The Anatomy Of

The Grasshopper

or

A closer look at the leaper:

being

an attempt to revive interest in a fine escapement

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(TECHNICAL)

Author's Note — This monograph grew, almost spontaneously, from efforts originally directed towards adding a bit of dynamic interest to a talk presented to some Rochester, New York, members of Chapter 13, NAWCC.

Almost exactly one year ago, Anthony Prasil, our Chapter president, asked me to prepare a short talk for the 10 November meeting of the Rochester group and tell how I managed to design and construct a 32-day weight-driven regulator movement with wooden wheels, plates, escape wheel, and pallets.

As there is a limit to show-and-tell personal chest-thumping beyond which no prudent speaker must venture (even before beloved and non-violent fellow NAWCC members), it seemed expedient to provide something for lagniappe.

As a consequence, a Harrison grasshopper escapement model with a seven-inch diameter escape wheel was fabricated from odds and bits of materials found during the week preceding the meeting night. The proportions of its construction were scaled directly from diagrams in books on hand. No effort at kinematic analysis was made at the time.

When the model was assembled the escapement immediately worked. Although it was out-of-beat and out of adjustment — it worked!

I am pleased to say that audience-reaction to the talk brought me no bodily harm. Nor was I called out of my name by anybody. True, some members were observed fondling their conglomerates of rubble, vegetables, fruit, and worse, but not one missile was launched. I believe the mention of the Harrisons, John's tenacity in designing the marine timekeepers, and the ultimate demonstration of the grasshopper escapement were responsible for the deferment of ballistics exercises. One member discarded his ammunition and stentoriously observed, "I see it but I don't believe it! Lemme get a closer look."

Indeed, his reaction was understandable, for the model was a poor thing. But the long, ponderous pallet-arms' leaping great distances from the escape wheel teeth to the unnecessarily remote spring buffer-stops was amazing to watch. At the conclusion of its leap, each pallet-arm would bounce

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off its stop, jiggle a bit, and become quiescent barely in time to be seized for the next impulse. The bouncing and jiggling were suggestive of a dainty house-cat's shaking its hind-legs to dislodge unwanted accumulations of cold dew-drops.

Anyhow, the model went back home to semi-retirement. Occasionally, over the succeeding months, I would let it perform, wonder at its faults, ponder its virtues, and even try to do a little constructive thinking on days when I could stand the pain.

The conclusions I have since drawn convince me the marvelously designed device frequently, and unjustly, has been maligned by people who should have known better.

I am chagrined to have been guilty of quoting them on one or two occasions. This paper is partial penance for my sin of gullibility.

It is to be hoped that horological students and artisans of greater stature than the author's may direct their more capable minds and hands towards dispelling the mists of misrepresentation surrounding the efforts of the Harrisons and kindred giants of the past. It is not a good thing to minimize the great legacy passed to us by these ancient and honorable people.

It is further hoped this modest effort may help to recreate favorable interest in constructing and experimenting with this splendidly flexible example of the art and science of escapement design.

To the Harrisons and their kind, wherever they now may be, this trifling work is dedicated.

Lang may their lums reek!

Rochester, New York
22 October, 1971

I GENERAL DISCUSSION AND SOME OBSERVATIONS

The Brothers Harrison, John and James, were both credited with the invention of a curious, antic mechanical clock-linkage which has been known through many years as "The Grasshopper Escapement." That the popular name was well-given is attested by all who have had the good fortune to see a specimen in action and to witness the crisp, crepitant, leaping motion of the pallet-arms' departure from the escape wheel teeth.

Since there appears to have been no conflict between the brothers as to which invented the escapement, it seems pointless and academic at this late date to speculate on its exact origin.

Although the escapement was used by John Harrison in three of his famous marine timekeepers and was used with outstanding accuracy in long-case timepieces by both brothers, the principle was virtually abandoned by subsequent clockmakers.

Possibly the neglect of this fascinating device was due to improper analysis and understanding of its functions and advantages.

Eminent writers have tended to diminish the popularity of this ingenious invention by quoting one another regarding grave faults said to be inherent in its design.

In the very few and sketchy accounts this writer has been able to research, so far, the general consensus of others is that the following faults are common in the basic principle of the grasshopper escapement:

A. Difficult to adjust
B. Excessive recoil necessary
C. Great pendulum amplitude necessary

On the basis of published schematic diagrams, the author constructed a large operating model of the grasshopper escapement driven by a weight barrel mounted immediately below (and geared to) the escape wheel arbor.

That the system was difficult to adjust proved, in practice, to be completely fallacious! Even in very reduced scale, the adjustment should not prove to be as tedious as that in most standard regulator mechanisms commonly in use. Anyone who can make, adjust, and put "in-beat" a classic Graham dead-beat escapement should find the grasshopper escapement easy to make and to adjust. And he might be surprised and amused to discover the escapement almost insists on performing well even when subjected to misguided tinkering!

Proper understanding of the escapement's geometry and functioning demands more study, however.

Further observation and experiments with the large scale working model indicated that two of the three faults mentioned above did, indeed, manifest themselves. It was readily apparent that a moderate amount of recoil was directly realized from the large pendulum arc and excesses of motive force. In addition, the escapement tripped if most of the motive force was removed while the pendulum was swinging.

That these faults were directly associated with unsound construction g e o m e t r y was not immediately recognized.

The geometry now has been refined, and it turns out that adherence to strictly correct principles can eliminate the previously published "faults."

No additions or second-guess gadgetry are necessary or desirable for designing a most satisfactory, efficient and elegantly simple escapement.

Given a sound design basis, original thinkers should properly be tempted to extract a legion of derivative mechanisms from these principles that promise immense flexibility. One or two very interesting variants have occurred to the author during the preparation of this work. Their investigation may stimulate further writings in the not-distant future.

II ACTION, PERFORMANCE, AND ADJUSTMENT

Description of the grasshopper escapement action requires the mention of "articulated" or jointed pallet-arms, "static" friction locking, "non-sliding" impulses, "spring-biased" pallet-arms, and other interesting phrases not often found in the nomenclature of other escapements. Figures 1 through 6 show the sequence of events in the action of a geometrically sound version of the grasshopper escapement.

In Figure 1, the shaded tooth on the clockwise rotating escape wheel of fifteen teeth* is giving impulse to pallet (A). The connecting pallet-arm (AH), pivoted at (H), communicates the impulse to the crank-arm (GH) and causes it to move in a clockwise arc about a center (G) which is also the pivot-point for the crutch (or pendulum) of a timepiece. The pendulum is depicted by the heavy broken line (GK) at the mid-point of its swing to the left and momentarily coincident with the line of centers (GO). Crank-arms (GH) and (GJ) are rigidly connected to the crutch arbor and move through the same angular arc.

