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Productive Design and Calculation of Intermittent Mechanisms with Radial Parallel Cams

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Abstract**—**The paper deals with the kinematics and automated calculation of intermittent mechanisms with radial cams. Currently, electronic cams are increasingly applied in the drives of working link mechanisms. Despite a huge advantage of electronic cams in their re-programmability or instantaneous change of displacement diagrams, conventional cam mechanisms have an irreplaceable role in production and handling machines. With high frequency of working cycle periods, the dynamic load of the proper servomotor rotor increases and efficiency of electronic cams strongly decreases. Though conventional intermittent mechanisms with radial cams are representatives of fixed automation, they have distinct advantages in their high speed (high dynamics), positional accuracy and relatively easy manufacture. We try to remove the disadvantage of firm displacement diagram by reducing costs for simple design and automated calculation that leads reliably to high-quality and inexpensive manufacture.

Keywords**—**Cam mechanism, displacement diagram, intermittent mechanism, radial parallel cam

I. Introduction

INTERMITTENT mechanisms are important elements in manufacturing systems fixed and flexible automation. These are often the manipulation movements in the various forms of rotary tables in production lines, packaging machinery, machine tools, etc. Mechanisms with radial cams are interesting because of a very precise manufacturing process. Despite the strong expansion of electronic cam mechanisms, there are still many tasks where it is not possible and even in the foreseeable future it will not be possible to replace mechanical cams with electronic cam mechanisms. In particular, they stand out for their dynamic properties that are improved with the advent of new materials.

Design of these mechanisms can be divided into two steps. In the first step a displacement diagram, which forms the starting point in the synthesis of cam mechanisms, is assembled. A displacement diagram is in principle an input-output function that defines the movement of the follower depending on the rotation drive shaft, resp. cam.

The second step, which will be given the most attention, is the synthesis of cams; cam profile is calculated according to the defined displacement diagram. The calculation of course is affected by other parameters (distance axis of rotation cam and follower, follower length, roller radius and number of rollers) that determine the actual dimensions and dynamic characteristics of the resulting mechanism. Based on these data the active surface (contour) of the actual cam, resp. production coordinates of contour, is calculated. In the case of symmetrical displacement diagrams (more precisely displacement diagrams, whose working part is symmetrical to its center) the resulting parallel cams mechanism, which is composed of two identical cams that are rotated each other mirrored, is created. This fact allows the production of both parts at once with inverse accuracy two cams to 0.01 mm.

However, a synthesis algorithm does not require these symmetries and thus allows creating also mechanisms based on unsymmetrical displacement diagrams and mechanisms with unsymmetrical initial rotation rollers. The disadvantage of these intermittent mechanisms is the different shape of the cams, which are needed to be produced individually. These results in significantly lower manufacturing precision compared with symmetric mechanisms.

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II. Design of Displacement Diagram

A displacement diagram can be defined as a function assigning the position of another element of the cam mechanism to the position of a specific element. For conventional cam mechanisms, it is thus a function assigning a specific location of the driven element (follower) v to a certain position ψ of the drive shaft. Along with the displacement diagrams, its first and second derivativeswith respect to position, which is often called as *first* and *second transfer function,* are usually defined. A displacement diagram itself, which means zeroth derivation, is often called only as a *displacement*. For the clear definition of shape displacement diagrams, their description in the form of unit displacement diagram is used. This diagram is defined by the displacement of size 1. The size of interval of independent variable varies depending on the category of displacement diagram (from -1 to 1, or from -0.5 to 0.5). The example of a simpledisplacement diagram in the unit form isin the figure below.

