

Nov. 25, 1969

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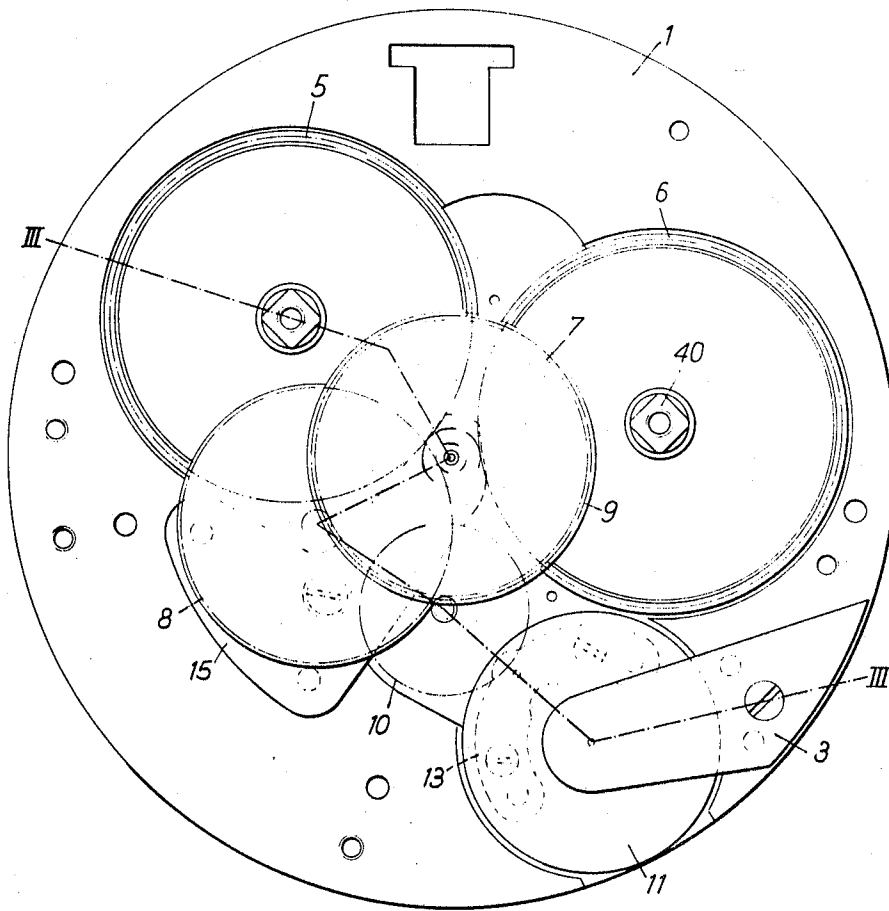
3,479,813

