 7 are partial views explaining the forces that act in the centrifugal advance mechanism.

Fig. 8 is a graph showing the relationship between the maximum coefficient of friction and the angles θt and θw.

Figs. 9 and 10 are graphs showing the relationships between advance angle and θt and θw respectively.

Fig. 11 is an overall plan view of a conventional centrifugal advance mechanism.

Fig. 12 is a diagram explaining the forces that act in the mechanism of Fig. 11.

Fig. 13 is a graph showing examples of advance characteristics.

Fig. 14 is an overall view of an automotive distributor equipped with the centrifugal advance mechanism of Fig. 1.

In Figs. 1 to 7, showing the centrifugal advance mechanism embodying the invention, the reference numbers 1 - 12 are used for the parts corresponding in function and name to the parts having the same reference numbers in the conventional device of Fig. 11. These parts need not be described again except to explain how they differ.

In the centrifugal advance mechanism of Fig. 1, the centre of gravity 12 of each weight 5 is located as shown in Fig. 3, so that the tangent line to the arc around the axis of the driving shaft 1 passing through the centre of gravity 12, this tangent line extending from the centre of gravity 12 towards the weight pivot 3, lies between the axis of the driving shaft 1 and the line connecting the centre of gravity 12 and the weight pivot 3. In other words, the shape of the weights 5 and the position of the weight pivots 3 are determined so as to achieve this result. With this arrangement, the moment Fw.e around the weight pivot 3 generated by the inertial force Fw arising upon deceleratiion of the rotation of the shaft 1 tends to close the weight 5 inwards.

Thus the angle between (i) the radius line joining the centre of gravity 12 of the weight to the driving shaft axis 1 and (ii) the line joining the centre of gravity 12 of the weight to the pivot axis 3 is greater than 90°, over a range of radial positions of the weight including the inner end of the travel of the weight. This range is in practice more than 80% of the weights' travel but as the weight swings around the pivot 3 this angle decreases and may come below 90°. It can be seen from Figs. 1 and 3 that the outer surfaces of the weights 5 facing away from the shaft 1 are concave, which helps to locate the centre of gravity 12 in the desired position.

Secondly, the cam holes 10 in the timing lever 8 have the shape shown in Figs. 4,5,6 and 7 which is determined so that the angle θw between (i) the common normal line at the contact point of the weight pin 6 and the edge of the cam hole 10 and (ii) the direction of movement Mw of the weight 5 around the pivot axis 3 at this contact point is large (Fig. 4). On the other hand, the angle θt between (i) the common normal line at the contact point of the weight pin 6 and the edge of the cam hole 10 and (ii) the direction of movement Mt of the timing lever 8 around the axis of the shaft 1 at this contact point is small (Fig. 5).

In the case where no friction force exists between the weight pin 6 and the cam hole 10, i.e. the coefficient of friction μ is zero, the force F arising between the weight 5 and the timing lever 8 is equal to the normal force N that acts in the direction of the common normal line at the contact point of the weight pin 6 and the cam hole 10. Under the condition, the component of the normal force N applied by the timing lever 8 to move the weight 5 along the hole 10 is $N.\cos\theta w$. Likewise, the component of the normal force N applied by the weight 5 to move the timing lever 8 in rotation about the shaft 1 is $N.\cos\theta t$.

Next will be explained the case when a frictional force exists between the pin 6 and the cam hole 10, referring to Fig. 5 which shows how the weight 5 is driven, and Fig. 7 which shows how the timing lever 8 is driven. In this case, the force F acting between the weight 5 and the timing lever 8 is the resultant of the normal force N acting along the common normal line and the frictional force μN between the weight pin 6 and the cam hole 10, and the value of this resultant force F is equal to $N.(1+\theta^2)^{1/2}$.