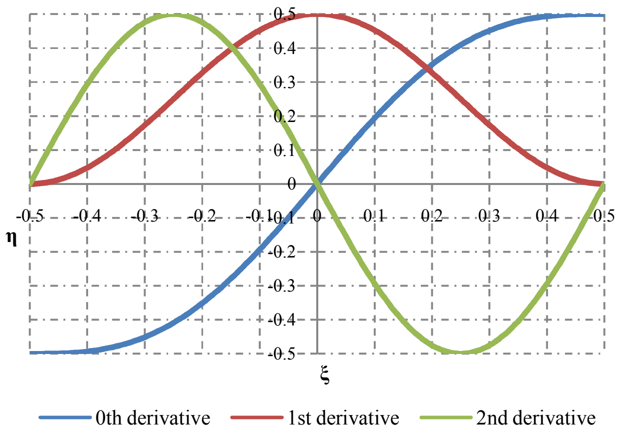

Fig. 1 Illustrative example of a displacement diagram in unit form

The first inevitable step was the creation of a tool that will be able to create these functions. The software tool is named KINzz 2 and includes a library of about 60 analytically given displacement diagrams taken from [1]. KINzz 2 is universally designed software, so the resulting displacement diagram is created by the composition of any number of elementary displacement diagrams and dwell intervals. The reusability of this tool is enhanced by adding options to import displacement diagrams of numerically specified displacement diagrams and ability to modify some basic types of displacement diagrams by the German standard VDI 2143 [2]. More detailed analysis of this issue transcends the scope of this article and is not necessary for the introduction of the research. Of course, there are options to save, load project and export into the data file for spreadsheet programs.

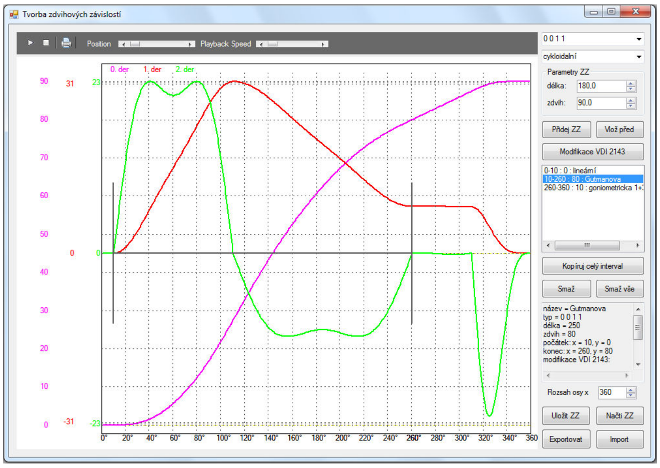

Fig. 2 Demonstration of program KINzz 2

The figure above demonstrates the features of KINzz 2. The horizontal axis represents the rotation of the cam, whereas the vertical one denotes zeroth, first, and second derivative of rotating follower by the cam rotation. Since the angular velocity is usually constant, the shape of function can be regarded as a derivativewith respect to time. An example illustrates the displacement diagram composed of a short dwell interval followed by two elementary displacement diagrams. Both are also modified by the standard VDI 2143 so that the transition between them was formed by non-zero the first derivative, resp. speed.

III. The Synthesis of Cam

The second tool that has been developed for the purpose of synthesis of the radial parallel cams is named KINy 2. The diagram below represents a block structure of the implementation of the program. Individual blocks will be discussed in the following paragraphs.

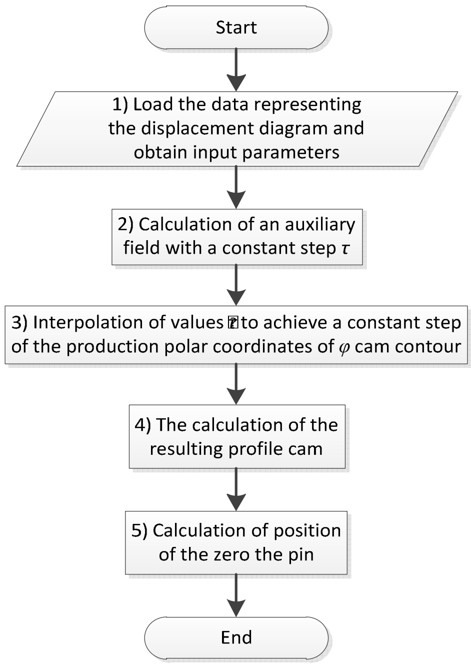

Fig. 3 Structure of synthesis cam

1) The program KINy 2 accepts as an input any of a symmetric or unsymmetrical displacement diagram. From a user it also requires dimensional parameters for the cam mechanism: *a*, *b*, *c* and the *number of rollers*. The chart below graphically illustrates the meaning of the first two parameters and other important variables.