Meanwhile, the pallet (B) and another shaded escape wheel tooth are approaching each other and will lock together when the escape wheel has rotated one quarter of a tooth pitch

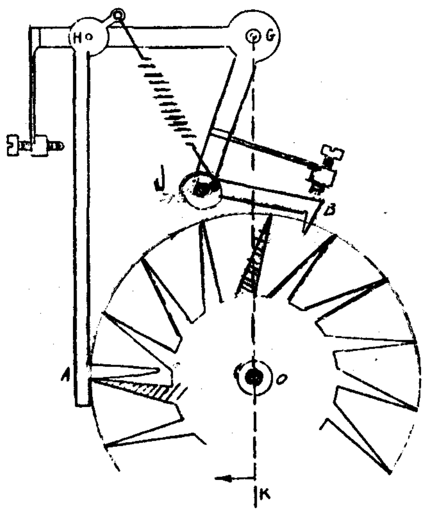


Fig. 1 The escape wheel tooth is engaging the entry-pallet (A) and is impulsing the pendulum (K) to the left. The exit-pallet (B) is seeking engagement with its nearest approaching tooth

clockwise from the instantaneous position shown in Figure 1.

Figure 2 shows the pallet (B) locked to its adjacent shaded tooth. The clockwise drive of the escape wheel is now arrested. However, the pendulum, having momentum, continues its swing to the left and the crank-arm (GJ) continues to pull the pallet (B) and the locked tooth to the left and forces the escape wheel to recoil a small amount, i.e., about five to ten minutes of angle. As a consequence of this recoil, the escape wheel tooth, associated until now with the pallet (A), backs away (counter-clockwise from (A).

Being relieved of the escape wheel tooth load, the pallet-arm (AH) is immediately urged against a resilient buffer-stop (M) by a low-force biasing spring (L). (M) and (P) are fixed to and move with the crank-arms (GH) and (GJ). Figure 3 shows the pallet-arm (AH) at rest against the buffer-stop (M). The tip of the pallet-nib of (A) must clear the circumference of the escape wheel sufficiently to allow the shaded tooth adjacent to pallet (A) to pass under the pallet-nib when the pendulum starts its swing to the right.

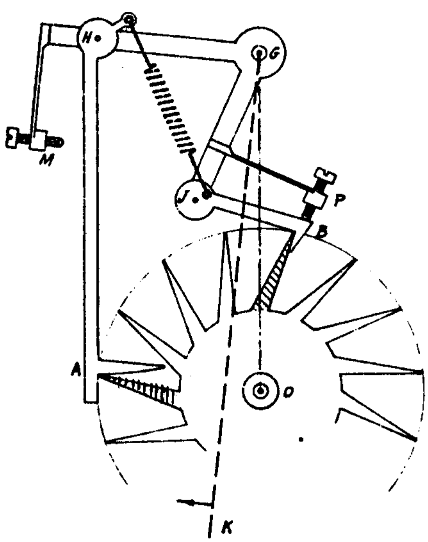


Fig. 2 The exit-pallet has engaged its tooth. The pendulum momentum to the left is about to cause escape wheel recoil and the release of the entry-pallet

Figure 4 shows the pallet (B) being impulsed to the right by its adjacent shaded escape wheel tooth. The escape wheel has rotated one-quarter of a tooth pitch clockwise, the pendulum (GK) is again momentarily coincident with the line of centers (GO) as it reaches the mid-point of its swing to the right, and the pallet (A) has cleared its previous impulsing tooth (shaded). The pallet-arm (BJ) has been drawn away from its buffer-stop (P) against the mild resistance of the bias spring (L). As the escape wheel continues to impulse the pendulum to the right, the pallet (A) dips behind its previous impulsing tooth and moves to meet the following tooth as both seek a locking condition.

In Figure 5, the escape wheel has rotated clockwise another one-quarter tooth pitch. The pallet (A) has locked on its appropriate tooth and clockwise rotation of the escape wheel has again ceased. Meanwhile, the pendulum's momentum to the right is about to force the escape wheel again to recoil slightly.

Figure 6 shows how, immediately after the escape wheel recoil, pallet (B) jumps to a rest position against

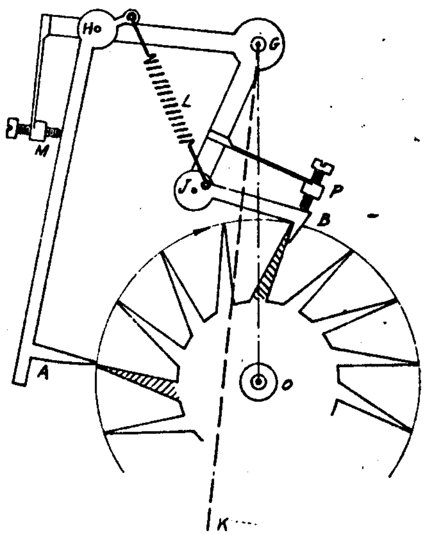


Fig. 3 Recoil has taken place, the entry-pallet has cleared its wheel tooth, and the pendulum is about to be impulsed to the right by the wheel tooth engaging the exit-pallet

the buffer-stop (P) as urged by the force of the bias spring (L). The tip of the pallet-nib of (B) must clear the circumference of the escape wheel sufficiently to allow the shaded tooth adjacent to pallet (B) to pass under the pallet-nib when the pendulum starts its swing to the left.

The configuration of Figure 1 again presents itself, and the cycle is repeated.

Careful study of the action cycle will make it apparent that, unlike the time-honored Graham dead-beat and other classic escapements, the Harrisons' grasshopper escapement possesses near-zero sliding friction. A very trifling increment of sliding friction must, of a necessity, exist at the instant a pallet unlocks from a wheel tooth at recoil and the pallet bias spring overcomes the resistance of the rapidly diminishing static friction-hold of the tooth on the pallet.

The pallets should not be lubricated. They do not require the choice of any special material in combination with the escape wheel teeth as regards low-friction characteristics.

Since energy put into the pallet-arm bias springs is lost energy, the springs should have just a trifle more strength than that necessary to assure raising the pallet-arms against the buffer-stops. In past years, some experimenters have counterweighted the pallet-arms to achieve this same effect with near-constant force. This substitution for springs has merit and may well be easier to adjust and control in applications where space permits the proper installation of correct poise-arms.

The buffer-stops must be resilient enough to yield as the pendulum continues to swing by its momentum after the locking of the escape wheel. In any event the buffer-stops merely need to be stiff enough to positively and accurately locate the pallet-arms as they are alternately moved away from the escape wheel by the bias springs.

Following impulse and locking, the supplementary arc of the pendulum is modulated by the energy absorbing offices of the buffer-stops and the recoiling gear train of the timepiece. Except for gear train friction losses and internal molecular friction of the buffer-stop material, most of the ab-

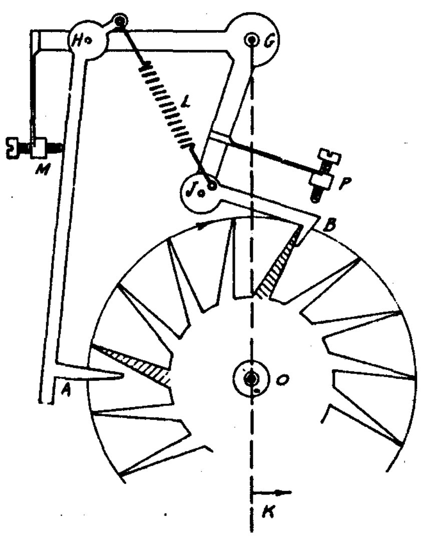


Fig. 4 The engaged escape wheel tooth/exit-pallet is impulsing the pendulum to the right. The entry-pallet is seeking engagement with its nearest approaching tooth

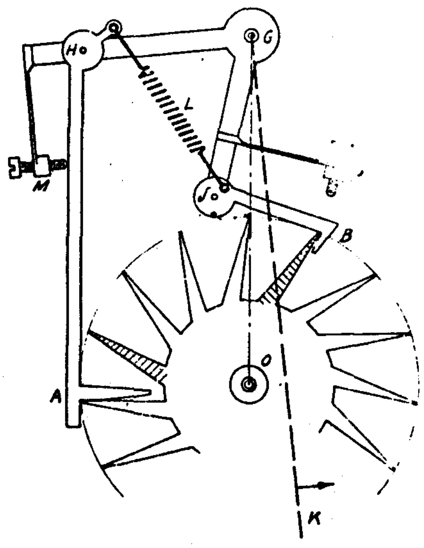


Fig. 5 The entry-pallet has engaged its tooth. The pendulum momentum to the right is about to cause escape wheel recoil and the release of the exit-pallet

sorbed energy is put back into the pendulum during its return.