If the angle between the resultant force F and the direction of motion Mw of the weight 5 is φw and the angle between the resultant force F and the direction of motion Mt of the timing lever 8 is φt, the following relationships exist: $\varphi w=\theta w+\tan^{-1}\mu$, $\varphi t=\theta t+\tan^{-1}\mu$, $\varphi w>\theta w$ and $\varphi t>\theta t$. Therefore, the component of the resultant force F applied by the timing lever 8 to move the weight 5 is $F.\cos\varphi w$, and $F.\cos\varphi w<N.\cos\theta w$. Likewise, the component of the resultant force F applied by the weight 5 to move the timing lever 8 is $F.\cos\varphi t$, and $F.\cos\varphi t<N.\cos\theta t$. As the coefficient of friction μ increases, the values of $F.\cos\varphi w$ and $F.\cos\varphi t$ decrease, and finally those values become zero when φw and φt reach 90°.

Some explanation of the shape of the cam holes 10 now follows.

The basic function of the centrifugal advance mechanism i.e. the advance characteristic has to be determined only by the centrifugal force generated in the weight 5. In other words, the advance characteristic should be determined by the rotational speed of the driving shaft 1. The weight pin 6 should freely drive the timing lever 8, but the weight pin 6, i.e. the weight 5, should not be easily driven by the moment of the inertial force of the timing lever 8.

The cam hole 10 should, in accordance with these conditions, have a shape such that the angle φt ($\varphi t=\theta t+\tan^{-1}\mu$, in other words θt) is as small as possible, in order to maintain $F.\cos\varphi t$ at a large value so that the weight pin 6 can drive the timing lever 8 easily. On the other hand, the shape of the cam hole 10 should be such that the angle φw ($\varphi w=\theta w+\tan^{-}\mu$, in other words θw) is as large as possible in order to maintain resultant force $F.\cos\varphi w$ at a low value so that the weight pin 6 should not be easily driven by the timing lever 8. As shown in Fig. 13, the region of large rate of advance is particularly where the rotational speed (angular velocity) of the driving shaft is low, that is when the advance is small. Consequently the cam hole 10 is shaped so that the angle θw between the resultant force F and the direction of motion Mw of the weight 5 is extraordinarily large.

All of the above explanations are qualitative. Here are some considerations from a quantitative point of view.

As is evident in the above, the cam hole 10 is formed so that the angle $\varphi t(=\theta t+\tan^{-1}\mu)$ should be as small as possible in order to maintain the component $F.\cos\varphi w$ at a large value. On the other hand, the angle $\varphi w(=\theta w+\tan^{-1}\mu)$ should be as large as possible in order to maintain the component value of the force $F.\cos\varphi t$ at a low value.

μmax is taken to be the maximum conceivable coefficient of friction between the cam hole 10 and the weight pin 6 during designing. The necessary condition for the weight pin 6 to drive the timing lever 8 is $\cos\varphi t>0$, that is $\varphi t<90°$. Using this condition, when the maximum coefficient of friction is μmax, the value of the angle θt is given by $\varphi t=\theta t+\tan^{-1}\mu max<90°$, i.e. $\theta t<90°-\tan^{-1}\mu max$. On the other hand, the necessary condition for the weight pin 6 not to be driven by the timing lever 8 is $\cos\varphi w<O$, that is $\varphi w>90°$. Using this condition, when the maximum coefficient of friction is μmax, the value of the angle θw is given by $\varphi w=\theta w+\tan^{-1}\mu max>90°$, i.e. $\theta w>90°-\tan^{-1}\mu max$.

Fig. 8 represents the value of the maximum coefficient of friction μmax and the values of θt and θw as explained above. The curve in Fig. 8 is given by $\mu=\tan(90-\theta)$.

Fig. 9 and Fig. 10 show θt and θw respectively against advance angle for the profile of the cam hole 10 in this embodiment of the invention. Fig. g shows generally a small value for θt of $-30°<\theta t<30°$. Fig. 10 shows a value for θw as much as $30°<\theta w<90°$. Fig. 10 also shows that θw is close to 90°, i.e. an especially large value, at the region of small advance angle where a large rate of advance often occurs.

