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Parameter selection in the design of displacement and motion functions by means of B-splines

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Abstract

This work analyses the incidence of the parameter selection of B-spline curves, used in the design of displacement and motion functions, on its degree of freedom and shape. A complete design process based on the use of non-parametric B-spline curves and the convenience of selecting the curve parameters from the point of view of its practical application is shown. In order to make easy the design and use of the displacement function, the algorithms for derivation and integration of the B-splines used are presented. Three case studies validate the proposed design process and the selection of the adequate parameters. The first case presents the design of a displacement function of a roller follower driven by a disk cam; the corresponding cam profile and its prototype are shown. The second case presents the design of the motion function corresponding to the cutting unit of a manufacturing cardboard tube machine. The third case exposes the design of the displacement function of the bar feeding mechanism in a single-spindle automatic lathe, to produce a partial thread screw of hexagonal head.

Keywords: Parameters selection; Displacement function; Motion function; Non-parametric B-splines; Design process

1. Introduction

In the mechanism and machine design process, the designer must often deal with specific design requirements related with the movement of their elements and also must ensure the continuity conditions of the displacement or motion function. This function describe the evolution of a kinematic variable (displacement, velocity, acceleration) in function of the time or of one generalized coordinate. Such functions should be defined taking into account that they must comply with the restrictions associated with the technological task of a certain device. For example, in the case of a cam-follower mechanism, very often used as movement generator, its process of synthesis begins with the definition of the displacement function of the follower according to the design requirements demanded [1]. Traditionally, for the definition of such displacement functions, piecewise curves have been used, defined by means of basic functions [1, 2] such as: cycloidal function, modified trapezoidal function and polynomial function on canonical base; all of them allow the general design requirements to be met. However, the use of the mentioned curves is highly conditioned when the set design applications specify requirements, as for example: displacement of the follower with constant velocity, peak acceleration values, etc., all guaranteeing the global continuity of the displacement or motion function.

Displacement functions are often mentioned in the specialized literature about cam-follower mechanisms, in which the use of polynomial functions by means of Bézier and B-spline schemes is introduced. The mentioned functions have arisen in the field of computer aided geometric design (CAGD), and are adequated for the synthesis of movement functions according to the specific design requirements such as those mentioned above [1, 2]. Sahu, L. K et al [2] present a large review of the state of the art study about the definition of displacement or motion function using basic and synthetic curves, particularly applied to cam-follower mechanisms. The authors reache the conclusion that the trend of modern cam design is that splines –Bsplines, NURBS and Bézier– are replacing basic curves as the mathematical representation of the cam profile because of their versatility, ease of application and flexibility. Sateesh et al [3] propose the design of a velocity curve using a B-spline polynomial of degree three and with six control points, with which they obtained a B-spline curve equivalent to a Cycloidal curve, with lower values of maximum acceleration; thus they optimize the shape of the planar velocity curve and obtain, by means of the integration and derivation of such curves, the displacement and acceleration functions. The authors explain the B-splines function advantages in the design of displacement functions. Hua Qiu et al [4] propose a procedure to optimize the design process of a displacement function by means of a uniform B-spline and

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optimize the values of the control points by taking the design requirements into account.

Xiao and Zu [5] perform an optimization study of a cam profile for an internal combustion engine that is driven by cylindrical groove cams with translating followers –positive drive cams. In the paper, the authors make a comparison of the results of different combinations of optimization methods and tools in order to define the follower displacement function. The combination of the use of a B-spline and a Genetic Algorithm method gives the authors the best results obtained in their work.

Jamkhande, A. et al. [6] expose the impact of different cam profile options designed using Polydyne, N-Harmonic and B-spline methods on a field problem of cam wear for high speed engine application. The authors conclude that the cam lobe designed by B-spline method gives the better results and it is the better design option. Neamtu, M. et al. [7] show how to design cam profiles using Non-Uniform Rational B-Splines (NURBS) curves which support functions are trigonometric splines. The authors conclude that the trigonometric splines are an attractive alternative to classical polynomial splines for cam design and that NURBS curves have the useful property that their offset are of the same type and hence they also have an exact NURBS representation.

Angeles [8] proposes a synthesis method of plane curves that comply with the pre-established local geometrical properties using periodic splines. The author argues for the use of these splines due to the facility of manipulation of the obtained equations. The proposed method permits a set of unkown interpolation points to be determined. The author also studies the problem of the modification of the geometrical properties of functions and curves as a particular case of the presented procedure. The author gives two examples where the proposed procedure is validated; the first one shows the synthesis of a rise phase in a follower movement function. The second example refers to the synthesis of the cover for a cylindrical pressure vessel guaranteeing adequate continuity in the union between segments. As a result, the author shows that the use of periodic splines in the synthesis of plane curves to solve classical problems seems to have an advantage over the traditional methods.