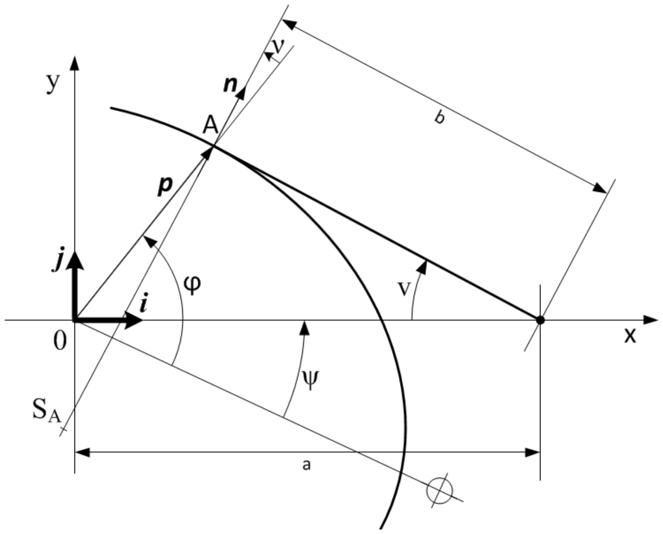

Fig. 4 The normal cut radial cam with a roller follower

In the center of the coordinate system there is the axis of cam rotation. The axis of rotation of the follower is located on the x-axis at a distance of *a*. The coordinate z-axis is not shown because it is parallel to the axis of the cam and follower, and should not make sense. Point *A* indicates the position of the axis of roller follower that moves the active cam surface. The trajectory center of the roller is known as a *pitch profile* that defines the vector *p* and the current position of center curvature S_A. The rotation angle of the follower denotes the letter *v* and the length of the follower indicates letter *b*. The normal angle is important to determine the cam surface and is denoted by the Greek letter *ν*. For correct understanding of the scheme, it is important to realize that besidesthe follower rotation there is also the rotating of the cam. The rotation angle of the cam against its zero point is marked by the Greek letter *ψ*. To determine the *pitch profile* of the cam,a vector function *p(v)*of the position of point *A* against the point zero of the cam is determined. The vector function *p(v)* is then only converted to the polar coordinates where *u* represents the length of vector and *φ* angle against zero point.

$$\boldsymbol{e}(\varphi) = \boldsymbol{i}\cos\varphi + \boldsymbol{j}\sin\varphi \tag{1}$$

$$\boldsymbol{p} = u(\varphi)\boldsymbol{e}(\varphi) \tag{2}$$

The reverse expression of the functions$u(v)$and $\varphi(v)$comes from the classical notation

$$\boldsymbol{p}(v) = \boldsymbol{i}x(v) + \boldsymbol{j}y(v) \tag{3}$$

It is possible to obtain a general expression using the basic formulas of the goniometric functions

$$u(v) = \sqrt{x^2(v) + y^2(v)} \tag{4}$$

$$\varphi(v) = \psi(v) + S\,arccos\frac{x(v)}{u} \tag{5}$$

After substitutingforthe generalfunction $x(v)$ and $y(v)$we get the basic relations for the synthesis of the pitch profile.

$$u(v) = \sqrt{a^2 - 2ab\cos v + b^2} \tag{6}$$

$$\varphi(v) = \psi(v) + S\,arccos\frac{a - b\cos v}{u} \tag{7}$$

For the calculation ofthe angleφ, the general relationship is supplemented bythe signfunction$S$, whichprovidesa correctcalculation for thefull 360°. It removestherestriction thatcyclometricfunctionsbroughtinto the calculation.The function takes the values 1 and 1 depending on which half-plane(defined by the x-axis) is point A.

$$S(v) = sign\,y(v) \tag{8}$$

To obtain the actual cam profile, it is necessary to express the angle of the normal, which is denoted by the Greek letter ν. The contactpoint of the roller and cam for any radius of the roller always lies on the line determined by the normal vector, as shown in the figure below.