The motive force in the timepiece should be adjustable to assure reliable unlocking of the pallets at the extremities of the pendulum's swing. Great recoil is not required for the proper performance of the grasshopper escapement. With a well made gear train, the motive force required to drive the train and escapement will be relatively small, because the grasshopper escapement is efficient.

Impulse is continually given during the clockwise rotation of the escape wheel, friction is almost non-existent at the pallet/tooth interfaces, and there are no "shakes" or "drop" losses.

It is probable that some degree of isochronism could be realized through proper balance of the motive force, the pendulum bob weight, and the stiffness of the buffer-stops.

The corner formed by the intersection of a pallet's pressure face and a plane passing through the pallet-arm's pivot axis should capture the escape wheel tooth-point as precisely as can be managed. By providing the buffer-stops with fine-pitch adjusting screws, this condition easily can be met. Adjustment of a buffer-stop position causes a pallet to meet its next engaging wheel tooth earlier or later as necessary to secure this ideal cornering (Figure 7). This adjustment can moderately be mismanaged and the escapement will continue to function well. But good cornering is readily realized; careful craftsmanship dictates minimizing all possible variables.

The pendulum should not be beat-adjusted in this fashion of moving one buffer-stop and then the other. A separate beat-adjustment is much to be preferred.

The long pallet-nibs serve only as a safety function. They prevent accidental tripping of the escapement and the "running through" of the train. Classic diagrams fail to show the necessity of providing this safeguard. If the nibs are too short it is possible for the gear train to rapidly run down while the pendulum is

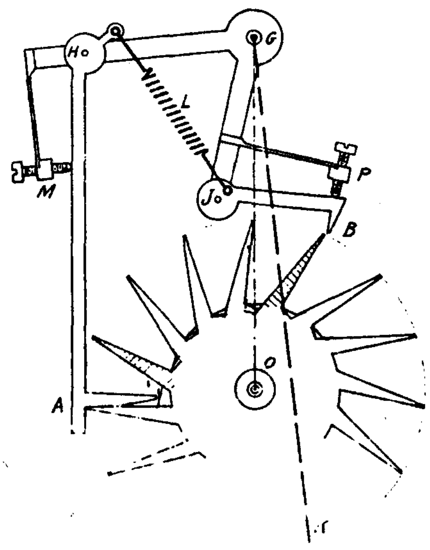


Fig. 6 Recoil has taken place, the exit-pallet has cleared its wheel tooth, and the pendulum is about to be impulsed to the left by the wheel tooth engaging the entry-pallet. The next step in the cycle duplicates Fig. 1

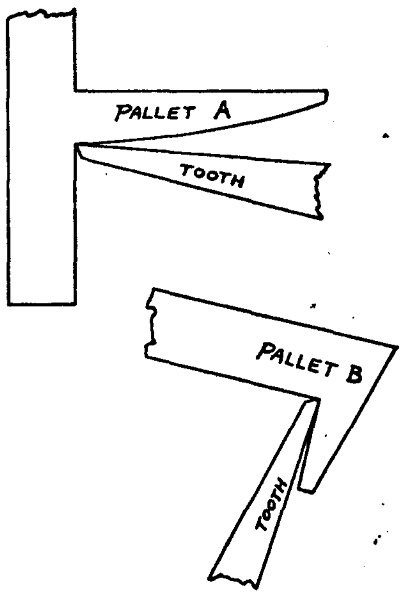


Fig. 7 Enlarged views of the two pallets show how the escape wheel teeth "corner" in the respective pallet angles at the instant of engagement

swinging and immediately after the driving force is momentarily removed from the train. This condition might well prevail when winding a timepiece devoid of a maintaining-work device.**

The point-to-point distance between the pallet-nibs must be less than the length of the chord which crosses the escape wheel circumference at the points where the pallets act. The pallet-nibs must not be excessively long, however. Sufficient tip clearance must be provided to pass appropriate escape wheel teeth at the proper portion of the pendulum's swing. These two requirements are not incompatible if correct construction is adhered to.

The evident unsymmetry of the pallet-nib lengths is explained by the fact that both pallet-arms operate through equal angular arcs,*** but are unequal in overall length from pallet to pivot. Therefore, the longer arm subtends a longer action-chord at the pallet end and the nib has to be longer in proportion.

The shape of the entry-pallet's pressure-face is a simple cylinder arc struck from the pallet-arm pivot center.

The shape of the exit-pallet's pressure-face is a plane set perpendicular to a plane containing the pallet-arm pivot axis.

The escape wheel's tooth profile works very well if it is well-pointed and the acting face is undercut ten to twelve degrees from a radial line drawn to the tooth tip. As in many Graham wheels, the tooth tips can be left "flat" with one-half degree of the circumference left untouched. Care must be taken to see that the gullets between the teeth are cut deep enough to accept the longer pallet-nibs at full-depth locking.

III PENDULUM AMPLITUDE CONSIDERATIONS

The choice of the amplitude of the pendulum arc is easily realized and accomplished by manipulating design geometry. The arc length varies directly as the radius of the escape wheel and inversely as the number of escape wheel teeth and the effective distance between the pallet-arm's line of action and the crutch-pivot axis. The pallet-arm's distance is equal to the geometrical span of the pallet crank-arm, in most cases.

The following little formula is useful for planning the pendulum arc length:

$$\theta = \frac{180\ R_1}{Z\ R_2}$$

where:

θ = the pendulum arc in degrees
R_1 = the escape wheel radius
R_2 = the distance of the pallet-arm's line of action from the crutch pivot axis
Z = the number of teeth in the escape wheel

To have an example of this simple computation, suppose we take the now familiar diagrams, Figures 1 through 6, and resolve their mutual basic geometry into a basic linkage diagram as shown in Figure 8:

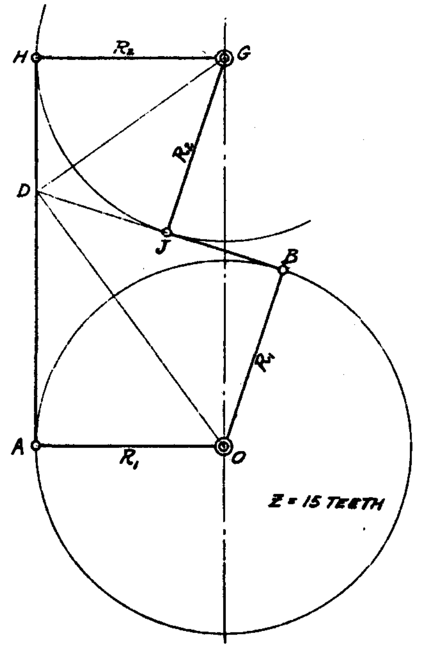


Fig. 8 This is the basic linkage geometry of the diagrams as shown in Figs. 1 through 6

let:

$R_1 = 1$
$R_2 = 1$
$Z = 15$ teeth

then;

$$\theta = \frac{180 \times 1}{15 \times 1} = 12°$$

Now, twelve degrees is a fairly large arc, as is often mentioned in the writings of others who discuss the grasshopper escapement.