Hidalgo et al. [9] propose optimizing cam profiles with negative radius followers using Bézier curves to describe the follower motion function. They take a Bézier ordinate as a parametrization parameter.

In previous works [10, 11], the authors of this paper have used non-parametric Bézier curves to synthesize the follower displacement functions in constant-breadth cam mechanisms that drive both translating and oscillating followers. The authors' works present the desmodromic condition that the follower displacement functions must meet, as well as the calculation algorithms that permit such functions to be obtained; the procedure that automatically guarantees their C^2 global continuity is also shown. Additionally, expressions that permit the derivatives of the follower displacement function and the generation of the cam profile to be obtained are presented.

Bézier curves and B-splines are both functions defined by control points and the difference between them is the effect that the change of the control points' position has on the curve shape. Since, in Bézier curves, the change of position of a control point affects the whole curve –it produces a global change–, in B-spline curves this change only affects a segment of the curve; it means that a B-spline curve has the property of local control [2, 12]. Thus, the B-spline curve is a good tool that permits the designer to respond to stricter design requirements more effectively, which coincides with the criterion exposed by the authors above mentioned. Ganesh [13] exposes that use of the de Boor algorithm in the evaluation of the B-spline curve does not requiere the kwnoledge of the B-spline basis function, which is a great advantage from the numerical evaluation point of view.

The design of the motion function is often based on the inclusion of a high number of free parameters in the definition of the function and the subsequent adjustment of these by optimization procedures.

In the present work, the authors propose to establish a set of parameters –pass points and derivatives in them– depending on the requirements of the motion function and to leave a limited number of parameters of free choice to make the final adjustment of the motion function. This work analyses the incidence of the parameter selection of B-spline curves, used in the design of displacement (motion) functions, on its degree of freedom and shape. A complete design process based on the use of non-parametric B-spline curves and the convenience of selecting the curve parameters from the point of view of its practical application is shown. In order to make easy the design and use of the displacement function, the algorithms for derivation and integration of the B-splines used are presented. Three case studies validate the proposed design process and the selection of the adequate parameters. The first case presents the design of a displacement function of a roller follower driven by a disk cam; the corresponding cam profile and the 3D model and a prototype of this cam are shown. The second case presents the design of the displacement function corresponding to the cutting unit of a manufacturing cardboard tube machine. The third case exposes the design of the displacement function of the bar feeding mechanism in a single-spindle automatic lathe, to produce a partial thread screw of hexagonal head.

2. Non-parametric B-splines curves: characteristics, De Boor algorithm, derivation and integration of a B-spline

A non-parametric B-spline curve of degree *n* can be defined as the join of *L* polynomial segments of degree *n*, where each one is the image of an interval $\left[u_i, u_{i+1}\right]$ of the domain, with *u* being the parameter of the curve. The real numbers u_i, which are freely chosen, are known as knots. These knots are given as an ascending sequence called knots vector

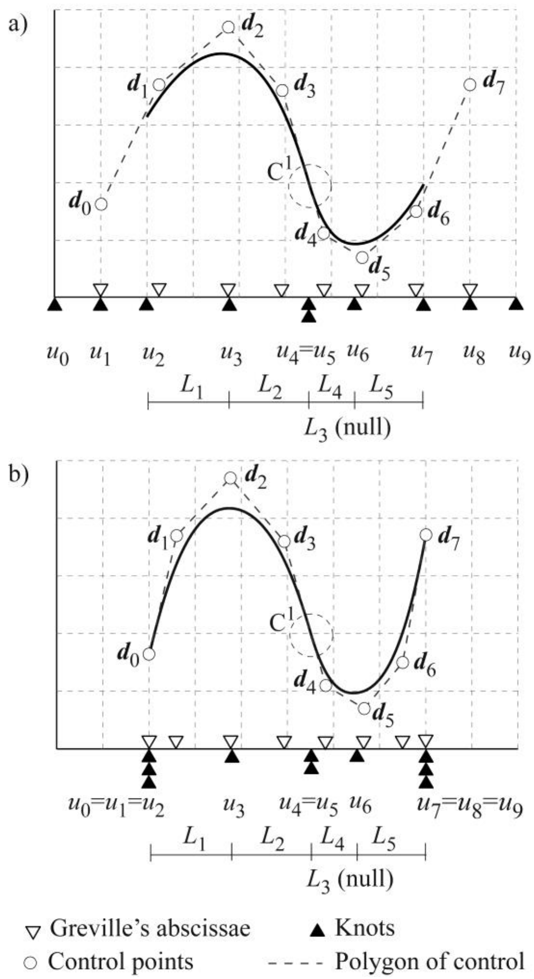

Fig. 1 Non-parametric B-splines curves of degree $n = 3$ and its parameters. L_i are polynomial segments.