DRIVE SPRING ARRANGEMENT FOR WATCH MOVEMENT USING TWO BARRELS

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5 Sheets-Sheet 1

FIG. 1



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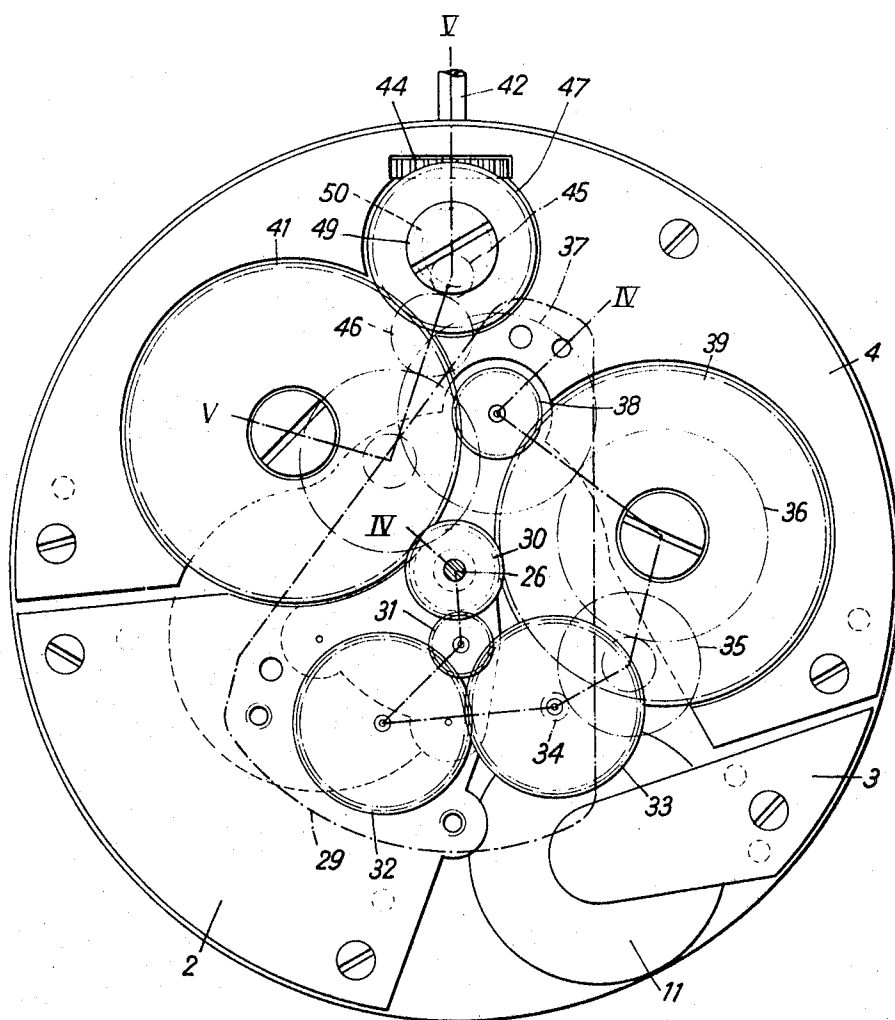
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FIG. 2



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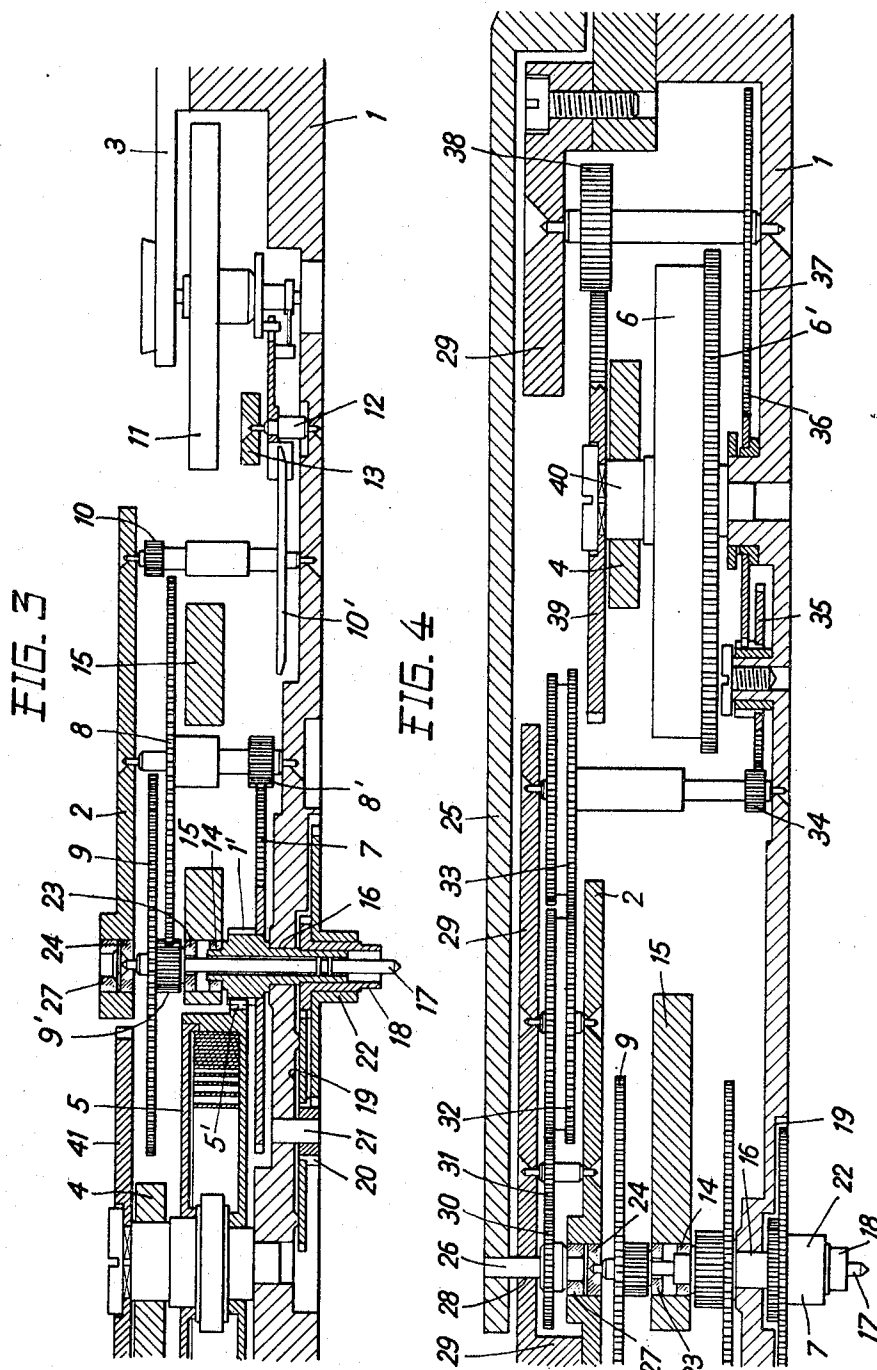
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DRIVE SPRING ARRANGEMENT FOR WATCH MOVEMENT USING TWO BARRELS