In more detail, Figs. 9 and 10 show that at a region including the radially inner end (small advance angle) of the travel of the weights and at a further large advance region remote from the radially inner end, θt is close to zero, i.e. $-15°\leqq\theta t\leqq 15°$. Preferably at these regions $-10°\leqq\theta t\leqq 10°$ and more preferably $-5°\leqq\theta t\leqq 5°$. Similarly at these regions of small advance angle and large advance angle, θw approaches 90°, i.e. $60°\leqq\theta w\leqq 120°$. Preferably in these regions $80°\leqq\theta w\leqq 100°$, more preferably

$85° \leqq \theta w \leqq 95°$.

The profile of the cam hole shown in Figs. 1, 9 and 10 is only one example of the invention. The invention disclosed can generally be applied to achieve stable advance characteristics in centrifugal advance mechanisms of various advance characteristics, regardless of the variation of angular acceleration of the driving shaft.

Fig. 14 is an overall view of an automotive distributor in which the centrifugal advance mechanism of this invention is applied.

Fig. 14 shows a coupler 14 of a vacuum advance control mechanism (not illustrated) which adjusts ignition timing according to the load conditions of the engine (i.e. according to inlet manifold vacuum). The distributor has a stationary base 15 fixed to the distributor body 13. The coupler 14 is connected to a mobile base 16 that is mounted on the stationary base 15, but can rotate freely under the action of the coupler 14. A magnet 17 and a stator core 18 are fixed to the mobile base 16. The stationary base 15 has a ignition timing control device 20 with a detector coil 19. The engine drives the driving shaft 1 which is inserted into the rotor shaft 9. The rotor shaft 9 is housed together with retractors 21. Therefore, when the gap between the stator core 18 and the retractors 21 changes with rotation of the engine, the magnetic flux through the magnetic path composed of the magnet 17, the stator core 18, the retractor 21, the rotor shaft 9, and the mobile base 16, changes. This change of the magnetic flux generates a signal voltage at the detector coil 19, and through the signal voltage, the ignition timing control device 20 determines the spark timing. In a conventional manner this signal voltage alternates the time of generation and ignition in compliance with the vacuum advance control mechanism and the centrifugal advance mechanism of this invention.

The centrifugal advance mechanism of this invention can provide stable advance characteristics with little or no effect caused by the angular acceleration fluctuation of the engine. Ignition timing is controlled properly and combustion improved.

Claims

1. A centrifugal advance mechanism having a plurality of rotating weights (5) which travel under centrifugal force outwardly from an axis of a driving shaft (1) against the action of resilient means (11) urging the weights inwardly, the weights carrying driving cam members (6) which interact by cam action with a driven cam member (8) which is thereby driven in rotation by the weights, the relative rotational position of the driven cam member (8) and the driving shaft (1) being determined by the radial position of the weights relative to the driving shaft (1), characterised in that, by means of the geometry of the weights (5) and the interacting cam surfaces of the driving and driven cam members (6,8), the inertial force of at least one of (a) the weights (5) and (b) the driven cam member (8) arising upon angular deceleration of the driving shaft (1) is controlled so as to have no outward action on the weights (5) over at least a part of the outward travel of the weights.

2. A centrifugal advance mechanism having a plurality of rotating weights (5) which travel under centrifugal force outwardly from an axis of a driving shaft (1) against the action of resilient means (11) urging the weights inwardly, the weights carrying driving cam members (6) which interact by cam action with a driven cam member (8) which is thereby driven in rotation by the weights, the relative rotational position of the driven cam member (8) and the driving shaft (1) being determined by the radial position of the weights relative to the driving shaft (1), characterized in that, over at least a portion of the travel of the weights (5), the inertial forces of the weights at their respective centres of gravity arising upon angular deceleration of the driving shaft (1) urge the weights (5) inwardly.

3. A mechanism according to claim 2 wherein said inertial forces arising upon angular deceleration urge the weights (5) inwardly over at least 50% of the travel of the weights.

4. A mechanism according to claim 2 wherein said inertial forces arising upon angular deceleration urge the weights (5) inwardly over at least 80% of the travel of the weights.