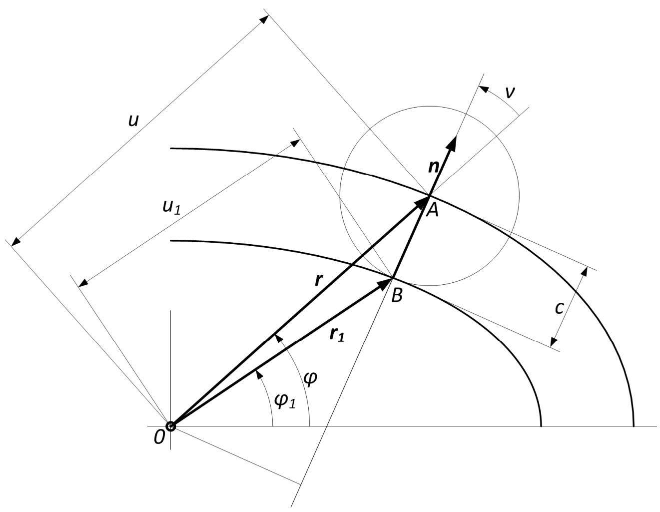

Fig. 1 Determination ofactual cam profile

The derivation of the analytical relation for the normal angle is a fairly complicated procedure, which is detailed in [1], so here only the final relationship is mentioned.

$$\nu(v) = -arctg\frac{\dot{v}ab\sin v}{\dot{\psi}(a^2 - 2ab\cos v + b^2) + \dot{v}b(a\cos v - b)} \tag{9}$$

In the relationtwo timederivatives appear,which do not presenta problem.$\dot{\psi}$represents theangular velocityof the rotation ofthe cam, which isconstantforour purposes. Due to thisconsideration,the time derivative $\dot{v}$can be converted to thederivativeaccording to the position of thecamv', which is given bydisplacement diagram (first transfer function). With the knowledge of thenormalangle,the equations for the recalculation of the pitch profileto the actualcam profile can be find using thegoniometricequationsand the law of cosines. The radius of therollerin the relationshipactsas a parameter*c*.

$$\varphi_1 = \varphi + arctg\frac{c \sin v}{u + c \cos v} \tag{10}$$

$$u_1 = \sqrt{u^2 + 2cu \cos v + c^2} \tag{11}$$

The last no less important parameter is the *number of rollers*. To determine the final cam profile, it is necessary to first determine the paths of all the rollers of which the smallest profile is then "cut out". The paths are closed curves, after which the individual rollers move. Before the calculation it is necessary to determine their count and length. Firstly it is important to define the total displacement, which defines the angle by which the follower rotates per one revolution of the cam. It is necessary to determine the *minimum number of revolutions*, after which a given roller returns to its initial position.

$$minimum\ number\ of\ revolutions = \frac{(least\ common\ multiple\ (360, total\ displacement))}{total\ displacement} \tag{12}$$

$$number\ of\ paths = \frac{number\ of\ rollers}{minimum\ number\ of\ revolutions} \tag{13}$$

2) Using the above equations for u and φ and substituting$v$ as a function of time τ (at constant revolutions $v$can be as well considered as the function of cam rotation ψ); we get the paths of the rollers centers, as the following figure demonstrates.

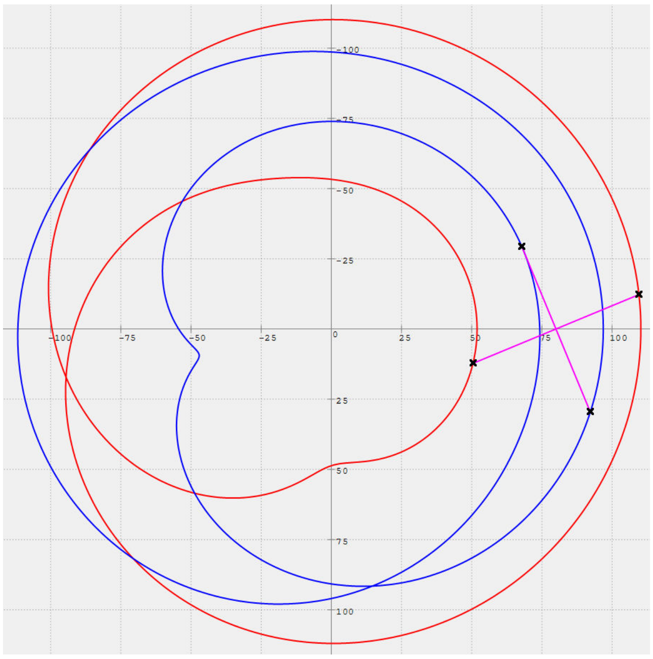

Fig. 2 Calculation of paths of centers rollers

In addition, the angle of normal and hence the actual cam profile for each point are determined.

