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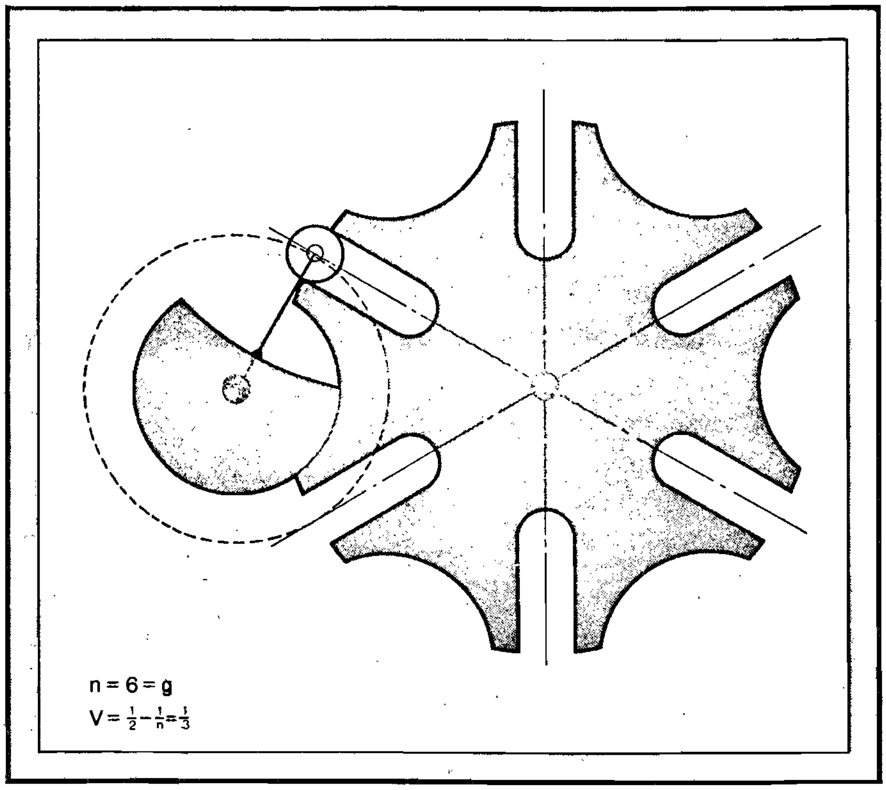


Design of a mechanism to give intermittent motion often relies on the conventional Geneva wheel. Here we describe the gear driven version of this mechanism, which gives a smoother transmission of power and permits input and output shafts to be coaxial

There are many ways to drive Geneva wheels. Normally, this is done by a single crank bearing a pin with a roller that intermittently interlocks with the Geneva wheel. The centre of the pin then traces a circle that touches the centre-lines of two successive slots that are engraved in the wheel. However, a mechanism that generates intermittent motion in this fashion shows a number of disadvantages:

- Input and output axes are not co-axial.
- Output angles of 360° or 180° are not possible.
- The ratio of times V, which is the ratio of times between the motion period of the wheel and the cyclic period which is the time-lapse for motion and dwell together, is dependent on the number of stations n. It is therefore not possible to choose V and n independently from one another. This limits the mechanisms' usefulness to the designer. For an *external* Geneva wheel, for instance, the ratio V equals the value represented by the equation:

