Radial Valve Gears Again

PART 12 – WALSCHAERTS VALVE GEAR

Duncan Webster sheds light on what is often seen as a complex subject.

Continued from p.567
M.E.4728 October 20

Introduction

Valve gears described in the preceding articles have all derived their outphase component via a vibrating lever driven vertically up and down a slide. The three gears that follow differ in that a return crank at 90 degrees to the main crank gives the outphase component, whose magnitude is then determined by some other mechanism before being added to the inphase component.

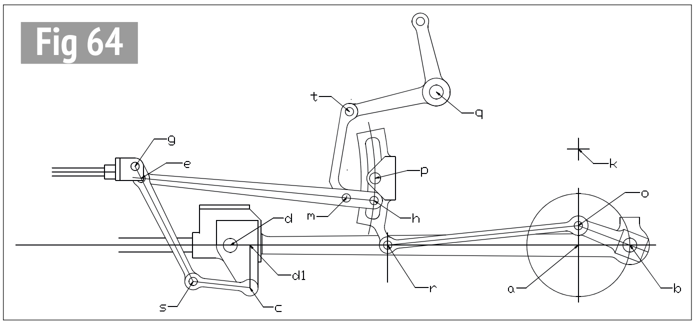
The first to be described is Walschaerts, probably the most widely used of all locomotive valve gears, both in full size and model engineering. It was invented by Belgian engineer Egide Walschaerts, the earliest patent being dated 1844, but this was not quite the gear as we know it today. It is equally suited to inside cylinders, outside cylinders, or even inside cylinders with the valve gear outside (later batch of LNWR Prince of Wales class).

Basic principles

The overall layout is shown in **fig 63** (inside admission) and **fig 64** (outside admission).

The outphase motion is driven by the return crank,

Fig 64



Walschaerts valve gear – outside admission.

which is displaced 90 degrees (not always exactly, see later) from the main crank, and its magnitude is altered by moving the radius rod up and down the curved slot in the expansion link. The inphase motion is derived directly from the crosshead via the anchor link, and so does not suffer from the connecting rod angularity distortion mentioned in part 5 (M.E.4722, July 28). The combination lever *es* adds the inphase and outphase components together. Simple so far isn't it?

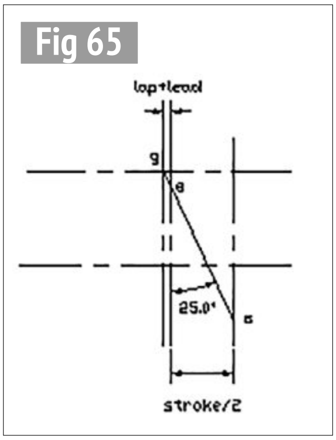
Two methods of suspending the radius rod are shown. In fig 64, the radius rod is hung from the lifting arm *qt* by rod *tm*. This arrangement was used on BR Standard locos amongst others. Some care is needed to get clearance between this rod and the expansion link. I got it wrong initially on one of my locomotives, and have had to make *tm* shaped like a tuning fork so that the expansion link can go into the slot. You can't win them all! Figure 63 shows the arrangement used by the GWR and LNER. Here, the radius rod is extended beyond the expansion link and has a second slot, in which runs a die block pivoted on *qt*.

Design of the combination lever, union link etc.

When I originally investigated this gear, I derived a lot of equations for the lengths of the combination lever, union link etc. These were of course accurate but the advent of CAD systems has made them unnecessary as you get the same answers to better accuracy than you can make them by drawing. Outside admission will be described and reference to fig 63 will show how to adapt this for inside admission.

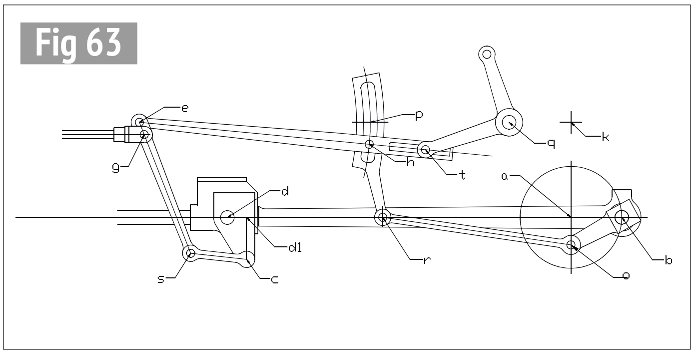
First, draw two horizontal lines (**fig 65**), one on the cylinder centre line and the other on the valve spindle

Fig 65



Combination lever geometry (a).