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DRIVE SPRING ARRANGEMENT FOR WATCH MOVEMENT USING TWO BARRELS

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FIG. 5

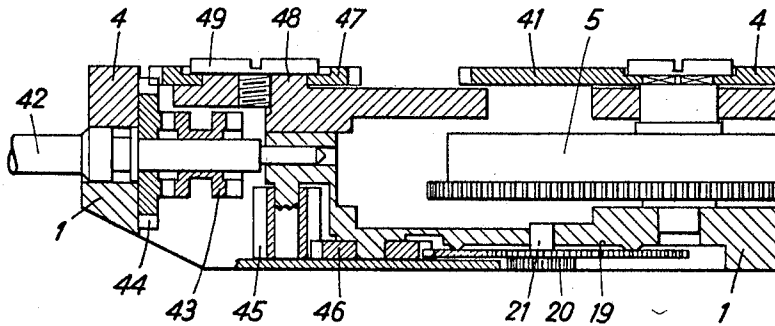
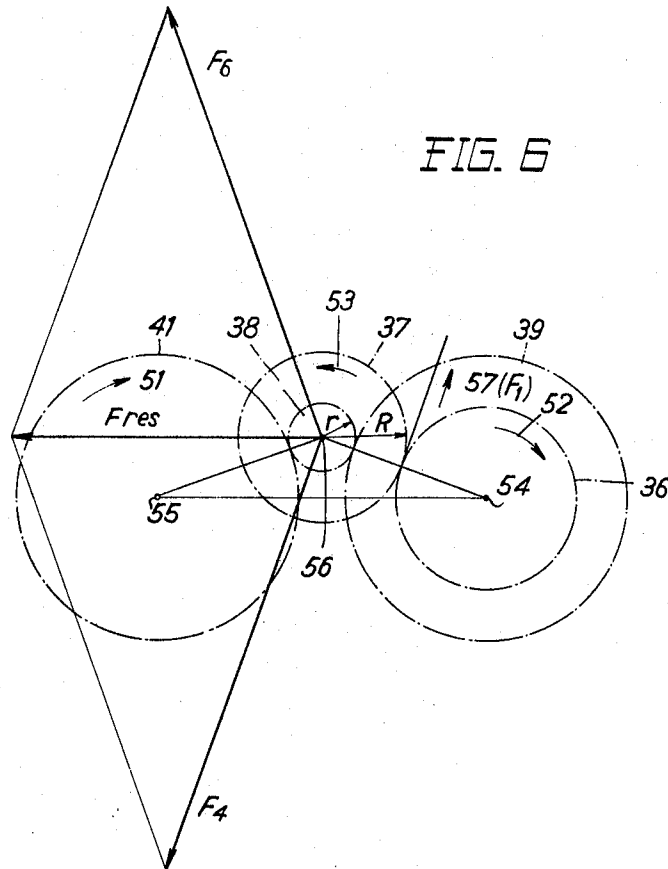