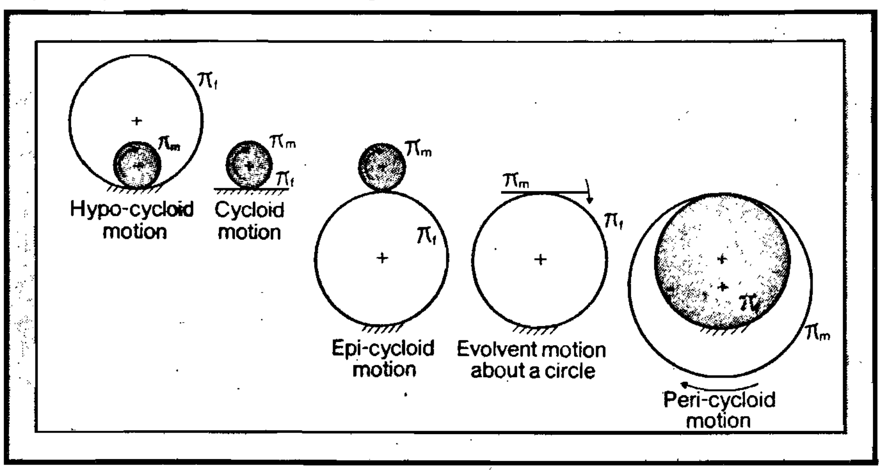
$$V_{ex} = \frac{1}{2} - \frac{1}{n}$$

For an internal Geneva wheel the ratio is represented by

$$V_{in} = \frac{1}{2} + \frac{1}{n}$$

- An angular jerk appears at the start and at the end of the motion of the

1 (top): *External Geneva wheel, driven by a single crank.*
2 (below): *Cycloidal curves of a gear-driven mechanism*



by Dr. E A. Dijksman, Liverpool Polytechnic

wheel, caused by the regular circle motion of the pin.

When the driving part of the mechanism is replaced by a gear-wheel mechanism, however, the input and output axes may be made coaxial. The dependency of V and n may be relieved and the starting jerk reduced.

The curve traced by the driving pin is then a cycloidal curve instead of a circle. The pin that produces the curve is attached to a gear-wheel that rolls about another, fixed wheel. The fixed wheel has a pitch circle which is the *fixed polode* (π_f) of the motion. Similarly, the moving wheel is comprised by the *moving polode* (π_m).

If the moving polode rolls inside the fixed polode, a *hypocycloid* motion results. If it rolls over and about the fixed polode an *epicycloid* motion occurs. Finally, if the moving polode embraces the fixed one, a so-called *pericycloid* motion is generated. All these possibilities are shown in Figure 2.

A point attached to the moving polode in these cases will trace a hypocycloid, an epicycloid or a pericycloid respectively: if the centre of the pin lies inside the moving polode, the point traces a contracted cycloid; if outside, a *protracted* cycloid. There are always two gear-wheel pairs that will produce the same curve, and the two gear-wheel mechanisms that are linked in this way are *curve-cognates* of one another.

As pointed out earlier, the input and output axes of the mechanism may be made coaxial, which means that the input-crank M_oM and the Geneva wheel must rotate about the same fixed centre M_o. So, in order to obtain smooth output motion—without a jump in the angular velocity of the wheel—the driving pin of the wheel may only enter or leave a slot if the tangent to the curve is directed on the centre M_o. This fact dictates some dimensions of the mechanism. To investigate this relationship, let's consider the *design position* of the mechanism, which is the position where the driving pin just enters or just leaves the Geneva wheel. Such design positions are shown in Figures 4 and 5.

As in the design position the tangent to the curve at C is directed on the centre M_o, the point C of such a position has to join a locus that resembles the circle with diameter M_oP, point P being the velocity pole of π_m with respect to π_f in the design position. Generally, therefore, the mechanism considered allows for two degrees of freedom in design—the choice of the point C on the mentioned locus, and the chosen ratio R/R_o of the polodes. Thus, in comparison to the Geneva wheel that is driven by a single crank, we have obtained an additional degree of freedom in design. This gives us in turn the freedom to choose the ratio of times V_o and the number of stations n independently from one another.

Designing practical mechanisms

We now define γ as the angle enclosing two wheel diameters touching a lobe of the curve on both sides. We further confine ourselves to angles γ for which

$$\gamma = \frac{2\pi}{n} \qquad \ldots\ldots\ldots(1)$$

where n is a positive integer ≥ 4.