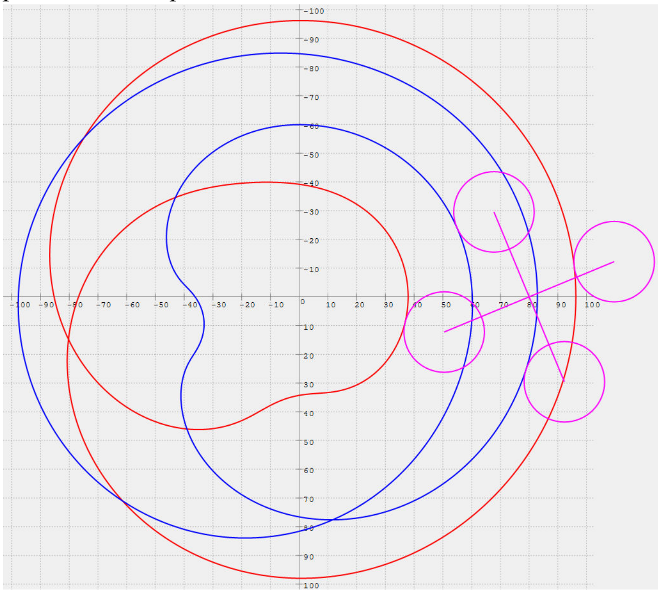

Fig. 3 Calculation of paths of rollers

3) For the purposes of further analysis and transfer to the production coordinates, it is required that the resulting coordinates were calculated with a constant step φ. The creation of cubic interpolation spline, which represents the relationship between φ and τ, provedto be the best solution in the end. With its help there are backward interpolated values τ to achieve a constant step φ. If the cam is well specified (it is geometrically feasible) the deviations of the calculated values φ from the desired values are lower by several orders of magnitude than the precision of the manufacturing process.

4) With the newly obtained values of τ a similar calculation as in step 2is done. After calculating the paths of individual rollers, the minimum profile that will represent a real cam will be cut off, as demonstrated in the figure below.

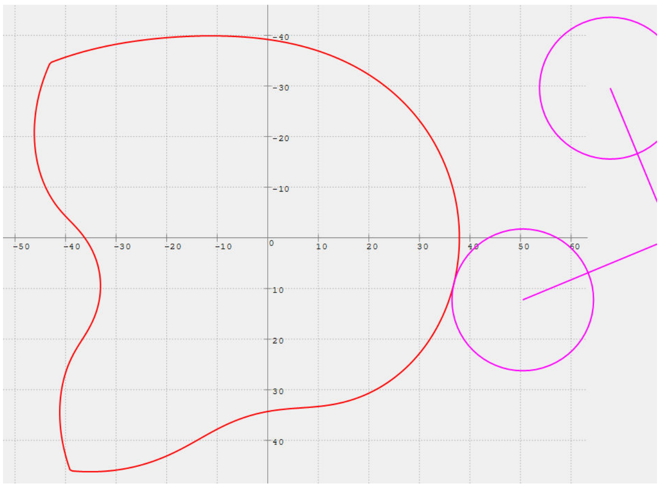

Fig. 4 Cut off the actual cam profile

5) For the completion of the whole mechanism it is necessary to produce a second cam. As mentioned in the introduction, when all the above mentioned symmetries are complied, the second cam is created from the same profile, just made after the mirror rotates. For the correct connection of both cams, a hole,the “zero pin”, is created in the profileto connect two profiles and ensure the correct mutual position. For the right determination of this point the center of the working part of displacement diagrammust be found. Since the displacement diagrams entering into the synthesis are specified numerically, it may not be the real center directly included in the data. Therefore,the interpolation by cubic spline is used again. To find a specific value quickly, Newton's method of tangents [3], which was adapted for this type of function, is used. The figure below shows the assembly of the cam mechanism from the previous images. For the better illustration of the function, the cam is rotated by 165°.