FIG. 6



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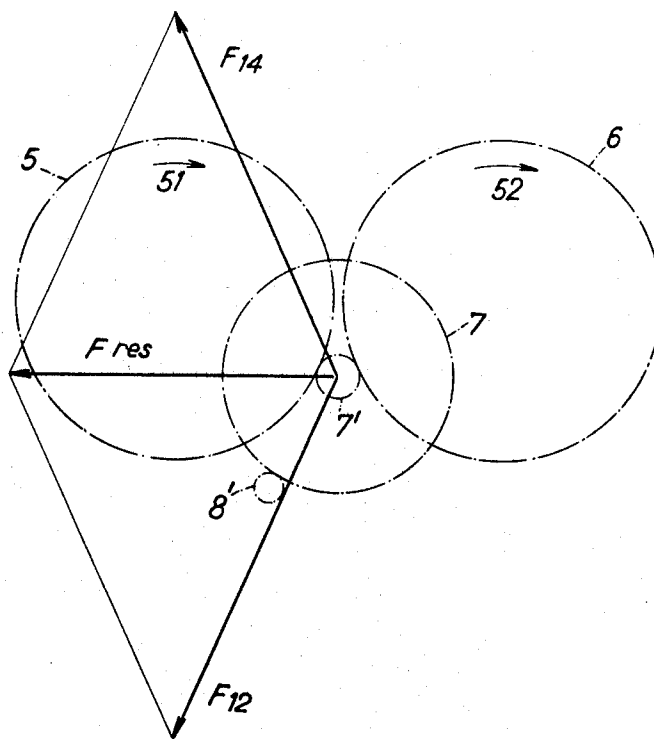
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DRIVE SPRING ARRANGEMENT FOR WATCH MOVEMENT USING TWO BARRELS

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FIG. 7



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DRIVE SPRING ARRANGEMENT FOR WATCH MOVEMENT USING TWO BARRELS

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Int. Cl. G04b 33/00, 1/16

U.S. Cl. 58—59

14 Claims

ABSTRACT OF THE DISCLOSURE

A watch movement having two separate drive springs housed in respective barrels which are mounted about parallel rotative axes, both said barrels being drivingly connected to the center-wheel runner of the time train, and including a self-winding train wherein the ratchet wheels of both barrels are connected to a third step-down runner in the winding train.

This invention relates to an improved power spring arrangement for mechanical watch movements of the type wherein the power spring means is required to drive the oscillating balance member at frequencies in the order of 30,000 times per hour and higher.

In watches it is known that the time keeping accuracy can be increased by increasing the oscillations of the balance mechanism above the customary 18,000 per hour to values in the order of 30,000 and more per hour. This results in an increase in accuracy at the cost, however, of increased energy consumption by the movement. Furthermore, the driving torque required in such circumstances is in the order of two or three times that required in conventional wrist watches. Such a higher driving torque in the rotative members necessitates correspondingly larger and sturdier bearings for said members, especially for the first runner of the time train, and for the last runner of the winding train in the event that a self-winding watch is involved.

Heretofore it has been found that if the first runner of the step-up train is a center-wheel runner upon whose shaft is carried the cannon pinion to which the minute hand is fastened, and if the drive spring is housed in a customary barrel, it is impossible to have a movement, and especially a wrist watch movement, of small enough dimensions in conformity with modern day requirements while still providing bearings which are sufficiently robust to withstand the higher torques applied to the rotative elements. This disadvantage prevails even if a self-winding weight which is pivoted coaxially with the center-wheel runner is omitted.

Watch movements are well known comprising two barrels, each housing a separate power spring of which one drives the time train of the movement while the other powers an alarm sounding mechanism. It has been proposed to modify this concept so that both springs are utilized to power the time gear train whereby the running time of the movement is prolonged to a considerable extent. In such an arrangement, the individual barrels would turn more slowly than in those movements using a single time train power spring, and the same applies to the first runner of the step-up gear train which is driven simultaneously by both barrels.

The use of two barrels, mounted about parallel axes relative to each other, could provide double the driving torque without disturbing the symmetrical disposition of parts as would occur if a single equivalent large barrel were used instead of two smaller ones. In instances where-

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in two barrels are employed, the time train can comprise five runners and the first runner thereof may be located in front of the center wheel. In other instances, the train comprises only four runners and the minute-indicating member is indirectly driven. Furthermore, it has been contemplated to employ an oscillating weight to achieve a self-winding of the two springs.

An object of this invention is to provide a high torque watch movement whose size is within the limitations of modern day standards.

A specific object is to provide an improved watch movement whose balance member oscillates more than 30,000 times per hour.