If we try another escape wheel with thirty teeth and keep the other values the same as before, we will find the arc reduced to six degrees. Six degrees of arc are better, but the arc remains about twice the amplitude associated with good regulators.

From here on, if desired, obvious steps can be taken to diminish the arc by resorting to the legitimate expedients of increasing the radius (R_2), the escape wheel tooth count (Z), or both at once relative to the escape wheel radius (R_1):

let:

$R_1 = 1$, $R_2 = 2$, $Z = 60$ teeth

then;

$\theta = 1.5°$! . . . enough said.

IV THE CORRECT GEOMETRY FOR THE GRASSHOPPER ESCAPEMENT

The whole secret in getting the right start in laying out the grasshopper escapement is the understanding of a "mutual tangents" concept, or, insuring that the pallet-arms lay on tangents common to the escape wheel circle and the crank circle on which the pallet-arms are pivoted.

Referring back to Figure 8, note that pallet-arms (AH) and (BJ) not only coincide with the tangents, but also their lengths are determined by the distance between the points of tangency on the respective circle circumferences.

The linkage motion, as carried on over small arcs, is analogous to two pulley wheels coupled together one-half of the time by a common loop belt so as to rotate together in the same sense. The other half of the time they are coupled together with a crossed belt and they rotate in an opposite sense with respect to each other.

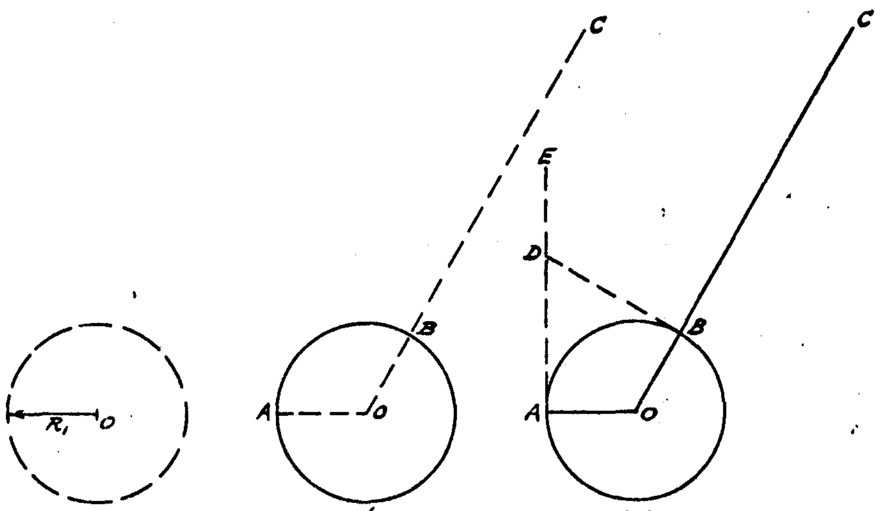


Fig. 9 (a) through (e) illustrate the sequence of the dimensionless schematic layout of the grasshopper escapement. The identification lettering is consistent with all previous (and subsequent) diagrams

Again, from Figure 8, it can be seen that if center distances are altered, the points of tangency must shift. This, in turn, means that the number of tooth pitches spanned by the pallets is affected and suggests that some logical procedure is necessary for imposing desired characteristics into the system.

V THE LAYOUT PROCEDURE

The procedure for laying-out follows for a general-case escapement. Specific dimensional values are not considered, as they are at the discretion of the escapement designer. For convenience in layout, the pallet-arm segments are all represented in Figure 9 (a through e), as in Figure 8, i.e., in the locked mid-position, or tangent to the escape wheel circle. In practice, each would be spring-biased away from the wheel circumference and against the buffer-stops. The bias angle equals one-half the pendulum arc.

The layout steps in Figure 9 are as described in keys (a through e), in sequence:

a. Select a convenient radius (R_1) and draw the escape wheel circle about a center (O).

b. Lay out a sector whose angle (AOB) equals the subtense of (N + .5) tooth pitches. Extend one arm (OBC) of the sector.

c. Construct tangents (AD) and (BD) to the sector radii (OA) and (OB). Extend tangent (AD) to a convenient point (E).

d. Bisect the angle (BDE), letting the bisector originate at (D) and terminate at the intersection point (F) on (OBC). For reference, observe that angles (AOB) and (BDE) are equal. Let the halves of bisected angle (BDE) be identified by the symbol (*α*) alpha.

e. Any point (G) between (D) and (F), on line (DF), may be used as the pallet-crank center. Using a convenient radius (R_2), centered at (G), draw arc (HIJ) tangent at points (H) and (J) on the arms (BD) and (DE) of the bisected angle. Draw (GH) and (GJ).

This concludes the layout and includes the complete structural frame-

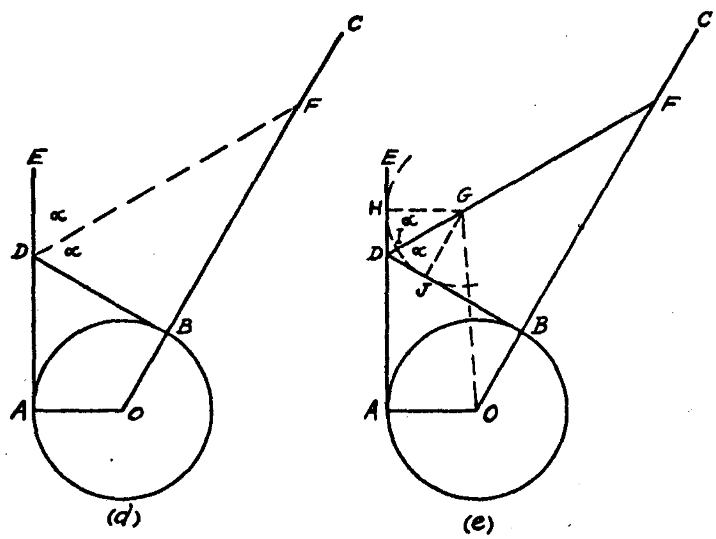


Fig. 9 continued

work on which a correct escapement can be based. Figure 10 illustrates a few of the many configurations, all correct, that can be created by this layout method. In some of the cases the pallet-arms are shown pivoted at a common point (D) on a single crank-arm. This is a valid construction method**** and is especially useful in cases where the exit pallet-arms become unmanageably short as in cases where center (G) closely approaches (F) in the layout diagram. In either construction, for small motions, the effective line of action of the pallet-arms is equivalent to a line tangent to radius (R_2). It is important that this be kept in mind when making pendulum arc calculations.