5. A mechanism according to claim 2 wherein said inertial forces arising upon angular deceleration urge the weights (5) inwardly over the whole of the travel of the weights.

6. A mechanism according to any one of claims 2 to 5 wherein the portion of travel of the weights at which said inertial forces arising upon angular deceleration urge the weights inwardly includes the radially inner end of the travel of the weights.

7. A mechanism according to any one of claims 2 to 6 wherein each weight (5) is carried by a pivot (3) which is moved by the driving shaft (1) around the driving shaft axis, with the centre of gravity (12) spaced from the pivot (3) and located relative to the axis of the pivot (3) so that, at said portion of the travel of the weights, the angle between the line joining the driving shaft axis to the centre of gravity of the weight and the line joining the centre of gravity to the weight pivot axis is greater than 90°, whereby the inertial forces arising upon angular deceleration urge the weights inwardly.

8. A mechanism according to any one of claims 2 to 7 wherein said weights have concave surfaces facing radially outwardly from the driving shaft axis.

9. A mechanism according to any one of claims 2 to 8 wherein the cam surfaces of the driving cam members (6) and the driven cam member (8) are so shaped that, over at least one portion of travel of the weights (5), the inertial force of the driven cam member (8) arising upon acceleration or deceleration of the driving shaft (1) does not act to cause move-

ment of the weights (5) inwardly or outwardly.

10. A centrifugal advance mechanism having a plurality of rotating weights (5) which travel under centrifugal force outwardly from an axis of a driving shaft (1) against the action of resilient means (11) urging the weights inwardly, the weights carrying driving cam members (6) which interact by cam action with a driven cam member (8) which is thereby driven in rotation by the weights, the relative rotational position of the driven cam member (8) and the driving shaft (1) being determined by the radial position of the weights relative to the driving shaft (1), characterized in that the cam surfaces of the driving cam members (6) and the driven cam member (8) are so shaped that, over at least one portion of travel of the weights (5), the inertial force of the driven cam member (8) arising upon acceleration or deceleration of the driving shaft (1) does not act to cause movement of the weights (5) inwardly or outwardly.

11. A mechanism according to claim 10 wherein said portion of the travel of the weights at which the inertial force of the driven cam member does not act to cause movement of the weights is at the radially inner end of the travel of the weights.

12. A mechanism according to claim 10 having two said portions of the travel of the weights, one said portion being at the radially inner end of the travel and the other said portion being at the radially outer end of the travel.

13. A mechanism according to any one of claims 10 to 12 wherein, at the or each said portion of the travel of the weights at which the inertial force of the driven cam member does not act to cause movement of the weights, the angle θw between (i) the common normal line at the contact point of the driving cam member (6) with the driven cam member (8) and (ii) the direction of the centrifugal travel of the driving cam member at said contact point is given by $60° \leqq \theta w \leqq 120°$.

14. A mechanism according to claim 13 wherein $80° \leqq \theta w \leqq 100°$.

15. A mechanism according to any one of claims 10 to 14 wherein, at the or each portion of the travel of the weights at which the inertial force of the driven cam member does not act to cause movement of the weights the angle θt between (i) the common normal line at the contact point of the driving cam member (6) with the driven cam member (8) and (ii) the tangential direction relative to the driving shaft axis at said contact point is given by $-15° \leqq \theta t \leqq 15°$.

16. A mechanism according to claim 15 wherein $-10° \leqq \theta t \leqq 10°$.

17. A mechanism according to any one of claims 1 to 16 used in ignition timing of an internal combustion engine.

18. A distributor for an ignition system of an internal combustion engine having a mechanism according to any one of claims 1 to 16.