In addition, we define the angle α as the angle needed for the input-axis to turn from the position in which the

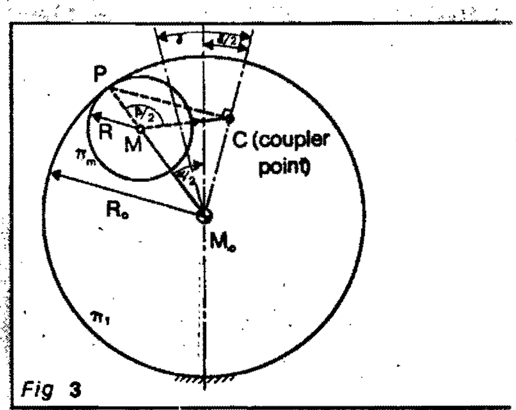
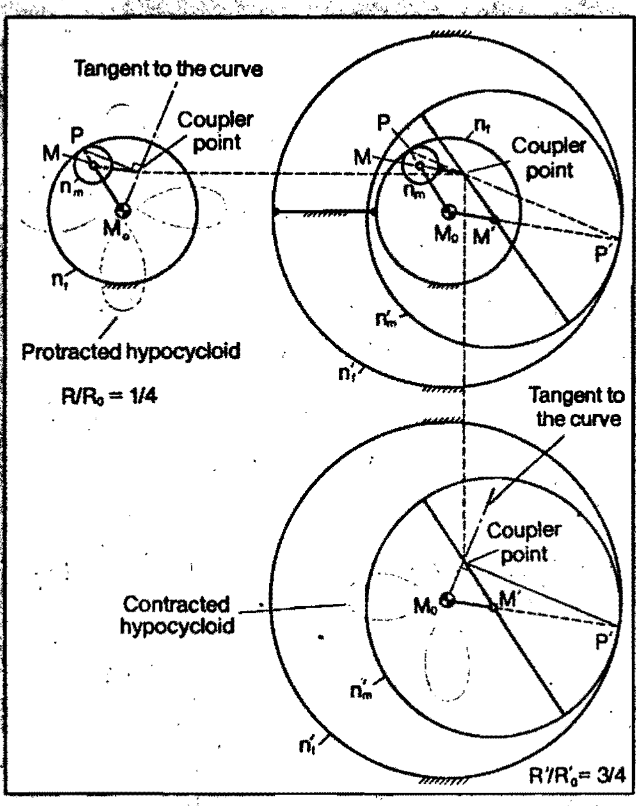
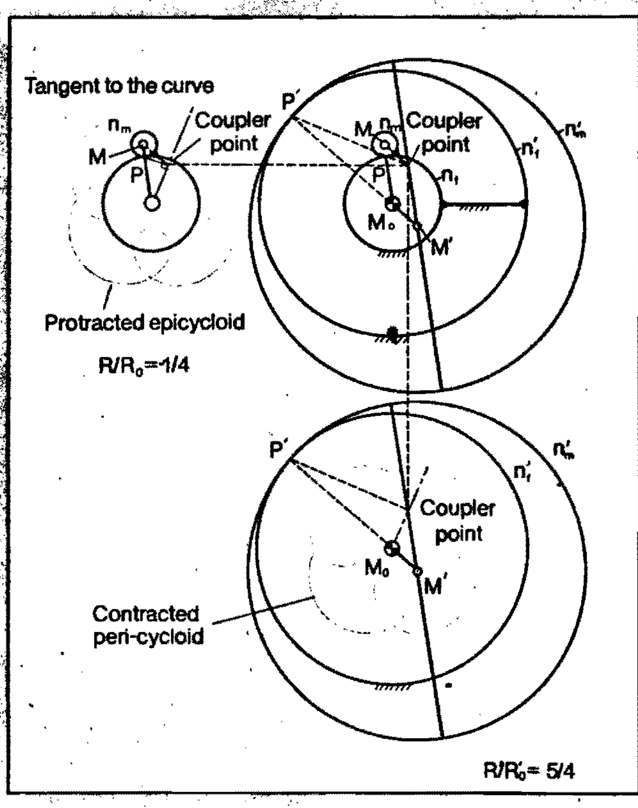