VI FORMULAS FOR THE LINKAGE DIMENSIONS (Use Figures 8 and 9e)

From the construction diagrams, and from their content of many similar (or identical) right triangles, the following dimension-finding formulas can be deduced:

A. Pallet span angle =

$$AOB = \frac{360 \times (N + .5)}{Z}$$

where:

$(N + .5)$ = a whole number of teeth plus one-half tooth

Z = total number of teeth in the escape wheel

B. Half span angle $\alpha = \frac{AOB}{2}$

Note: the angle α is the key to all the following formulas:

C. Working dimension DO = R_1 secant α

(DO) dimension is required

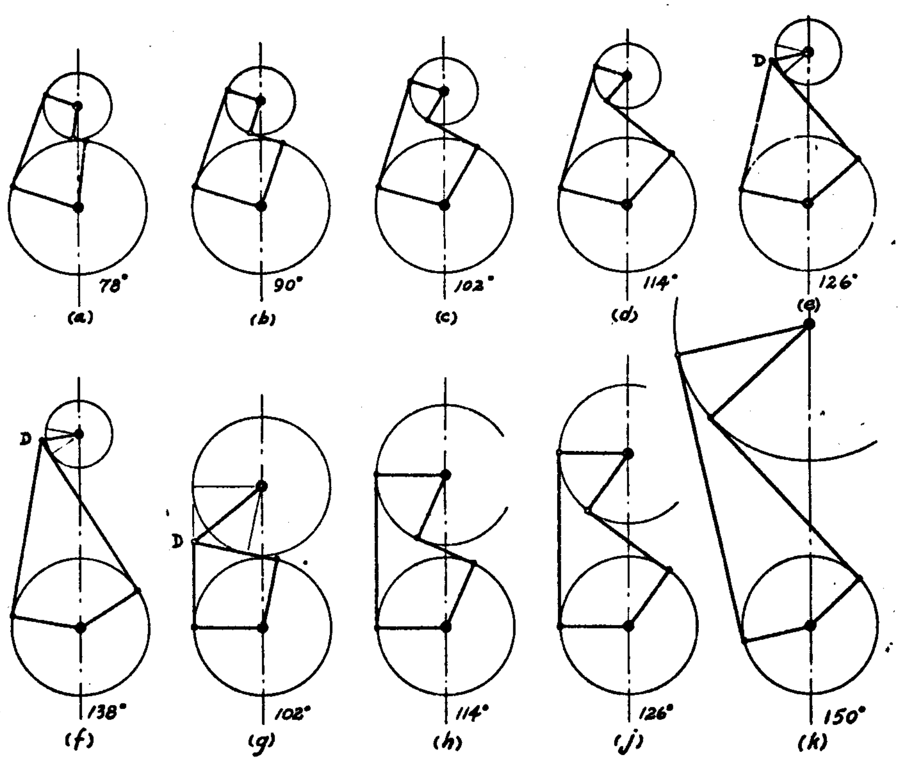


Fig. 10 (a) through (k) illustrate a number of possible linkage configurations. The lower circle represents an escape wheel of constant size through the entire group. Note that three sub-figures (e, f, & g) depict the pallet-arms pivoted on common centers for finding the center distance (GO)

D. Working dimension AD = BD = R_1 tangent α
(AD or BD) dimension is required for finding pallet-arm lengths.

E. Limit dimension BF = R_1 tangent^2 α
(BF) is the dimension which should not be exceeded by (R_2)

F. Working dimension DG = R_2 cosecant α
(DG) dimension is required for finding the center distance (GO)

G. Working dimension DJ = DH = R_2 cotangent α
(DJ or DH) dimension is required for finding pallet-arm lengths

H. Center distance GO = $\sqrt{DO^2 + DG^2}$

I. Exit pallet-arm length BJ = BD − DJ

J. Entry pallet-arm length AH = AD + DH

K. Exit pallet-nib length BR = $\frac{BJ\pi\theta}{180}$

L. Entry pallet-nib length AQ = $\frac{AH\pi\theta}{180}$

M. Special case:
If the pallet-arms are pivoted on a common center (D), then (AD and BD) become the working lengths of two identical length pallet-arms. However, the pallet - nib lengths (BR and AQ) are still calculated from (BJ and AH) as shown above.

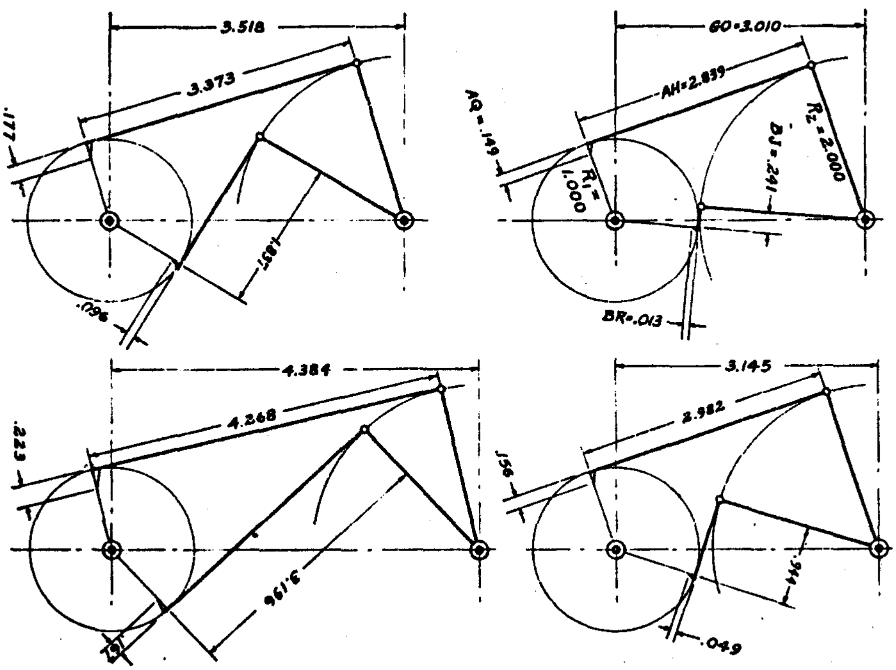


Fig. 11 (a) through (d) show dimensioned diagrams of four possible linkages. All are designed for a 3° pendulum arc. The tooth subtenses on a thirty tooth wheel are: (a) 9½, (b) 10½, (c) 11½, and (d) 12½ teeth. All dimensions can be proportionally varied to suit specific space and duty requirements. See text and Table B for additional information

VII TABLE OF CONSTANTS FOR COMPUTING DIMENSIONS OF GRASSHOPPER ESCAPEMENTS CONTAINING 30 OR 60 TEETH

Table A (end of article) has been compiled to provide constants that can be multiplied by the appropriate radii to secure correct dimensions for designing grasshopper escapements with thirty or sixty tooth escape wheels. Those persons possessing trigonometry tables can use the formulas above for special cases, taking note that the reciprocal functions can be used if their tables do not contain columns of secants and cosecants.