19. An ignition system of an internal combustion engine having a distributor according to claim 18.

FIG. 1

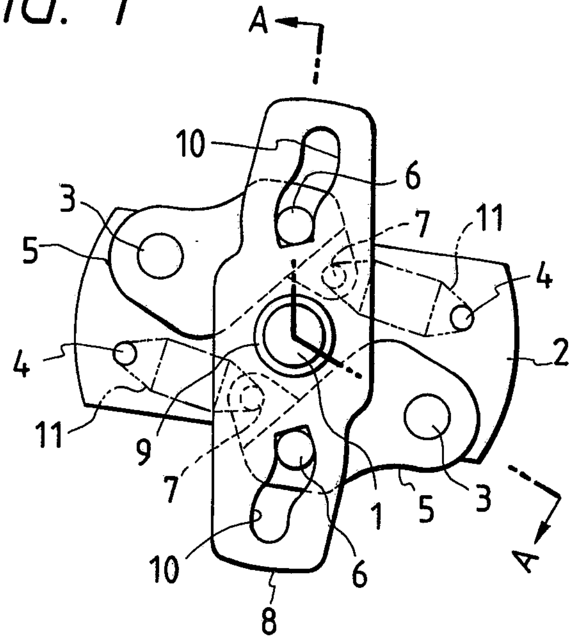


FIG. 2

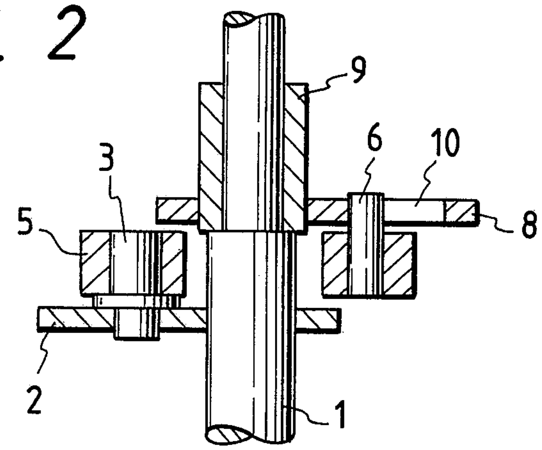


FIG. 3

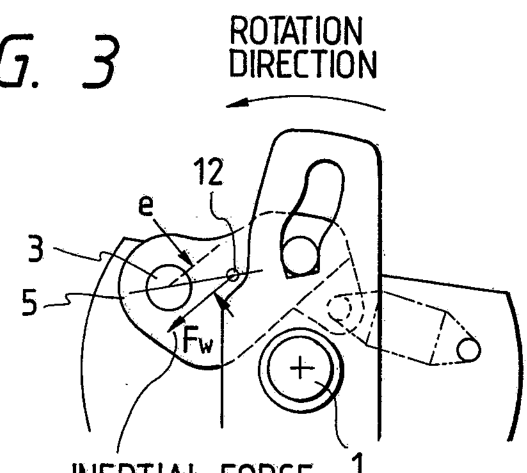


