2 For conventional double-dwell motions, the 3-4-5 polynomial or modified sine functions are good all-around “workhorse” choices, and are better than modified trapezoidal motion when vibration of the follower is considered.

3 For low vibrations in conventional double-dwell motions, the 4-5-6-7 polynomial is superior and the cycloidal curve is an acceptable substitute as long as their somewhat higher accelerations do not overstress the parts. Differences in theoretical peak accelerations between various functions are less important than the actual peak value when follower vibration is factored in.

4 To obtain extremely low vibrations in applications where camshaft speed is essentially constant, a polydyne or splinedyne approach is best.

5 For nonconventional (i.e., not double dwell) industrial motions, especially when intermediate follower positions or velocities are specified, a polynomial or B-spline function will usually give the best design. If many boundary conditions are specified, then a polynomial may not work (or will be suboptimal) and a B-spline will be needed.

6 Pressure angles should be limited to about 30° with conventional translating followers, though if a low-friction ball-slide is used to guide the translating follower, then pressure angles up to about 35° may be tolerable. For oscillating arm followers, a maximum pressure angle of about 35° is generally acceptable.

7 When a roller follower is used, the absolute value of the radius of curvature ρ of the path of the roller centerline (pitch curve) must be kept larger than the radius R_f of the roller at all points to avoid undercutting. A ratio of $\rho \,/\, |R_f| > 2$ is a good target, though a ratio of 1.5 has been used successfully in some applications. The Hertzian surface stress should definitely be checked when this ratio is small.

8 When a flat follower is used, there can be no negative radius of curvature allowed in the cam surface contour.

9 When air cylinders are used as follower return springs, an accumulator should be used and the fittings, hoses, valves, etc., connected to the cylinder should be as large in diameter and as short as possible to minimize impedance.

10 “Full-complement” needle roller bearings used as cam followers may be shorter lived than caged roller bearings despite the superior load capacity of the former type. This is due to their poorer grease storage capacity and the fact that the uncaged needles rub on one another.

11 While a cylindrical roller follower has theoretically lower surface stress than a crowned follower in the same application, unless parallel alignment of the axes of the cylindrical roller follower and the cam is accurately and stiffly maintained, a cylindrical roller follower can actually have potentially higher surface stress than a crowned roller. If the alignment between cam and roller axes cannot be made accurate (including dynamic deflections), then a crowned roller may be needed, though its gentle crown radius can accommodate only slight axial misalignment.

12 Track cams with single roller followers will experience crossover shock, and the roller will have to reverse direction at the load reversal points, causing slip and wear. Some designers of track cams provide a follower spring to load the roller follower always against one side of the track. The other side of the track then becomes essentially an “insurance policy” against possible gross follower jump in the event of a follower spring failure or a tooling jam. Another approach to cure crossover shock and roller reversal is to use two rollers in the track, each contacting only one side of the groove and spring-loaded apart to accommodate slight deviations in groove width. A ribbed cam with two rollers pinching across the rib gives the same effect.