Those persons lacking trigonometry tables (or the inclination to exercise rusty mathematics memories) can still fare extremely well by laying out a large-scale drawing of the special case and very carefully measuring the results and dividing them by the drawing scale factor.

VIII DESIGNING AN ESCAPEMENT: AN EXAMPLE

By utilizing the formulas and table-constants already supplied, a sample escapement can be designed as follows:

given:

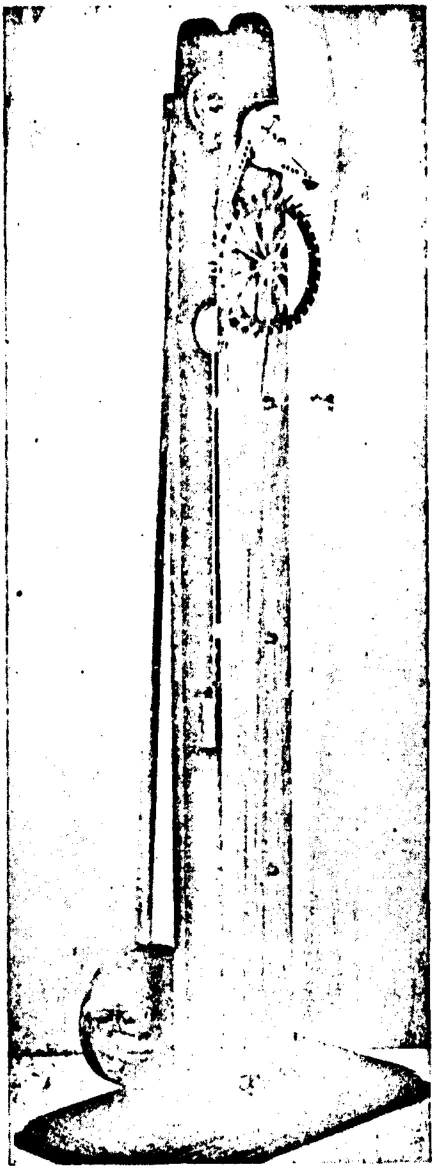
2.000 inch dia. escape wheel with 30 teeth, ($R_1 = 1$); ($Z = 30$)
3° pendulum arc, ($\theta = 3°$)
9½ teeth pallet span,
($N + .5$) = 9½

find:

Span angle, (AOB)
Pallet-arm crank radius, (R_2)
Center distance, (GO)
Exit pallet-arm length, (BJ)
Entry pallet-arm length, (AH)
Exit pallet-nib length, (BR)
Entry pallet-nib length, (AQ)

solution:

In Table A, column (1), find ($N + .5$) equals 9½. To the right of 9½, in column (3) find (AOB) equals 114°.



A. This is the original grasshopper escapement model as it was contrived for a Chapter 13 NAWCC meeting. The stand is made of Honduras mahogany; an attractive and dimensionally stable wood well-suited for large model-structures. The pendulum bob is a fine old piece of apple-wood. Note the dogwood gearing between the pallet-arbor and the pendulum-arbor; this prevented the pendulum's swinging through an uncomfortably great arc

In section III, our pendulum arc formula was: $\theta = \frac{180 R_1}{Z R_2}$

or:

$$R_2 = \frac{180 R_1}{Z \theta} = \frac{180 \times 1}{30 \times 3} = 2.000$$

In Table A, following the same line to the right, find under (5); (DO) equals 1.8361. Since $R_1 = 1.000$, or unity, we let (DO) remain equal to 1.8361 in conformance to the table footnotes.

Likewise, under (8), find (DG) = 1.1924. Since $R_2 = 2.000$, the table notes tell us to multiply the (DG) value found under (8) by R_2. Therefore, our new, or working (DG) equals 1.1924 x 2 = 2.3848.

From our formulas, section VI, we find:

$$(GO) = \sqrt{DO^2 + DG^2}$$

$$(GO) = \sqrt{1.8361 \times 1.8361 + 2.3848 \times 2.3848}$$

(GO) = 3.010 inches

From the table,

(BD) = (AD) = 1.540

and

(DJ) = (DH) = 2 x .6494 = 1.299

Then

(BJ) = 1.540 − 1.299 = 0.241 in.,

and

(AH) = 1.540 + 1.299 = 2.839 in.,

and

$$(BR) = \frac{.241 \times 3.1416 \times 3}{180} = .0126 \text{ in.}$$

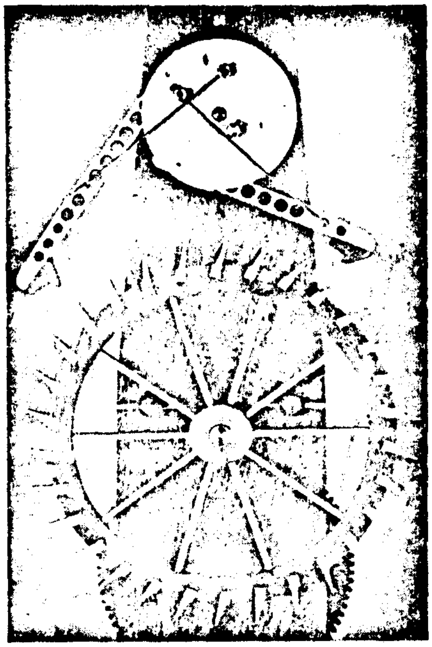
and, finally,

$$(AQ) = \frac{2.839 \times 3.1416 \times 3}{180} = .1486 \text{ in.}$$

Here, it should be noted that the value of $R_2 = 2.000$ is very close to the table value (BF) = 2.3716, the limiting value for R_2. It can be seen from previous diagrams and from direct numerical results, as in the present case, that as R_2 approaches the value of (BF), the exit pallet-arm (BJ) and its pallet-nib length (BR) become shorter; becoming non-existent when R_2 exactly equals (BF).

In the case just completed it is evident that the pallet-arm (BJ) is almost impractically short for convenient fabrication. By resorting to special case (M), in section VI, we can opt for placing both pallet-arms on common center (D), making the arms (AD) = (BD) = 1.540 inches. The pallet-nib lengths previously found must keep their values unchanged, however.

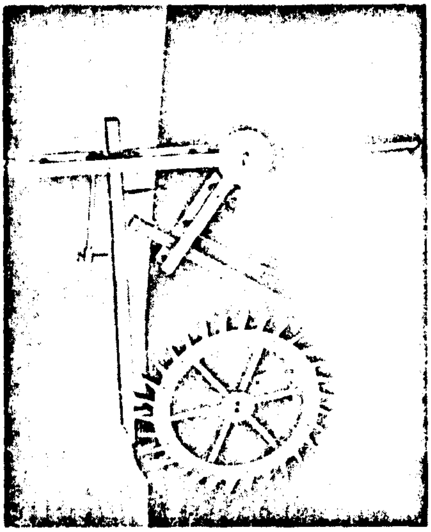
Exit pallet-arm-and-nib lengths could be increased by adding one or more teeth to the pallet-span (N + .5), and keeping the other given values the same as before. This will increase the center distance (GO).