FIG. 4

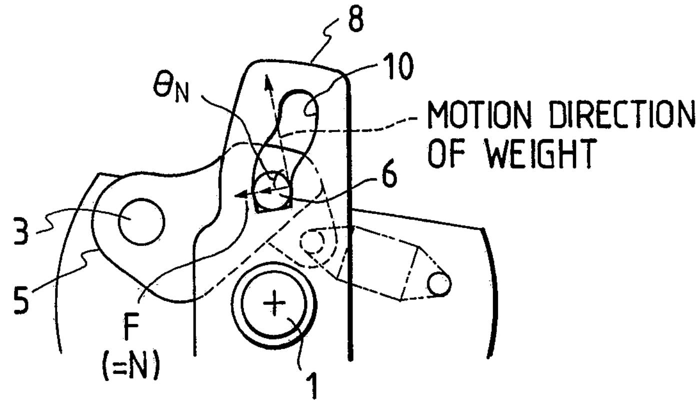


FIG. 5

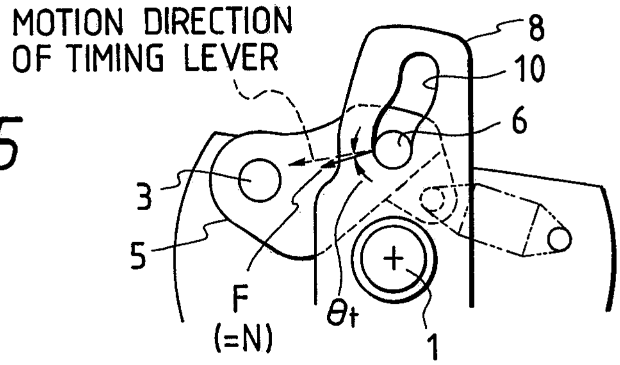


FIG. 6

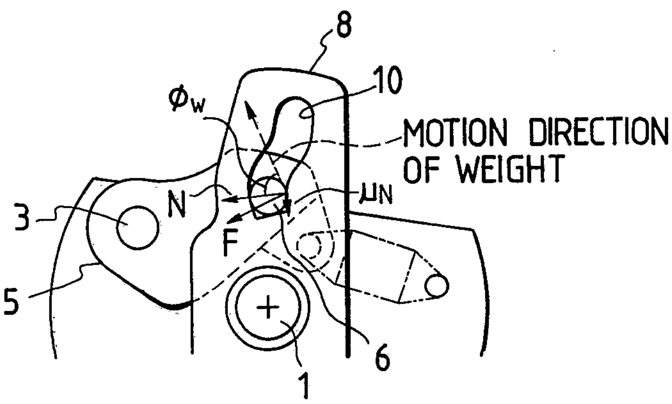


FIG. 7

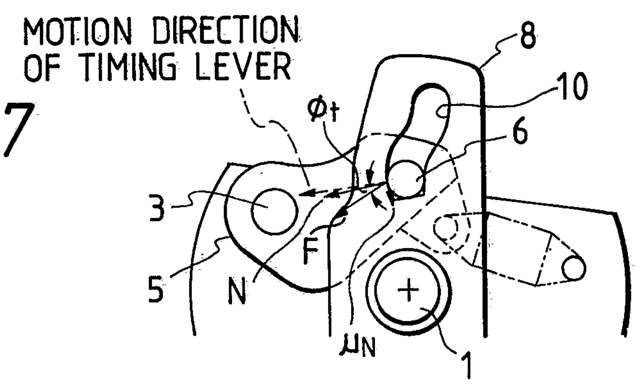
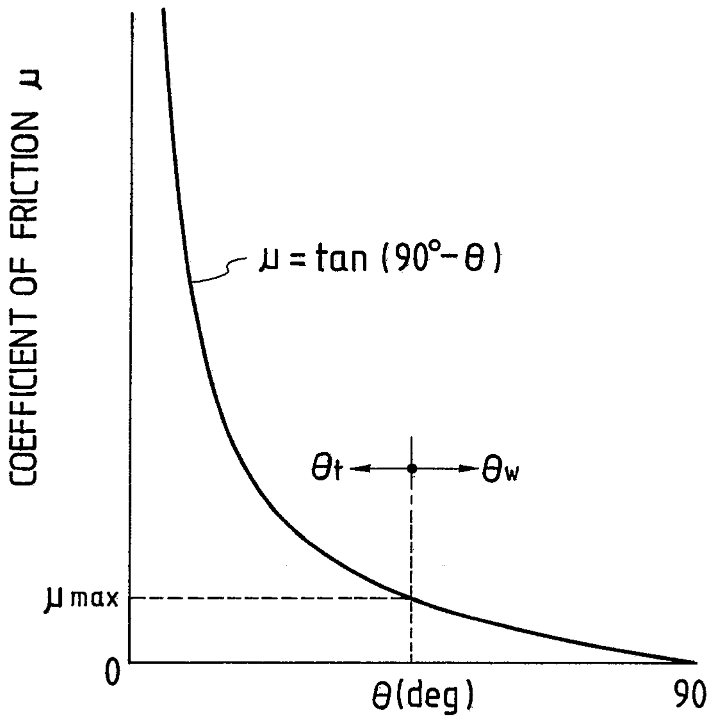