Fig 3

4: Identical hypocycloids traced by cognate gear wheel mechanisms



5: Epicycloid and pericycloid curves, similarly, are identical in each case



$\frac{\alpha+\gamma}{2} \leq \pi/2$

driving pin just enters the wheel to the position in which the pin is at the point of leaving the wheel.

For the hypocycloid motion this angle α may be calculated as follows: Since in the design-position, at C, the curve-tangent joins M_o, the tangent M_oC must be perpendicular to the path normal PC. Referring to Figure 5,

$$M_oC = R_o \cos \tfrac{1}{2}(\alpha+\gamma) \quad \ldots\ldots(2)$$

According to the rule of sines for ΔM_oMC, we additionally have

$$MC \sin \tfrac{1}{2}\beta = M_oC.\sin \tfrac{1}{2}(\alpha+\gamma) \quad \ldots(3)$$

in which $\tfrac{1}{2}\beta.R = \tfrac{1}{2}\alpha.R_o$

Thus $\beta = k\alpha \quad \ldots\ldots(4)$

Further: $R_o - R = M_oM$

$$= M_oC \cos \tfrac{1}{2}(\alpha+\gamma) - MC \cos \tfrac{1}{2}\beta \quad \ldots(5)$$

From equations 2, 3 and 4, we derive

$$\frac{MC}{R_o} = \frac{\sin(\alpha+\gamma)}{2 \sin \tfrac{1}{2}k\alpha} \quad \ldots\ldots(6)$$

Similarly, if we combine equations 2, 4 and 5 we find

$$\frac{MC}{R_o} = \frac{k^{-1} - \sin^2 \tfrac{1}{2}(\alpha+\gamma)}{\cos \tfrac{1}{2}k\alpha} \quad \ldots\ldots(7)$$

Thus, equating the right-hand sides of the last two equations, we get

$$\sin(\alpha+\gamma) = 2[k^{-1} - \sin^2 \tfrac{1}{2}(\alpha+\gamma)] \tan \tfrac{1}{2}k\alpha,$$

whence we find,

$$(2k^{-1}-1)\sin \tfrac{1}{2}\alpha k - \sin[\alpha(1-\tfrac{1}{2}k)+\gamma] = 0 \quad (8)$$

This formula is derived only for the hypocycloid motion for which $k>2$. The derived formula determines the value α if the gear-ratio k and the number of stations n are known. However, since in the design position the coupler point C joins the circle with diameter PM_o, only those values for α are permissible for which $\alpha+\gamma/2 \leq \pi/2$. Thus any value for α that is derived through eq.(8) has to meet the condition $\alpha \leq \pi - \gamma$. In case the derived α values do not meet this condition, no mechanism corresponds to the given gear ratio and given number of stations.

Equation 8 remains unchanged if we transform the driving mechanism into its curve-cognate, and hence must be valid also for the pericycloid-driven Geneva wheel, as well as for epicycloid and hypocycloid mechanisms.

Dimensions of each pair of curved cognates are related through the equations.

$$\frac{R'}{R'_o} = 1 - \left(\frac{R}{R'_o}\right)$$

$$k = \frac{R}{R_o} \quad k' = \frac{R'}{R'_o}$$

$$\frac{M'C}{MC} = \left(\frac{k}{k'}\right)^2 \frac{R}{R'}$$

$$\frac{R'}{MC} = \frac{M'C}{R} = \frac{k}{k'}$$

$$\frac{M_o'C}{R_o'} = \frac{M_oC}{R_o} \quad \frac{R}{MC} \text{ and } M_o' \equiv M_o$$