A further object is to provide a novel power spring arrangement whereby the bearing loads imposed upon the center-wheel runner and upon the winding train runner of the movement are considerably reduced relative to conventional power spring arrangements.

A further object is to provide a larger power spring means providing an increased reserve running time, especially for self-winding watches, without the larger power spring means disturbing the symmetrical arrangement of parts relative to the movement central axis.

A further object is to provide a novel arrangement of power spring means and the winding train therefor, in a self-winding watch, which arrangement is highly responsive to even small imbalances in the winding weight whereby the spring means is wound pursuant to even slight movements of the wearer's wrist.

The foregoing and other objects of this invention are generally realized through the provision of two separate drive springs, each housed in a respective barrel, the gear teeth of which are in engagement with a common single runner which is the first runner of the step-up gear train which drives the balance oscillator, said first runner being the center-wheel runner. As will be subsequently explained in detail, such an arrangement results in lower stresses being imparted to the bearings of the first wheel runner as compared to having an equivalent torque transmitted from a single barrel, and, consequently, the bearings may be of corresponding smaller size thereby resulting in a smaller watch movement.

The invention will be described in detail with reference to a preferred embodiment which is shown in the accompanying drawings wherein like reference characters in the several views denote like parts, and wherein:

FIGURE 1 is a top plan view of the movement according to this invention, various bridges and runners having been omitted for purposes of clarity;

FIGURE 2 is a view analogous to FIGURE 1 excepting that only the bridge of the self-winding mechanism has been omitted for clarity;

FIGURES 3 to 5 are sectional views taken along lines III—III, IV—IV, and V—V, respectively, in FIGURE 2;

FIGURES 6 and 7 are vector force diagrams related to bearings of respective runners of the movement.

With reference to FIGURES 1 and 3, the movement of this invention comprises two spring barrels 5 and 6, a center-wheel runner 7-7', a third wheel runner 8-8', a fourth wheel runner 9-9', an escape wheel 10' and a balance 11 all of which are mounted between circular plate 1 and respective ones of the time train bridge 2, the cock 3, and the barrel bridge 4. The escape lever 12 is pivoted in and between plate 1 and escape bridge 13 which is itself fastened to plate 1 and extends below the balance 11. An intermediate bridge 15 extends radially between the barrels 5 and 6 and above their respective gear teeth and supports a bearing 14 which constitutes the upper bearing for center-wheel runner 7.

The ring gears 5' and 6' of the respective barrels 5 and 6 both engage the pinion 7' of the center-wheel

runner and rotatively drive it at the rate of one revolution per hour. The step-up gear ratio between the center-wheel runner 7-7' and the third wheel runner 8-8' as well as the ratio between the third wheel runner and the fourth wheel runner are the usual ratios. On the other hand, the gear ratio between the fourth wheel runner 9-9' and the pinion 10' of the escape wheel runner 10-10' is 100 to 7. Furthermore, the escape wheel 10 has twenty-one teeth so that the balance 11 must oscillate 36,000 times per hour in order for the time train to turn at the conventional speed.

In FIGURE 3 it is seen that the center-wheel runner 7-7' comprises a hollow staff 16 through which extends the staff 17 of the fourth wheel runner 9-9', a cannon pinion 18 being coaxially mounted externally around an end portion of said staff 16. The cannon pinion 18, in turn, is housed in the conventional manner in an opening in plate 1 and meshes with a minute wheel 20 which is rotatively mounted on stud 21 which is fastened in plate 1. The minute wheel 20, in turn, meshes with and drives the hour wheel 22 which is coaxially mounted relative to the cannon pinion 18. The fourth wheel runner 9-9' is pivoted in two bearings, one of which 23 is mounted in bridge 15 and the other of which 24 is mounted in bridge 2.

A self-winding mechanism for winding the respective springs housed in barrels 5 and 6 is shown in FIGURES 2 and 4 and comprises a weight 25 rigid with shaft 26 and eccentrically disposed relative to the axis of said shaft. Shaft 26 is rotatively mounted in two bearings, one of which 27 is mounted in bridge 2 and the other of which constitutes an opening 28 extending through bridge 29. Bridge 29 serves as a bridge for the self-winding mechanism and is secured to bridge 2. Shaft 26 is rigid with a first pinion 30 of the winding train, pinion 30 being coaxial with the center-wheel runner 7-7'. The first pinion 30 meshes with an intermediate wheel 31 which, in turn, engages the two wheels 32 and 33 of a reverser assembly whose wheel 33 constitutes part of a first step-down runner 33-34, wheel 33 being rigid with pinion 34 which drives a second step-down runner 35 which is rotatively mounted about a cylindrical boss extending from plate 1 below barrel 6. Runner 35 drives intermediate wheel 36 which is coaxial with barrel 6, wheel 36 driving third step-down runner 37-38 whose pinion 38 engages the ratchet wheel 39 which is carried on the end of staff 40 of barrel 6 at the end opposite to plate 1.