FIG. 8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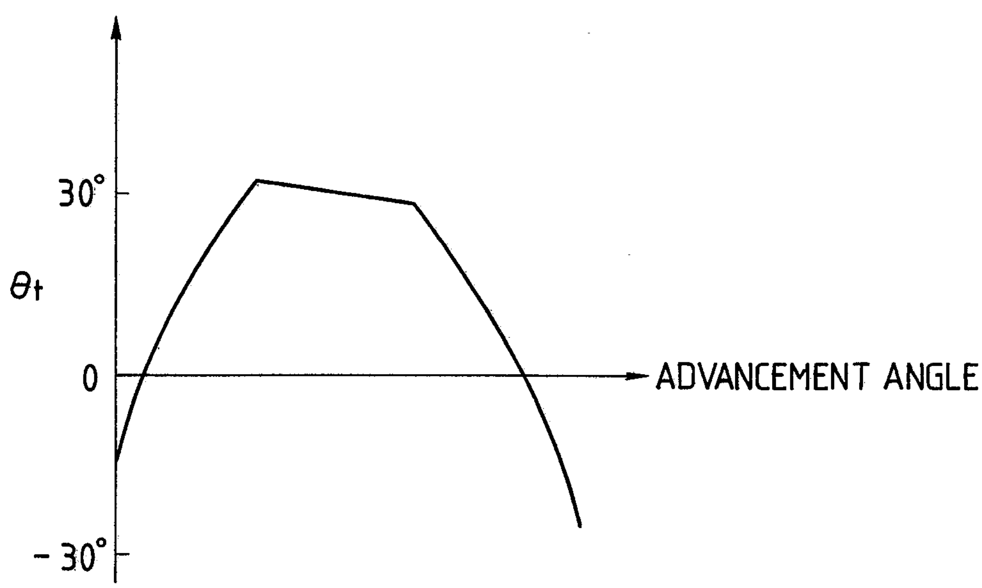
θt
θw
μ max
0
θ(deg)
90

FIG. 9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θt
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ADVANCEMENT ANGLE
− 30°