The ratio of times

According to its definition the ratio of times V_o answers the equation

$$V_o = \beta/2\pi$$

whence, according to equation 4

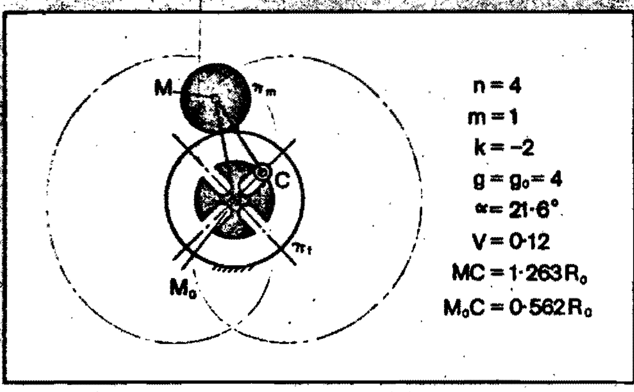
$$V_o = k\alpha/2\pi \quad \ldots\ldots(9)$$

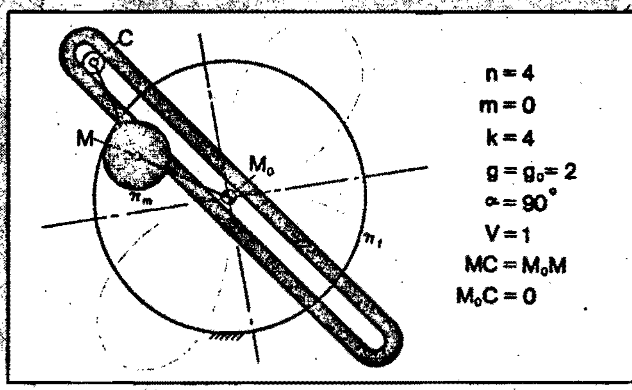
Apart from the sign, the two curve-cognates always spend the same time in motion in relation to the time needed for the full cycle.

Eliminating α from equations 8 and 9 then gives rise to the equation

$$(2k^{-1}-1)\sin \pi V_o - \sin \pi V_o(2k^{-1}-1) + \frac{2\pi}{n} = 0 \quad \ldots\ldots(10)$$

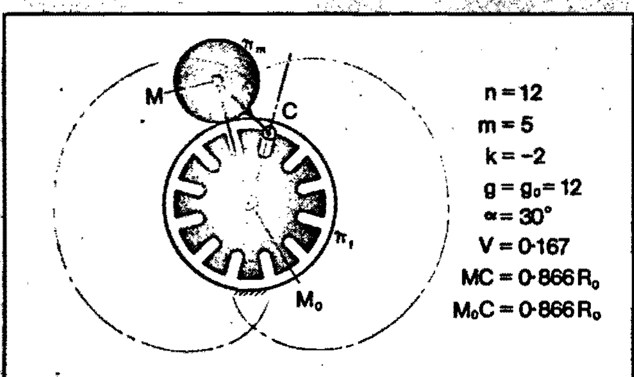
which, providing equation 9 holds, is still valid for all values of $k=R_o/R$.