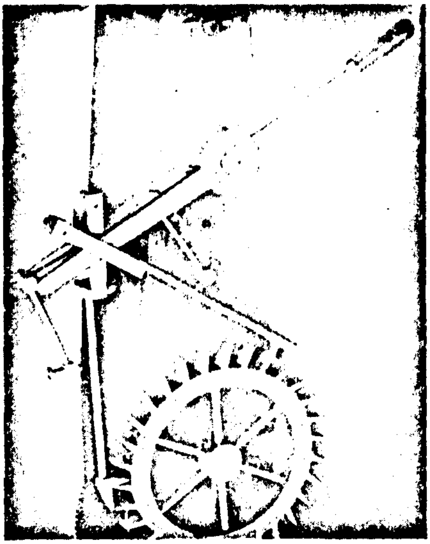
B. Here is a detail view of the original model. The escape wheel is made up of mahogany segments keyed together and containing inserted dogwood teeth, birch spokes, and a brass-bushed dogwood boss. The wheel teeth were indexed and finished to size after final assembly of the wheel components. The buffer-stops are straight lengths of music wire. The wooden pallet-arms are tuned for proper tooth-to-pallet engagement by bending the buffer-stop wires as required. The pallet-arms are cherry-wood and the pallets are made of African bubinga. The bias-springs are tiny coils of fine music wire pocketed in holes hidden within the dogwood crank-disc. Note, that with no loading to maintain tooth engagement, the pallet-nibs stand well clear of the tooth-tips and therefore unlock the escape wheel. The text explains how to prevent this

The pallet-nib lengths, as found by formula, are maximum lengths. Both nibs should be shortened enough to pass the escape wheel teeth after impulse. About ten per cent shorter is usually adequate for proper tooth clearance. The critical length of a pallet-nib is fifty per cent of the formula length.

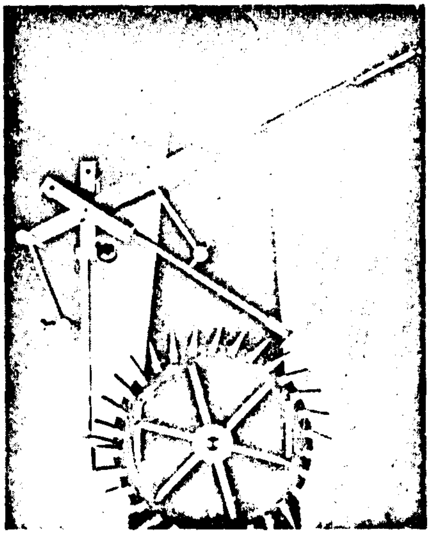
Table B is a short table of dimensions for a family of escapements based on parameters established in our sample calculation. Only the (N + .5) value is varied; therefore,



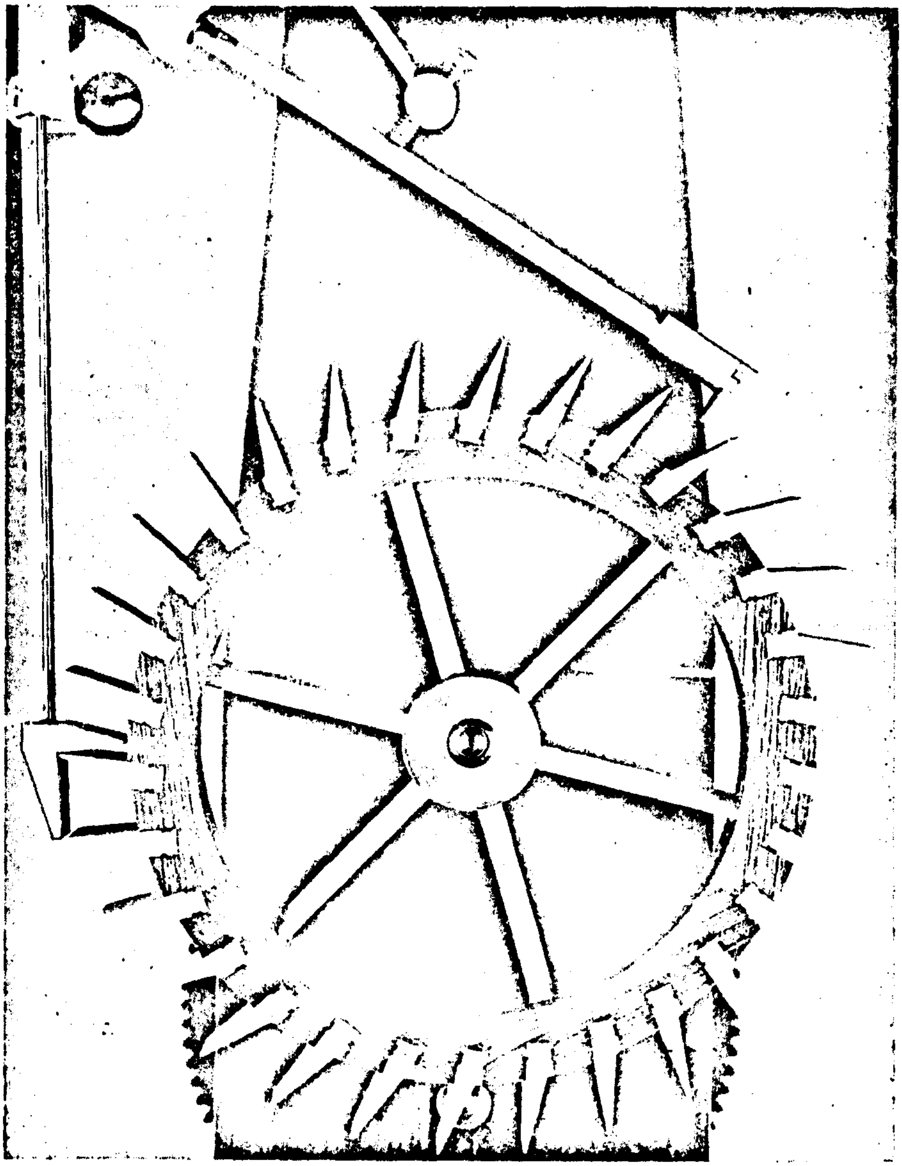
C. The escape wheel in this configuration is six inches in diameter; i.e., one inch smaller than on the original lecture model. The pallet/crank complex is properly proportioned to provide a pendulum arc of six degrees. No gearing is needed. The pallet-arbor is now common with the axis of the pendulum arbor. A sixty-tooth escape wheel would have decreased the active pendulum arc to three degrees if substituted for the thirty-tooth wheel shown. Gravity forces have here been substituted for spring forces mentioned in the text. The teeth and spokes of the wheel are made of hophornbeam wood. The pallet/linkage complex consists of mahogany, dogwood, stainless steel, and brass. The capsule-shaped object to the right is a brass poise-weight for the total assembly. The components are individually poised or loaded as required for correct function. Unloading this escapement will not unlock the wheel teeth to cause tripping



D. Here is another variation of the pallet/linkage assembly. The performance of the entire assembly duplicates that of the one shown in photograph (C). The pallet-arms share a common pivot-point as described in the text. This configuration will not voluntarily trip, either. Both of these latter constructions possess sufficient strength, accuracy, and durability to perform very well in a small tower clock



E. This is a straight front elevation view of the common-pivot version. Photographer's note: mahogany can be turned into ebony if you use blue-sensitive (color-blind) film



F. The tight close-up view here shows the wheel detail in large scale. Note the inserted hophornbeam teeth in the mahogany rim. Compare the entry-pallet/tooth engagement (left) with the diagram in Fig. 7

the effects of pallet-span changes are apparent.