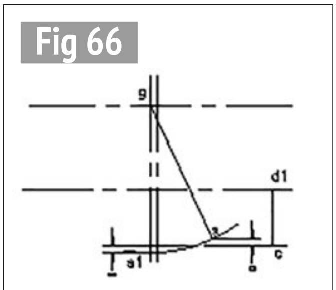
Fig 63



Walschaerts valve gear – inside admission.

centre line. Then draw a vertical line through what will be point *g* and two more verticals, one offset to the right by *lap + lead*, the other offset to the right by half the stroke (*ab*). You now need to decide how long to make the combination lever. Don Ashton recommended a maximum of 25 degrees and I have no reason to disagree with this. Other references (Yoder & Wharen) recommend a maximum of 20 degrees but this will lead to a longer combination lever, which can be a nuisance. Starting at point *g*, draw a line at 25 degrees to the vertical to intersect with the half stroke line at *s* as shown. Point *e* is the connection of the combination lever to the radius rod *eh*. All nice and simple so far. Make sure that points *g* and *e* are far enough apart to be able to accommodate rod ends and pins. Reference to published designs will help here. If they are too close then reduce the 25 degrees.

Now you have to position the combination lever with respect to the cylinder and crosshead, bearing in mind that at front centre the bottom end will come close to the cylinder cover and it needs to clear any nuts etc. at the little end. I can't help you here! Having done this draw the little end pin onto the sketch. In most cases point *s* will be well below the little end pin and so a drop arm *d1c* is provided. It is usual to have *c* directly below *d*. I see no need for this - in fact it reduces union link angularity to have the drop arm behind the little end pin as shown in figs 63 and 64. It seems reasonable that the anchor link should oscillate equally above and below horizontal as the combination lever swings to and fro.



Combination lever geometry (b).

Moving on now to **fig 66**, draw an arc centred on *g* through s to get point *s1*, and draw a horizontal line such that the vertical offsets from points *s* and *s1* are equal. Where this line crosses the vertical through *d1* gives point *c* and you can measure the length of the union link *sc*.

It can happen that the drop arm is uncomfortably long. To reduce this, you can offset the connection at point *g* above the valve spindle centre line and just shift the combination lever and union link up a bit. This means the drive to the valve is offset, so I'd only do it if the spindle is well guided.

Design of expansion link

On to the design of the expansion link itself. We now have to use sums, sorry! The position of the expansion link pivot should be chosen where convenient - about half way between *e* and the axle works, but it is not critical. Ideally it should be level with point *e*, but again this isn't critical - the BR standard 9F had this point well below its ideal. If *p* is below *b*, then at front and back centres a tangent to the curved slot will be normal to a line from *p* to *e*, and the gear is said to be inclined.

For outside admission slide valves, the outphase motion driven by the expansion link is magnified by the combination lever in the ratio

$$\frac{es+eg}{es}$$

For inside admission it is reduced by the combination lever in the ratio

$$\frac{es-eg}{es}$$

The motion from the expansion link can therefore be calculated as

$$out\_phase1 = out\_phase * \frac{es}{es+eg}$$

(outside admission), or

$$out\_phase1 = out\_phase * \frac{es}{es-eg}$$

(inside admission)

Now choose the angle of swing of the expansion link. Don Ashton recommended between 40 and 50 degrees total - 20 to 25 degrees either side of vertical - and again I have no quarrel with that figure. Neglecting die slip (see later) it can be seen that

$$out\_phase1 = ph * SIN\left(\varnothing/2\right)$$

where *ph* is known as the *depth in gear*, the distance that the centre of the die block lies above or below the pivot of the expansion link in full gear, and so

$$ph = \frac{out\_phase1}{SIN\left(\varnothing/2\right)}$$

where ø is the swing of the expansion link as above.

We can now choose a suitable value for *pr*. For reasons which will be explained later, this should be kept as small as possible but unless pin *r* is on the side of a box link arrangement, it will be found difficult to make *pr* much less than 1.25**ph*. Note that you will need to check for clearance using the drawing board, or preferably a CAD package.