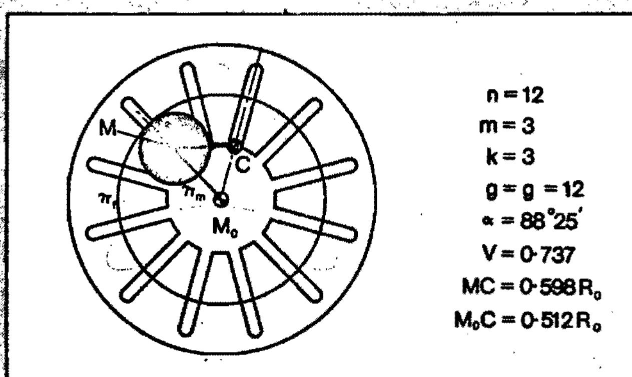


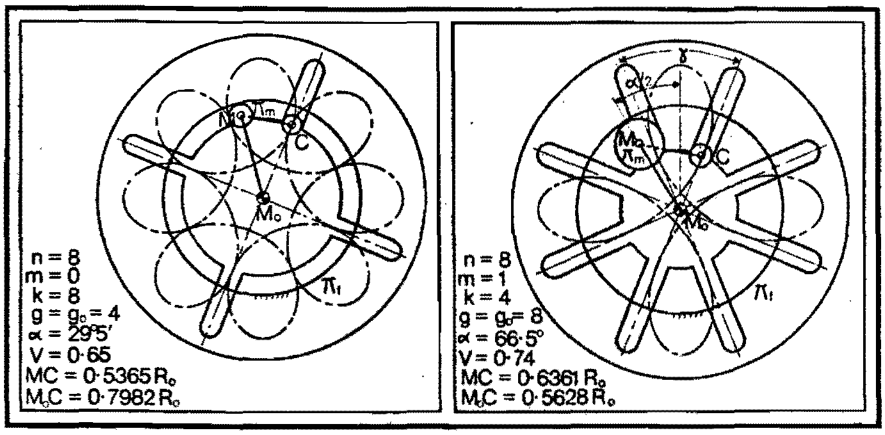


6 (above) and 7 (below): *Epicycloidal driven Geneva wheels and their parameters*

8 (above) and 9 (below): *Hypocycloidal mechanisms; slot in 8 gives instantaneous dwell*







10, 11: *Two hypocycloid gear-driven Geneva mechanisms*

This equation 10 shows that unlike the single crank-driven Geneva wheel, the ratio V_0 is not dependent on the number of stations alone, but may be varied instead by choosing other values for k. For the designer, this is very practical. He must remember, though, that not all the real values of k are allowed. They are restricted to *rational* numbers only. In order to give the reader more insight in this respect, we shall define a new number m, equal to the number of lobes that could be placed between two successive lobes of the curve. Thus, m equals the number of unreal lobes that just fit between two successive, real ones that actually appear in the curve.

If m is a rational positive number, instead of a positive integer, such as m_1/m_2, we say that m_1 unreal lobes just fit between the first and the (m_2+1)th lobes that are really there. So, we define m as the maximum possible number of lobes that would fit for any number of cycles (minus the actual number of lobes that appear for that number of cycles) divided by the actual number of lobes that appear for those cycles.

Therefore, the actual number of lobes S that appear in the curve is given by

$$S = \frac{n_1 n}{m+1} \quad \ldots\ldots\ldots\ldots\ldots (11)$$

For the protracted hypocycloid (k > 2) we find that

$$\frac{1}{k} = \frac{n_1}{s} = \frac{m+1}{n} \quad \ldots (12a)$$

For the contracted hypocycloid (1 < k′ < 2) we then have

$$\frac{1}{k'} = 1 - \frac{m+1}{n} \quad \ldots\ldots\ldots (12b)$$

Similarly, we find for the epicycloid (k < 0) the relationship

$$\frac{1}{k} = -\frac{m+1}{n} \quad \ldots\ldots\ldots (12c)$$

And for the pericycloid (0 < k′ < 1) we have

$$\frac{1}{k'} = 1 + \frac{m+1}{n} \quad \ldots\ldots\ldots (12d)$$

These equations agree with the fact that either

$$2\pi R = \pm(m+1)\gamma R_0 \ldots\ldots\ldots\ldots (13a)$$

$$\text{or } 2\pi R' = [2\pi-(m+1)\gamma]R_0' \ldots (13b)$$

So, if we choose the values *m* and *n*, in addition to the kind of curve we are going to apply, the gear ratio is fixed. This can be done using whichever equation 12 corresponds to our choice of curve or mechanism.