In Figure 11 (a through d) are dimensioned diagrams of all family members in Table B.

Of course, the family members are, as yet, skeletons; we flesh the structure as we see fit.

TABLE B

Vital Dimensions For Four Possible Grasshopper Escapements

$R_1 = 1.000$ $R_2 = 2.000$

$Z = 30$ $\theta = 3°$

(N + .5)	9½	10½	11½	12½
AOB	114°	126°	138°	150°
AD = BD	1.540	1.963	2.605	3.372
GO	3.010	3.145	3.518	4.384
BJ	.241	.944	1.837	3.196
AH	2.839	2.982	3.373	4.268
BR	.013	.049	.096	.167
AQ	.149	.156	.177	.223

Legend:

θ = Pendulum arc in degrees

R_1 = Radius of the escape wheel

Z = Tooth count of the escape wheel

R_2 = Effective radius of pallet-arm crank

(N + .5) = Tooth pitches spanned by pallets

AOB = Angle subtended by (N + .5) tooth pitches

AD = BD = Pallet - arm lengths if they are pivoted about common center (D)

GO = Center distance between the escape wheel and the crutch-pivot (or pendulum suspension point)

BJ = Exit pallet-arm length

AH = Entry pallet-arm length

BR = Exit pallet-nib length

AQ = Entry pallet-nib length

HGJ = Theoretical pallet crank-arm angle and can be obtained by subtracting angle (AOB) from 180°; i.e., (HGJ) = 180° − (AOB).

FOOTNOTES TO MANUSCRIPT PAGES

*(page 253) This coarse-pitched wheel was deliberately chosen to permit the showing of large motion amplitudes for better pictorial clarity.

**(page 257) Of all things created by the Harrisons, perhaps the grid-iron compensating pendulum and their very excellent maintaining-work device are the most used today.

***(page 257) These arcs, subject to construction and pivoting exigencies, are theoretically exactly equal to the pendulum arc. They are automatically obtained by the buffer-stop adjustment required for the proper meeting of the teeth to the pallet corners.

****(page 260) It can be demonstrated that each pallet-arm may be pivoted at other points along its tangent line (line of action); i.e., either pallet can be designed to be pushed, or pulled, as desired.

Author's Retronote — Subsequent to completing this paper (and prior to making additional working-model escapements for photographs and for proving practically the validity of theories, formulas, and tables in the text above) Colonel Humphrey Quill's very well-written article, "The Grasshopper Escapement," in the September, 1971 issue of "Antiquarian Horology" (see Reference 11), was brought to my attention.

Compared to the thin fare I had foraged from other sources, Colonel Quill's article was a veritable feast of Harrison lore and grasshopper entomology! I, therefore, delayed the completion of this modest effort until I could obtain as much as possible of his relevant published material. It is with pleasure that I have included additional items (12) and (13) to the list of references.

Other references in the list, although excellent works, mostly do better at delineating the Harrisons' story rather than presenting comprehensive grasshopper escapement material useful to the technical student or clockmaker.

Colonel Quill's fine article came perilously close to pre-empting the need for this article. The geometrical common-tangents concept was well presented and the accompanying illustrations throughout the article are superb.

However, the technical scope of Colonel Quill's article was somewhat short of my desires and the circulation of its publication is relatively small in this country; therefore, I *believe the material in this paper can be utilized to good ends by horological students and the cellar-scientists of the United States.*

Colonel Quill's able researches suggest the apparent probability that no original, complete Harrison grasshopper escapement is now in existence; and, further, that many repairs and replacements of pallets and pallet-arms have occurred over nearly two and one-half centuries' span of time. Who is to say that dimensions of the pallet-arms and pallet-nibs were not altered to the detriment of sound geometry and prudent practice?

Photographs and drawings of Benjamin Vulliamy's escapements suggest it is fair to speculate he might have made correctly proportioned assemblies that did, in fact, satisfy good practice. And maybe he even knew how to prevent his escapements from unlocking and tripping . . . . perhaps our British friends could research this more fully.

I hope others will come forth to shed more light on the grasshopper story; if they do so, I hope I can be where I can watch them glow!

Rochester, New York
27 December, 1971

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See next two pages for Table A

TABLE A

Constants for Computing Dimensions of Grasshopper Escapements Containing 30 or 60 Teeth

Pallet Span = (N + .5)	Teeth	Span Angle = (AOB)	α	DO	AD, BD	BF	DG	DJ, DH
					× R1		× R2	
† (1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
	12½	75°	37.5°	1.2604	0.7673	00.5887	1.6425	1.3032
6½		78	39.0	1.2868	0.8098	00.6558	1.5890	1.2349
	13½	81	40.5	1.3151	0.8541	00.7295	1.5399	1.1708
	14½	87	43.5	1.3786	0.9490	00.9006	1.4526	1.0538
7½		90	45.0	1.4142	1.0000	01.0000	1.4142	1.0000
	15½	93	46.5	1.4526	1.0538	01.1105	1.3785	0.9490
	16½	99	49.5	1.5396	1.1708	01.3708	1.3151	0.8541
8½		102	51.0	1.5890	1.2349	01.5250	1.2868	0.8098
	17½	105	52.5	1.6426	1.3032	01.6978	1.2604	0.7673
	18½	111	55.5	1.7655	1.4550	02.1170	1.2134	0.6873
9½		114	57.0	1.8361	1.5399	02.3716	1.1924	0.6494
	19½	117	58.5	1.9139	1.6319	02.6634	1.1729	0.6128
	20½	123	61.5	2.0956	1.8418	03.3929	1.1379	0.5430

TABLE A (Continued)

Pallet Span = (N + .5)	Teeth	Span Angle = (AOB)	α	DO	AD, BD	BF	DG	DJ, DH
10½		126	63.0	2.2027	1.9626	03.8533	1.1223	0.5095
	21½	129	64.5	2.3229	2.0965	04.3932	1.1079	0.4770
	22½	135	67.5	2.6130	2.4142	05.8274	1.0824	0.4142
11½		138	69.0	2.7904	2.6051	06.7860	1.0711	0.3839
	23½	141	70.5	2.9958	2.8239	07.9750	1.0609	0.3541
	24½	147	73.5	3.5211	3.3759	11.3974	1.0430	0.2962
12½		150	75.0	3.8637	3.7321	13.9278	1.0353	0.2680

†The following notes, regarding use of this table, are keyed to the numbers of the columns above. The column headings adjacent to the key numbers are coded to agree with the alphabetic identities found in the previously shown diagrams.

(1) Enter the table in this column if the escape wheel has 30 teeth.

(2) Enter the table in this column if the escape wheel has 60 teeth.

(3) Read here the angular subtense of the chosen pitch value in (1) or (2).

(4) This is one-half of (3), or α.

(5), (6), (7) If R_1, the escape wheel radius, equals unity, use the table values. If R_1 is some value other than unity, multiply table values by R_1.

(8), (9) If R_2, the pallet-arm crank radius, equals unity, use the table values. If R_2 is some value other than unity, multiply table values by R_2.