It should be noted that the staffs of both runners, 33-34 and 37-38, are pivotably supported at their opposite ends in plate 1 and in bridge 29 of the self-winding mechanism, while the respective staffs of wheels 31 and 32 and the shaft 26 of oscillating weight 25 are pivotably supported in the time train bridge 2 and in bridge 29.

The inclusion of three step-down runners: 33-34, 35, and 37-38, provides for a high step-down ratio from the oscillating weight 25 to the winding barrels 5 and 6, whereby even a slight imbalance in said weight is sufficient to overcome the resistive torque of the ratchet wheels 39 and 41 in order for said weight to simultaneously wind the springs housed in said barrels.

The movement further comprises a winding stem 42 (FIGURE 5) which extends between the plate 1 and the barrel bridge 4, said stem being rotatively mounted between adjacent portions of said plate and bridge. Stem 42 is connectable to the other movement parts via a conventional transmission means comprising a clutch pinion 43 and a winding pinion 44, whereby said stem can either wind the power springs or set the indicating hands on the watch dial. In order to set the hands, a setting lever (not shown) brings the clutch pinion 43 into engagement with a first hour setting wheel 45 which, in turn, actuates a second or intermediate hour setting wheel 46 which is in engagement with the minute wheel 20. In order to wind the power springs, the clutch pinion 43 engages pinion

44 which drives the crown wheel 47 which, in turn, is rotatively mounted about a boss 48 and is held thereon by a screw member 49. Boss 48 includes a notch 50 (FIGURE 2) which permits displacement of crown wheel 47 whereby said ratchet wheel 41 can rotate freely of crown wheel 47 in a clockwise direction, relative to FIGURE 2, when said ratchet wheel is driven by the pinion 38. On the other hand, when stem 42 is actuated so as to turn crown wheel 47 counterclockwise, said crown wheel 47 is held engaged with ratchet wheel 41 whereby rotation of stem 42 is transmitted through elements 47, 41, and 38 to ratchet wheel 39.

In FIGURE 2 it will be noted that the axes of the staffs of barrels 5 and 6 and the axes of the staffs of runners 30 and 37-38 (runner 30 being along the central axis of the movement and runner 37-38 being the runner of the self-winding mechanism) are located at the corners of a substantially symmetrical diamond-shaped polygon, with the axes of runners 30 and 37-38 lying along an axis of the polygon which is perpendicular to and midway along an axis of the polygon which connects the rotative axes of barrels 5 and 6.

The importance of this arrangement is demonstrated in FIGURE 6 wherein the axes of barrels 5 and 6 are respectively denoted as 54 and 55 and the axis of runners 37-38 is denoted as 56, with the arrows 51, 52, and 53 indicating the rotative directions of the barrels and of the runner 38.

When winding weight 25 pivots about the axis of shaft 26, a winding torque is transmitted through the afore-described winding train to intermediate winding runner 36 which is coaxial with barrel 6, said runner 36 thereby being urged to turn in the direction of arrow 52 (FIGURE 6). This torque is, in turn, transmitted from runner 36 to wheel 37 and to pinion 38 which is interconnected with wheel 37 via a common stac. The laws of equilibrium of forces require that this torque be counterbalanced by an equal and opposite torque representing the sum of the resistance torques which are exerted upon pinion 38 by the two ratchet wheels 39 and 41 which are acted upon equally by each one of the power drive springs.

The direction of the tangential driving force exerted by runner 36 upon wheel 37 is indicated by arrow 57, the value of which will hereafter be denoted F_1 . The consequent rotative torque M imparted by F_1 upon runner 37-38 is given by the equation $M=F_1 \times R$ wherein R is the radius of wheel 37.