In practice, designers will be confined to the protracted hypocycloid and the epicycloid driven Geneva wheel mechanisms. If necessary, we can always apply the cognate transformation and use the curve-cognates instead of the ones mentioned. The dimensions for the curve-cognate mechanisms then are easily derived from the source mechanisms through the cognate transition formulas already given. So, for briefness' sake we shall only refer to equation 12 if it is written in the form

$$k = \pm\frac{n}{m+1} \quad \ldots\ldots\ldots (12e)$$

If we substitute this value into equation 10 we arrive at the relation

$$\left(\pm 2\frac{m+1}{n}-1\right)\sin \pi V_0 - \sin\left[\pi V_0\left(\pm 2\frac{m+1}{n}-1\right)+\frac{2\pi}{n}\right] = 0 \quad (14)$$

For each integer n and rational number m it is then possible to calculate the ratio V_0. From the resulting graphs we may choose the practical values V_0 and n and then determine the number m from which we calculate the gear ratio, using equation 12. We may then determine the values for α and γ, according to the equations 9 and 1 respectively.

The remaining dimensions, such as M_0C/R_0 and MC/R, are finally calculated through the relations

$$\frac{M_0C}{R_0} = \cos\tfrac{1}{2}(\alpha+\gamma) \ldots (k<0 \text{ or } k>2)$$

and

$$\frac{MC}{R} = \frac{k}{2}\,\frac{\sin(\alpha+\gamma)}{\sin\frac{1}{2}k\alpha}, \ldots$$

$$(k<0 \text{ or } k>2) \quad \ldots\ldots\ldots\ldots\ldots (15)$$

If the lobes that appear in the curve are all used to drive the wheel, $V = V_0 = k\alpha/2\pi$. But even if we use them all, the designer of this kind of intermittent motion mechanism is still left with a large number of values V that are equal to or less than one, and examples are illustrated in Figures 6 to 9. In each case the number of slots or grooves g_0 that have to be made in the wheel does not necessarily have to be identical to the number of stations n of the mechanism.

Clearly, the number of slots needed, equals either n, n/2, n/3, n/4, . . . or 1. Which number it actually is, is decided by the fact that as the driving pin leaves a slot, it enters the next one $(m+2)\gamma$ or $(m+2)\,2\pi/n$ radians further on the wheel. So on the wheel (for 2π radians) there are at least n (m+2) slots. If n/(m+2) is a positive integer, g = n/(m+2).

If it is not, we have to multiply it with the smallest possible positive integer so as to make it one.

Thus

$$g_0 = n/(\text{greatest common divisor of n and } m+2) \quad \ldots\ldots\ldots (16)$$

In order to reduce the value of V, we may diminish the number of slots. The lower values for V obtained in this way are sometimes very practical, since they represent the circumstances in which a relatively small portion of time is needed for the actual motion of the wheel. Naturally, if there are fewer slots, the locking (stationary) time of the wheel will be greater, and more time thus available for completion of products that are moving around with the wheel.

How to find the number of slots, in those cases, is now explained:

If the driving pin leaves a slot, it may find the next one $(m+2)\,2\pi/n$ radians further on the wheel. However, if no slot is available at that position, it may find another one $(m+1)\,2\pi/n$ radians further on, and so on.

Therefore, the slots that are used subsequently are either $(m+2)\,2\pi/n$ rad, $(2m+3)\,2\pi/n$ rad, or $(3m+4)\,2\pi/n$ rad apart on the wheel. As before, we find that the number of slots g in the wheel has to meet the equation:

$g = n/g_{cd}(n,m+2)$ or $g = n/g_{cd}(n,2m+3)$ or $g = n/g_{cd}(n,3m+4)$ etc, . . . where g_{cd} resembles the greatest common divisor of two positive integers, one of them being n, which is the number of stations. Which integer the other one has to be, depends on the number of unused lobes in the mechanism. For example, if 3 lobes are unused $g = n/g_{cd}(n,4m+5)$.