FIG. 10

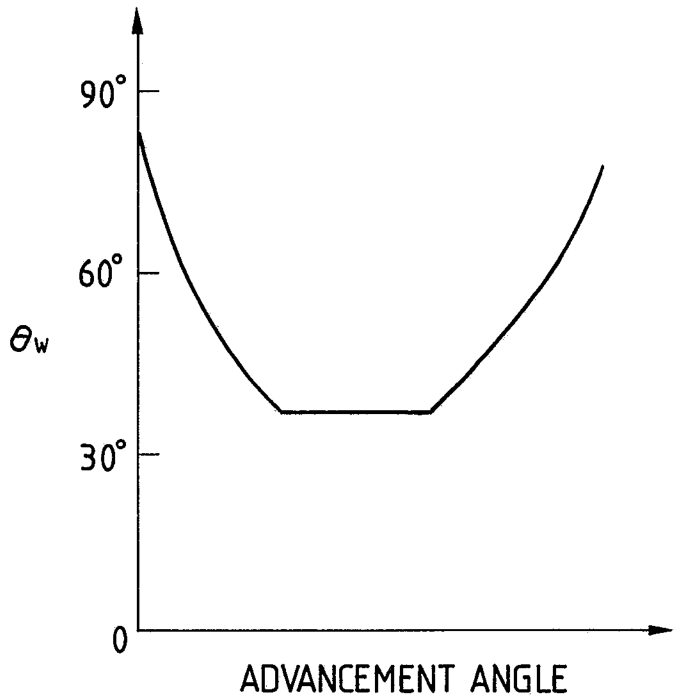

FIG. 11

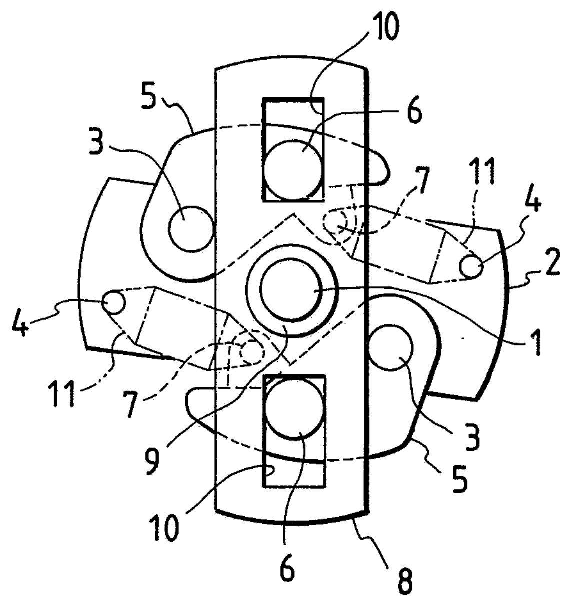

FIG. 12

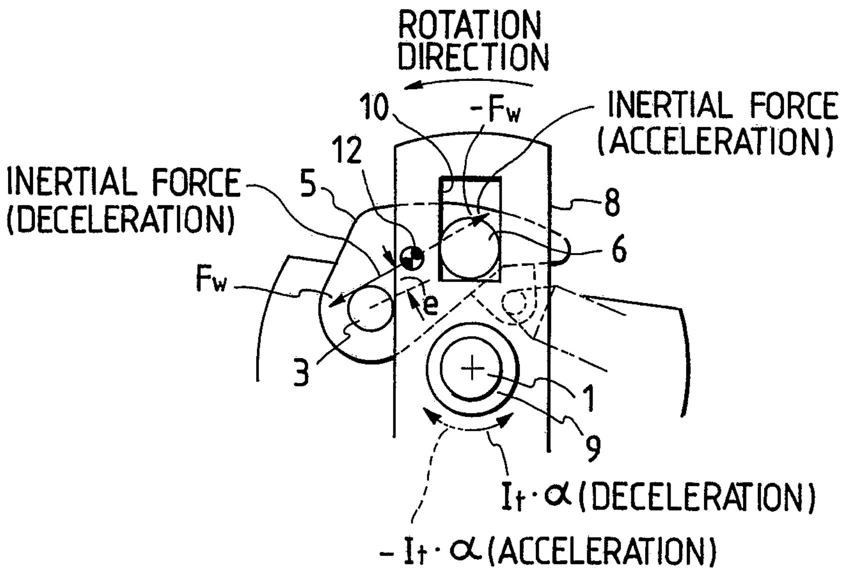
ROTATION
DIRECTION
10
-F_w
INERTIAL FORCE
(ACCELERATION)
12
INERTIAL FORCE
(DECELERATION)
5
8
6
F_w
e
3
1
9
$I_t \cdot \alpha$ (DECELERATION)
$-I_t \cdot \alpha$ (ACCELERATION)

FIG. 13

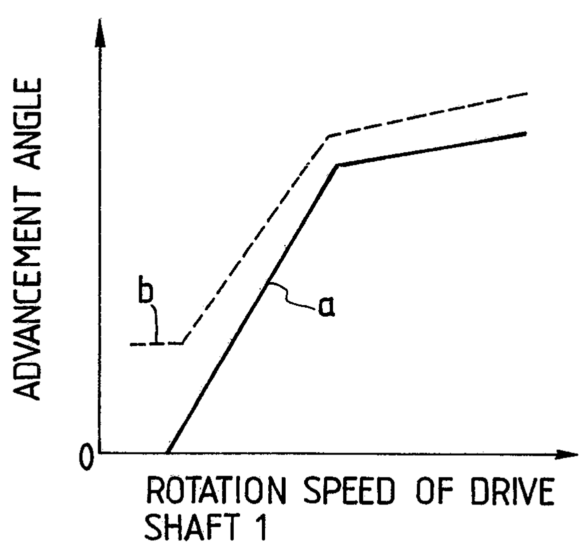
ADVANCEMENT ANGLE
b
a
0
ROTATION SPEED OF DRIVE
SHAFT 1

FIG. 14