$\boldsymbol{u} = \{u_0,...,u_{L+2n-2}\}$. The curve is associated to a control polygon that is defined by $L+n$ control points $\boldsymbol{d}_i$. Segments with null length may exist if there are coincident knots –knots with the same value. In such cases, only one knot with multiplicity *m* is indicated, which is equal to the number of knots that are initially coincident. Then the knots vector can be written compactly by giving each knot its value and multiplicity, $\boldsymbol{u} = \{\{u_0, m_0\},\dots,\{u_i, m_i\},\dots\}$. The abscissae of the control points –Greville's abscissae ξ_i – are calculated from the knots vector: $\xi_i = (u_i + ... + u_{i+n-1})/n$ where $i = 0,...,L+n-1$ and the ordinates d_i of these control points are freely chosen and can be conditioned by design requirements. Fig. 1 shows two B-spline curves that have been defined with two knots vectors with the same number of knots, of different values but with the same ordinates as the control points.

In B-spline curves, the minimum continuity in the union between polynomial segments is given by the degree *n* of the B-spline and by the multiplicity *m* of the knot in the mentioned union. Thus, in the knots with multiplicity *m*, the curve has a minimum continuity C^{n-m}. If the multiplicity *m* of a knot is equal to the degree *n* of the curve, then a control point arises, its abscissa ξ_i coincides with the knot and, in this case, the curve passes through this control point, thus obtaining a pass point. Fig. 1 b) shows that knots u_0 and u_7 have multiplicity $m = 3$ equal to the degree of the B-spline; therefore the end control points are defined as pass points of the curve and the Greville's abscissae are: $\xi_0 = u_0$ and $\xi_7 = u_7$. In Fig. 1, the knots u_4 and u_5 are coincident knots so, in this way, they define a single knot of multiplicity 2 and consequently the L_3 segment is a null length segment. In the design of a movement function, it is preferable or even necessary that the end points be also pass points and thus the end knots have multiplicity equal to the degree of the curve.

To calculate the ordinates $d(u)$ of the B-spline curve points the de Boor algorithm can be used. It is numerically stable and does not require the B-spline basis function to be known, which is an advantage in the evaluation process of the curve [13] and justifies the choice of the de Boor algorithm in this work.

2.1 de Boor algorithm

The de Boor algorithm is used for the calculation of the ordinates $d(u)$. It is a recursive algorithm with $k = 1,...,n-m$ levels and uses the following formulation, where *I* is the subscript of the first knot of the segment that contains the *u* value:

$$d_i^k(u) = \frac{(u_{i+n-k} - u)}{(u_{i+n-k} - u_{i-1})} d_{i-1}^{k-1}(u) + \frac{(u - u_{i-1})}{(u_{i+n-k} - u_{i-1})} d_i^{k-1}(u) \quad (1)$$

with $i = I-n+k+1,...,I-m+1$

The iteration process begins with $d_i^0 = d_i$ and $d_{i-1}^0 = d_{i-1}$ and the value of the B-spline curve in the abscissa *u* is:

$$d(u) = d_{I-m+1}^{n-m}(u) \quad (2)$$

2.2 The Derivative of a B-spline curve

The derivative of a non-parametric B-spline curve of degree *n* is another non-parametric B-spline curve of degree $n-1$. The derivation of a B-spline curve can be obtained from the knots vector $\boldsymbol{u} = \{u_0,...,u_{L+2n-2}\}$, the degree *n* and the ordinates of the control points d_i of the original B-spline curve.

The knots vector $\boldsymbol{u}'$ of the derivative curve is equal to the knots vector of the original curve, without its first and last knots. Its expression is:

$$\boldsymbol{u}' = \{u'_0, u'_1...,u'_{L+2n-4}\} \text{ with } u'_i = u'_{i+1} \text{ and } i = 0,...,L+2n-4 \quad (3)$$

The Greville's abscissae ξ'_i and the ordinates d'_i of the control points of the derivative curve are:

$$\xi'_i = \frac{1}{n-1}(u_{i+1} + ... + u_{i+n-1}) \text{ with } i = 0,...,L+n-2 \quad (4)$$

$$d'_i = n\frac{d_{i+1} - d_i}{u_{n+i} - u_i} \text{ with } i = 0,...,L+n-2 \quad (5)$$

2.3 Integration of a B-spline curve

The integration process of a B-spline curve is derived from the expressions Eq. (4) and Eq. (5) shown in the above section. First of all, one knot of value equal to or lower than the first knot and another knot of value equal to or higher than the last knot should be added to the knots vector of the B-spline curve that is going to be integrated. Then, the new Greville's abscissae of the integrated curve are calculated and, usually, the new ordinates are calculated from the first one.