As a starter value, the throw of the return crank *ao* can now be calculated as

$$ao = pr * SIN\left(\varnothing/2\right)$$

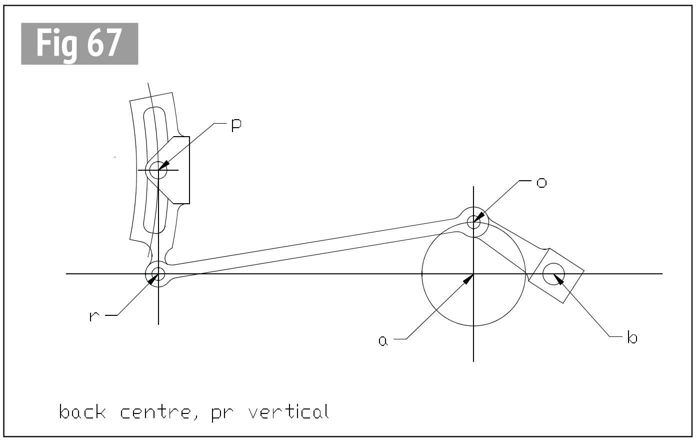
So far so good, and no great difficulty or complicated mathematics. Unfortunately, things are about to become somewhat more complex when we get to grips with backset.

Backset

Figure 67 shows a simplified arrangement of the return crank, eccentric rod and expansion link.

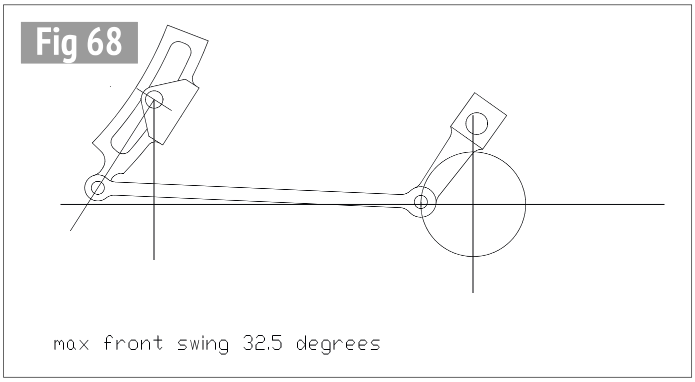
The return crank radius has been increased to exaggerate the ill effect. Point *r* is positioned such that *pr* is tangential to the curved slot in the expansion link at point *p*, the expansion link pivot. In fig 67, the piston is on back centre and the expansion link *pr* is vertical. You will notice that the horizontal distance between *a* and *r* is less than *or*, this being due to the angularity of *or*. When the axle turns such that *ao* and *or* are (more or less) in line as shown in **fig 68**, this angularity almost disappears and so point *r* moves forward by more than *ao*. In this case the expansion link swings 32.5 degrees. When the piston is at back centre the expansion link is vertical again as in **fig 69** but when the axle turns so that *ao* and *or* are in line again, as shown in **fig 70**, the angularity effect means that point *r* moves backward by less than *ao*, in this case by 27.5 degrees. This imposes a distortion on the valve events, leading to the cutoff at the front end being less than the cutoff at the back (or vice versa for inside admission).

The first thing to do is to attack the problem rather than the symptoms. If we make *pr* shorter, we reduce *ao*, which in turn reduces the angularity problem. This can be seen by comparing fig 63 with **fig 71**. The GWR 4 cylinder locomotives kept *pr* particularly



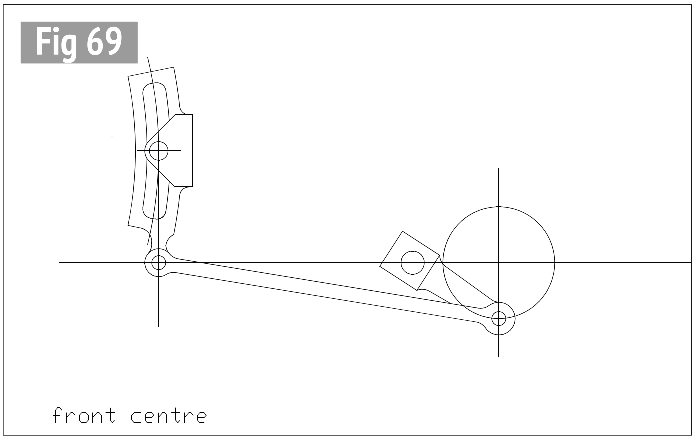
Backset at 0 degrees.