The resistance torque which counterbalances M is developed by the tangential resistance forces F_4 and F_6 imposed upon pinion 38 by the two ratchet wheels 39 and 41, respectively. In FIGURE 6 the resistance forces F_4 and F_6 have been drawn along radial lines extending from the rotative axis of runner 37-38 for the purpose of drawing a vector diagram. In any event, the lines F_4 and F_6 are each parallel to respective tangent lines which pass through the points of contact between pinion 38 and each of the ratchet wheels 39 and 41 and necessarily, therefore, lines F_4 and F_6 are each respectively perpendicular to straight lines extending between axis 56 and axis 54 and between axis 56 and axis 55.

Forces F_4 and F_6 are equal to each other in amplitude and conform to the equation

$$(F_4 + F_6) \times r = M = F_1 \times R$$

wherein r is the radius of pinion 38.

Force F_1 acts in the plane of wheel 37 while forces F_4 and F_6 act in the plane of pinion 38. The bearings which rotatively support runner 37-38 are subjected to bearing loads which are proportional to the magnitude of the respective tangential forces F_1 , F_4 , and F_6 . In the plane of wheel 37, the bearing load will be substantially equal to force F_1 while in the plane of pinion 38 the bearing load will be substantially equal to F_{res} which is the vectorial sum of F_4 and F_6 whereby

$$F_{res}=0.7F_4=0.7F_6$$

as is evident from the vectorial relationship shown in FIGURE 6.

The herein disclosed arrangement, therefore, reduces to a considerable extent the bearing load in the plane of the winding pinion 38 compared to what it would be if only one power spring, equivalent in strength to the two described herein, were employed. If only one equivalent spring were employed, the bearing load in the plane of the winding pinion 38 would be equal to F_4+F_6 or $2F_4$ or $2F_6$ (since $F_4=F_6$) as compared to $0.7F_4$ which results from the arrangement shown herein.

Even if only one spring were employed which was only equivalent to one of those referred to herein, the bearing load in the plane of pinion 38 would be equal to $1.0F_4$, this still being considerably higher than the value of $0.7F_4$ which the present invention provides even with twice the spring power.

A further important feature, however, derives directly from the fact that forces F_4 and F_6 act in one plane which is axially spaced away from the plane in which force F_1 acts. Since the respective forces act in different planes, they could establish a force couple between themselves tending to tilt the axis of runner 37-38. In other words, with reference to FIGURE 4, assuming that force F_1 had a component acting towards the right in the plane of wheel 37 and assuming that force F_4 had a component acting toward the left in the plane of pinion 38, this would establish a force couple tending to turn or tilt the entire runner 37-38 in a counterclockwise direction.

Relating the principle of the force couple to the instant situation, it is seen that if the total force acting in the plane of pinion 38 were relatively high, the tilting tendency upon runner 37-38 would be proportionally greater than if said force were relatively lower.

The bearings for runner 37-38 can, therefore, be considerably reduced in size for an arrangement in accordance with this invention relative to heretofore known power spring arrangements.

Furthermore, the thickness of bridges can be reduced as well as the overall dimensions of various bearing elements. The overall dimensions of the movement are correspondingly reduceable to acceptable standards notwithstanding the fact that the power spring means is capable of storing a relatively high reserve of power, namely, such as is required to obtain 36,000 oscillations per hour in the oscillating assembly. The high power reserve is especially important in self-winding watches which must allow for periods during which the wearer may perform few winding movement with its wrist.

With regard to the self-winding mechanism, the arrangement disclosed herein permits said mechanism to have a very high step-down ratio, this assuring effective winding even with very small wrist movements, notwithstanding the high value of torque provided by the two power springs, said torque being in the order of 1500 to 2000 g./mm.

FIGURE 7 is analogous to FIGURE 6 but is referred to the forces acting upon the center-wheel runner 7-7'. The ring gear on each barrel 5 and 6 engages pinion 7' and each ring gear exerts a tangential force F_{12} and F_{14} upon said pinion 7', with said barrels 5 and 6 turning in the directions of arrows 51 and 52. The torque imposed upon pinion 7' by forces F_{12} and F_{14} is counterbalanced by a resistance torque which results from a force exerted upon wheel 7 by pinion 8' of the third wheel runner 8-8'.

The tangential force which acts in the plane of wheel 7 can be neglected since it is quite low relative to the forces acting at the level of pinion 7' as a result of being reduced in the same ratio as that between the diameter of wheel 7 relative to pinion 7'.