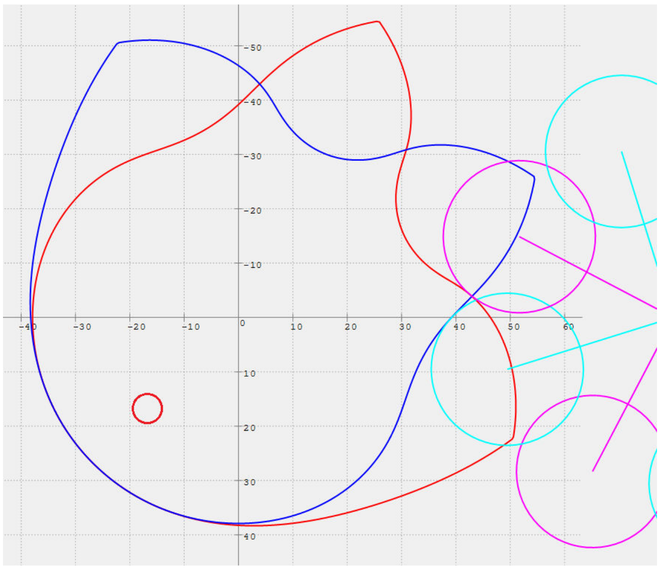

Fig. 5 Completed radial parallel cam

In the case of the violation of the symmetry of displacement diagram or symmetry of the initial rotation of the rollers, the resulting mechanism is composed of two different cams. The disadvantage of these mechanisms is a greater inaccuracy of production; however, they bring a huge amount of new possibilities in the construction of intermittent mechanisms. It is easy to implement mechanisms adapted to constant forceful exertion in one direction, which will have significantly better dynamic performance than the symmetrical arrangement.

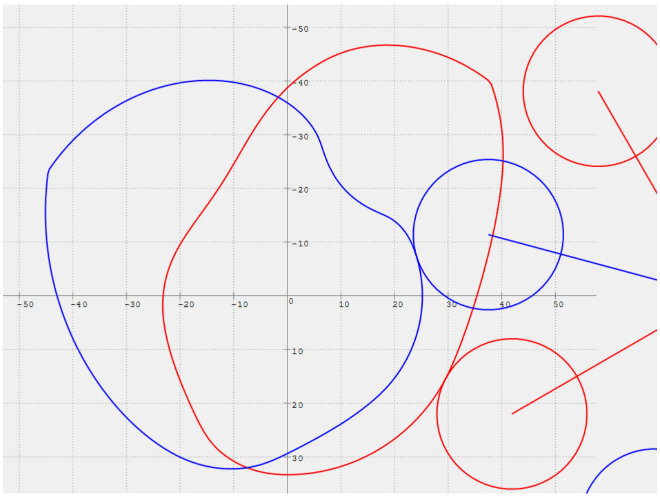

Fig. 6 Cam mechanism based on the unsymmetrical displacement diagram with unsymmetrical initial rotation of rollers

IV. Conclusion

Based on the real requirements of industry practice, there have been developed two software tools for automated design intermittent mechanisms with radial parallel cams. The first one allows very free design displacement diagrams, which allows its use for the design of various cam mechanisms or electronic cams. The second is directed towards the synthesis of radial parallel cams mechanisms. It also allows an easy and quick comparison of mechanisms with the different input parameters of cam profiles, and also selected kinetic-static parameters (angle of normal and pressure angle). Both tools are still being developed and improved according to the current requirements. Another aim is to implement advanced algorithms such as automated unloading cam profiles to improve the dynamic parameters of the mechanism.

Both tools are being developed in Microsoft Visual Studio 2010 in C# language and compiled for .Net Framework 4. Besides the standard Visual Studio libraries, two free accessible librarieswere used, too. For advanced mathematical functions library “Math.NET Iridium” [4] and a library for graphs A simple C # library for graph plotting [5].

Acknowledgment

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