Fig 68



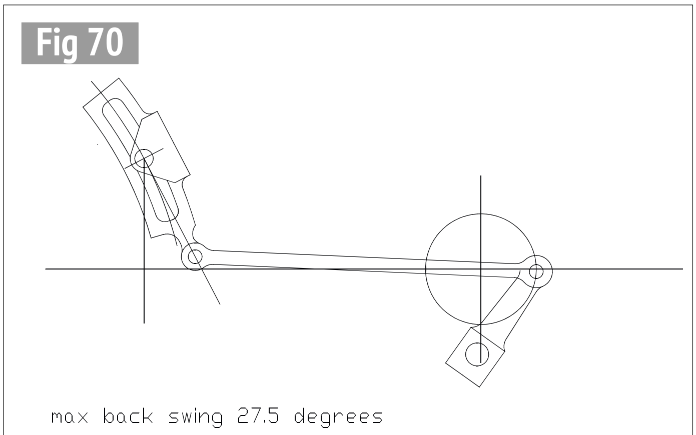
Backset at 90 degrees.

Fig 69



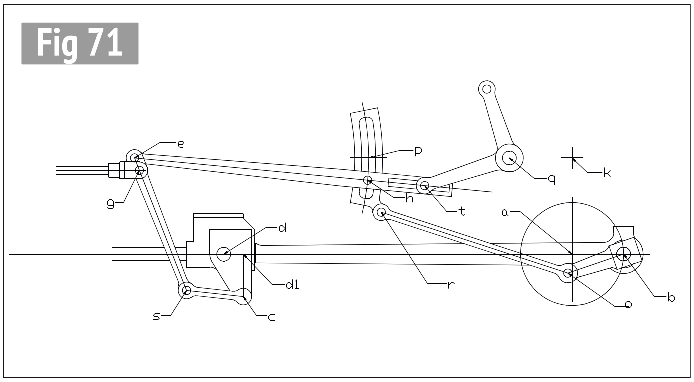
Backset at 180 degrees.

Fig 70



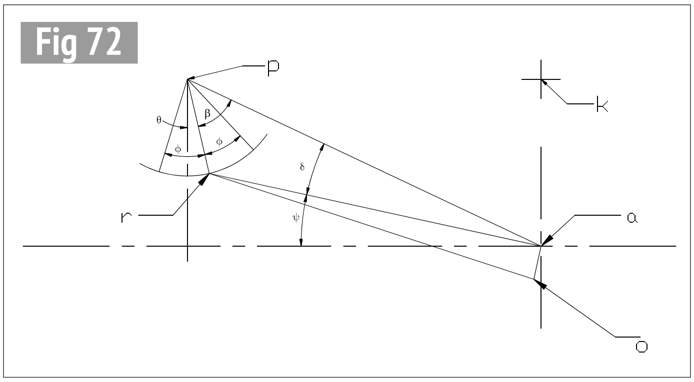
Backset at 270 degrees.

Fig 71



Reducing the angularity problem.

Fig 72



Geometry of angularity error.

small by mounting the pin *r* on the side of the link but this is unlikely to appeal aesthetically on an outside cylinder locomotive.

The way to reduce the remaining error is to reduce the length of *or* as shown in **fig 72**. Now *pr* is no longer vertical at front and back centres but the fore swing and back swing can be adjusted to improve the equality of the valve events. It seems to be accepted amongst model engineers that the maximum fore swing and back swing should be equalised and methods of achieving this by trial and error on a drawing board have been described many times, particularly by Martin Evans (not the current editor of ME, the other one!). Figure 72 shows a gear designed to achieve equal swings of the expansion link at ±30 degrees each side of the position at front and back centres. Note that the return crank radius is slightly reduced to get this swing and that the return crank is no longer 90 degrees from the main crank. Instead it is at 90 degrees to the line joining *a* to *r* when the piston is at front or back centre. Here, *pr* is no longer tangential to the curved slot at *p*, and the angle between the *pr* and that tangent is called the *backset angle*.