The tangential forces acting in the plane of pinion 7', however, conform to the equation

$$F_{12}=F_{14}=\frac{1}{2}\frac{M}{r_7}$$

wherein r_7 is the radius of pinion 7'. By vectorial analysis, the resultant of $F_{12}+F_{14}=F_{res}=0.83F_{12}=0.83F_{14}$.

It is seen, therefore, that the reduced bearing load conditions previously described with reference to the runner 37-38 are also analogously applicable to center-wheel runner 7-7' whereby the bearings of this runner may be correspondingly smaller than would be required in conventional watch movements.

The details described herein with reference to a preferred embodiment are given by way of illustration and not for purposes of limitation.

What is claimed is:

1. A watch movement comprising a power spring means and a balance oscillator means, a time train extending from said spring means to said oscillator means, said spring means comprising two springs each housed in a separate barrel, each barrel being directly engaged with a first step-up runner of said time train, said first step-up runner being the center-wheel runner, a self-winding mechanism comprising a winding weight and a step-down winding train connecting said weight to each of two ratchet wheels which are respectively associated with said power means, the last step-down runner in said winding train including a winding pinion in direct engagement with both said ratchet wheels simultaneously.

2. The movement of claim 1, wherein said winding weight includes a staff which is coaxial with said center-wheel runner.

3. The movement of claim 1, wherein the vectorial resultant of the combined tangential forces exerted by said ratchet wheels upon said pinion of said last step-down runner is equal to 0.7 times either one of said forces, said tangential forces being equal to each other.

4. The movement of claim 1, wherein the rotative axes of said barrels lie at opposite ends of a first straight line and the rotative axis of said center-wheel runner and of said last step-down runner lie at opposite ends of a second straight line, said first and second straight lines perpendicularly bisecting each other.

5. The movement of claim 1, said barrels each being supported on respective barrel shafts, both said shafts being rotatively mounted in common bridges.

6. The movement of claim 1, including a manual winding means for winding said springs.

7. The movement of claim 1, wherein said winding train comprises three step-down runners.

8. The movement of claim 7, wherein the first of said step-down runners includes a wheel which is part of a reverser gear means, a pinion rigid with said weight and engaged with an intermediate wheel which engages said reverser gear means.

9. The movement of claim 8, wherein the second of said step-down runners includes a pinion which engages an intermediate wheel which is coaxial with one of said barrels and which in turn engages the wheel of the third step-down runner which is the last of said step-down runners.

10. A watch movement comprising a power spring means and a balance oscillator means, a time train extending from said spring means to said oscillator means, said spring means comprising two springs each housed in a separate barrel, each barrel being directly engaged with a first step-up runner of said time train, said first step-up runner being the center-wheel runner, said center-wheel runner including a pinion in direct engagement with both said barrels and a gear wheel in driving engagement with a third wheel runner which in turn is in driving engagement with a fourth wheel runner, said fourth wheel runner being coaxial with said center-wheel runner.

11. The movement of claim 10, wherein said center-wheel runner gear wheel extends in a plane below the lower face of the barrels and wherein said fourth wheel

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runner includes a gear wheel which extends in a plane above the upper face of said barrels.

12. The movement of claim 11, including an intermediate bridge extending radially between the barrels at a level between the upper and lower axial ends of said barrels, a bearing in said bridge rotatively supporting said fourth wheel runner coaxially with said center-wheel runner.

13. A watch movement comprising a power spring means and a balance oscillator means, a time train extending from said spring means to said oscillator means, said spring means comprising two springs each housed in a separate barrel, each barrel being directly engaged with a first step-up runner of said time train, said first step-up runner being the center-wheel runner, said barrels both being engaged with a pinion of said center-wheel runner in a common plane, the vectorial resultant of the combined tangential forces exerted by said barrels upon said center-wheel runner being equal to 0.83 times either one

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of said forces, said tangential forces being equal to each other.

14. The movement of claim 13, wherein said spring means drives said oscillator means at a frequency of at least 30,000 times per hour.

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U.S. Cl. X.R.

58—86

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,479,813 Dated November 25, 1969

Inventor(s) HANS KOCHER

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 4, line 36, "stac" should read -- staff --.
Column 5, line 51, "movement" should read -- movements --.

SIGNED AND
SEALED
JUN 23 1970

(SEAL)

Attest:

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Attesting Officer

WILLIAM E. SCHUYLER, JR.
Commissioner of Patents