So far so good. You will however find that the maximum forward swing occurs when the crank has rotated 89.9 degrees from back centre, whereas maximum backswing occurs when the crank has rotated 83.8 degrees from front centre. Again, we are unlikely to get equal valve events at the front and back ends. Added to this we have the connecting rod angularity issue. The full-size locomotives I have data for do not comply with this method but it is the method advocated by many respected model engineers, and it does give acceptable results, so it would not be reasonable to dismiss it. However, some means of calculating the length of *pr* and the backset angle would be advantageous rather than the trial and error method.

Burrows' method

The first reference in *Model Engineer* to a method of calculating of backset was given by Dr F. Burrows, *Model Engineer*, 3rd September 1976. This method gives equal maximum swings of the expansion link and at equal rotations of the crankshaft from centres. Unfortunately, to achieve this Burrows had to abandon the usual convention that the centre of curvature of the expansion link is coincident with point *e* at front and back centres and so, as the cut off is increased, the lead at one end increases and the lead at the other decreases. There was a storm of correspondence involving Messrs Ashton, Ewins, Rowley, Holmes and Gettings, none of whom could bring themselves to agree with this feature and so, on 19th August 1977, Burrows gave a revised calculation giving equal leads and equal maximum swing. Burrows claimed that he could calculate exactly the position of all points in the gear exactly for a full revolution. I don't think this is possible and have no evidence that Burrows ever did it. He does however deserve credit for getting the ball rolling.

Gettings' method

In response to Burrows' later article, Mr Alan Gettings gave the following method, which is rather neater than Burrows' and gives exactly the same answers. Unfortunately, this article was very badly misprinted. The same method is given in Don Ashton's excellent little book (*Walschaerts Valve Gear for Model Engineers* - there is a similarly good little volume on Stephenson's and both have recently been republished as one volume). The following is extensively cribbed from this book. I've checked it and it is correct.

Referring to fig 72, having chosen the position of *p* and the angle of swing of the expansion link, we can calculate

$$ap = \sqrt{(ak^2 + pk^2)}$$

$$\beta = ACOS\left(\frac{pr * ap}{pr^2 + ap^2} * \{1 + cos\varnothing\}\right)$$

$$\theta = ATAN\left(\frac{pk}{ak}\right) - \beta$$

These equations are cribbed direct. We need not go into their derivation, which is a bit on the tedious side, but I have checked them and they are correct.

I mentioned earlier that the expansion link could be put higher or lower than calculation would suggest and that the gear would then be *inclined*. The angle of this inclination from horizontal is γ, horizontal being defined as the centreline of the cylinder, sloping upwards from *p* to *e* in mid gear being positive.

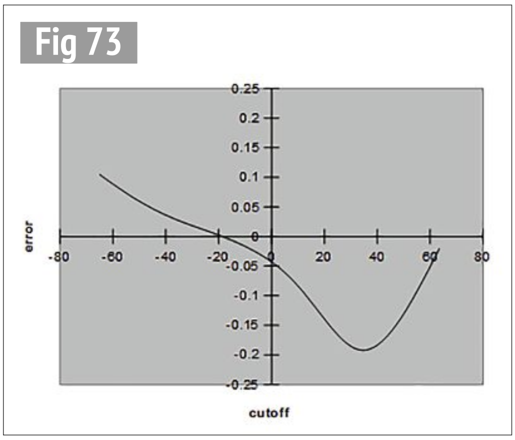
The backset angle must take account of any inclination of the gear and so

$$backset = \theta + \gamma$$

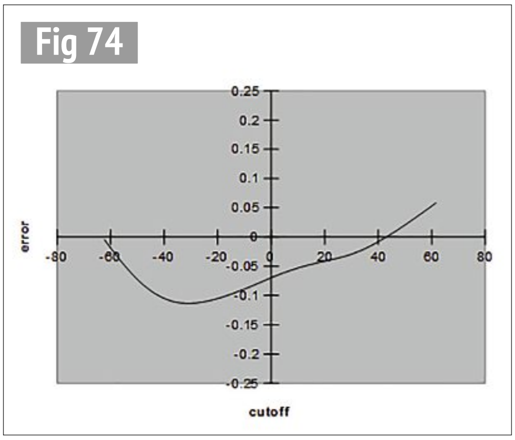
where θ is the inclination of the valve gear.

The return crank pitch circle radius is given by

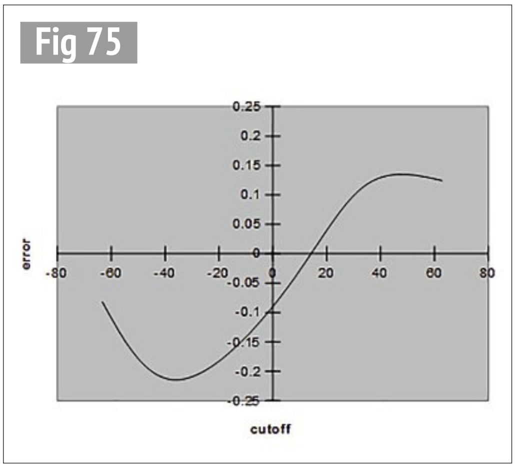
$$ao = \frac{ak * pk * sin\varnothing}{\sqrt{(ap^2 + pr^2)}}$$



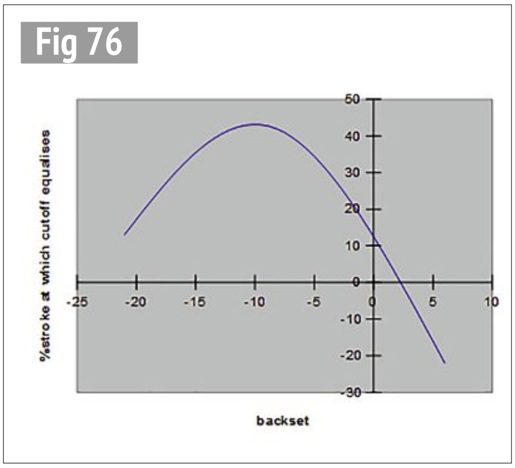
Inside valve motion – backset +6 degrees.



Inside valve motion – backset -8.2 degrees.



Inside valve motion – backset -21 degrees.



Inside valve motion – achieving equal cutoff.

The lengths of the eccentric rod *or*, the return crank length *ob* and the angle by which the return crank lags or leads the main crank can be got from the CAD. The original version of this article (ref 1, M.E.4718, June 2) gives a set of equations but they are a bit heavy. The computer model (later) actually does this for you anyway.

Results

Using the computer program which will be described later, I have calculated the motion of the inside valve of a GWR King for a full revolution of the axle and for a range of backsets and then plotted out the error against the cutoff in **figs 73**, **74** and **75**. The error is defined as

$$error = \frac{2 * (front\_cutoff - back\_cutoff)}{front\_cutoff + back\_cutoff}$$

to reflect the idea that a difference of say 5% end to end is more important at short cutoff than long cutoff. These three diagrams are then used to draw **fig 76**, which shows where equal cut off is achieved at the different backset angles.

As expected, no one value gives equality across the working range, say 15% to 75% cutoff. The surprising thing that comes out of this diagram is that wide changes in backset do not have that big an effect on equality of events. This is in agreement with Don Ashton's findings. The GWR men chose a backset that gives equality of cutoff at around 40%. I might have chosen a slightly shorter cutoff but they had to allow for the outside gear as well. The Kings had negative backset (frontset?) of -8.2 degrees. I think this is because it would be more geometrically sensible to define backset relative to *ar* (with the piston on front or back centre), which on a King slopes up quite steeply, but it would be a bit awkward to dimension the detail drawing to a construction line. The computer program defines the distance from pin *r* to the centre of curvature of the expansion link slot.

Lead will always be constant (because the gear was designed that way) and, as long as the port opening is enough at both ends, a small difference will not matter so we ought to aim at equality at the cutoff used most of the time. As well as the effect noted above, we need to take into account the angularity of the connecting rod, which distorts the motion of the piston, and any die slip which could make *depth in gear* different from one end to the other. How we at least partially achieve this will be explained in the next instalment.

●To be continued.

REFERENCES

Ref 20. *Backset in Walschaerts Valve Gear*, D.A. Webster, M.E.3960 7 January 1994