In curves that pass through the first and the last control points, if their knots vector $\boldsymbol{u}$, the degree n and the ordinates d_i are known, the expressions that make it possible to obtain the integrated curve are:

$$'\boldsymbol{u} = \{'u_0, 'u_1 ..., 'u_{L+2n}\} \text{ with } 'u_0 = u_0,\ 'u_i = u_{i-1},\quad 'u_{L+2n} = u_{L+2n-2} \quad i = 1,...,L+2n-1 \tag{6}$$

$$'d_{i+1} = 'd_i + \frac{d_i}{n+1}('u_{n+i+1} - 'u_i) \text{ with } i = 0,...,L+n-1 \tag{7}$$

$$'\xi_i = \frac{1}{n+1}('u_i + ... + 'u_{i+n}) \text{ with } i = 0,...,L+n \tag{8}$$

To calculate the ordinates of the control points $'d_i$, it is necessary to impose an integration condition on the integrated function, which is usually the initial value of the B-spline; therefore, it is equal to $'d_0$.

3. Motion function design process using non-parametric B-splines

Usually the motion functions are designed from a set of global specifications –i.e. continuity–, local specifications –i.e. dwell segments– and free segments – without special requirements. The most used local specifications are:

– Pass point: prescribed point of the motion function.
– Straight horizontal segment: segment of the motion function with a constant value.
– Straight segment with constant slope.

Fig. 2 shows a planar displacement function $s(\theta)$ with a set of the mentioned specifications and taking θ as an independent variable.

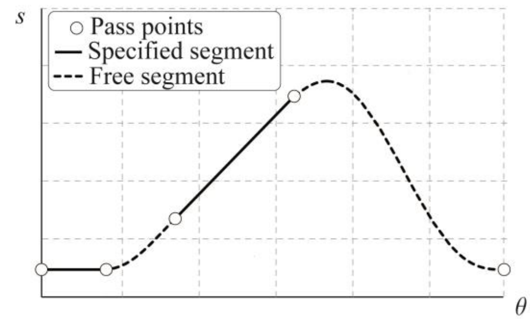

Fig. 2 Specification of the motion function.

The motion function design process proposed in this work uses a single non-parametric B-spline curve that allows the imposed specifications to be met. At the same time, it provides free choice parameters –ordinates of the control points d_i and knots u_i – to define the free segments.

The motion function design process consists of the following steps:

1. Making a sketch of the desired motion function, taking the specifications of the pass points and straight segments into account.

2. Defining the global continuity C of the motion function. Maximum continuity $C_{\max} = 3$ has been implemented in the computer application to guarantee the continuity until the third derivative.

3. Defining the pass points numerically and in order, including the initial point and the final point of the straight segments. For each pass point it is necessary to define its abscissa x_i, its ordinate y_i and also its successive derivatives y^m until the degree n imposed by the continuity. Thus, the notation to describe the pass points is $P_i = \{x_i, y_i, y_i', ...\}$. To define a straight segment between two consecutive points, their first derivative must coincide with the slope of the straight line and the successive derivatives in such points must be null.

4. Establishing the B-spline degree $n_0 = 2C + 1$, which does not require additional knots to be used in defining the B-spline.

In the case of taking the degree $n < n_0$, $n_0 - n$ middle knots must be given in the non-straight segments, thereby generating the additional control points that are necessary to guarantee the continuity of the motion function.

In the case of taking the degree $n > n_0$, there are some generated control points whose ordinates are not conditioned by the continuity. The choice of these non-conditioned ordinates permits the shape of the curve to be modified.

3.1 Examples of the design process

Two examples of the use of the above-mentioned design process are presented. In the first one, Fig. 3, a double dwell displacement function $s(\theta)$ is designed using continuity $C = 2$ and degree $n = n_0 = 5$; therefore in this case, additional middle knots to define the whole B-spline curve are not required. Here, five pass points have been defined; the first two points define the beginning and the end of the first dwell segment, the third and the fourth points define the second dwell segment and the fifth one defines the end point of the displacement function. The pass points, expressed in degrees, millimeters and the consistent units for the derivatives, are:

$$P_1 = \{0°, 10\text{ mm}, 0, 0\}, \quad P_2 = \{40°, 10\text{ mm}, 0, 0\},$$
$$P_3 = \{240°, 40\text{ mm}, 0, 0\}, \quad P_4 = \{300°, 40\text{ mm}, 0, 0\},$$
$$\text{and } P_5 = \{360°, 10\text{ mm}, 0, 0\}$$

Thus, the displacement curve is defined by 5 knots with multiplicity 5 and consists of four non-null polynomial segments of degree 5. The control polygon has 21 control points, all of them with ordinates conditioned by the continuity; the knots vector written in brief is: $\boldsymbol{u} = \{\{0°,5\},\{40°,5\},\{240°,5\},\{300°,5\},\{360°,5\}\}$.

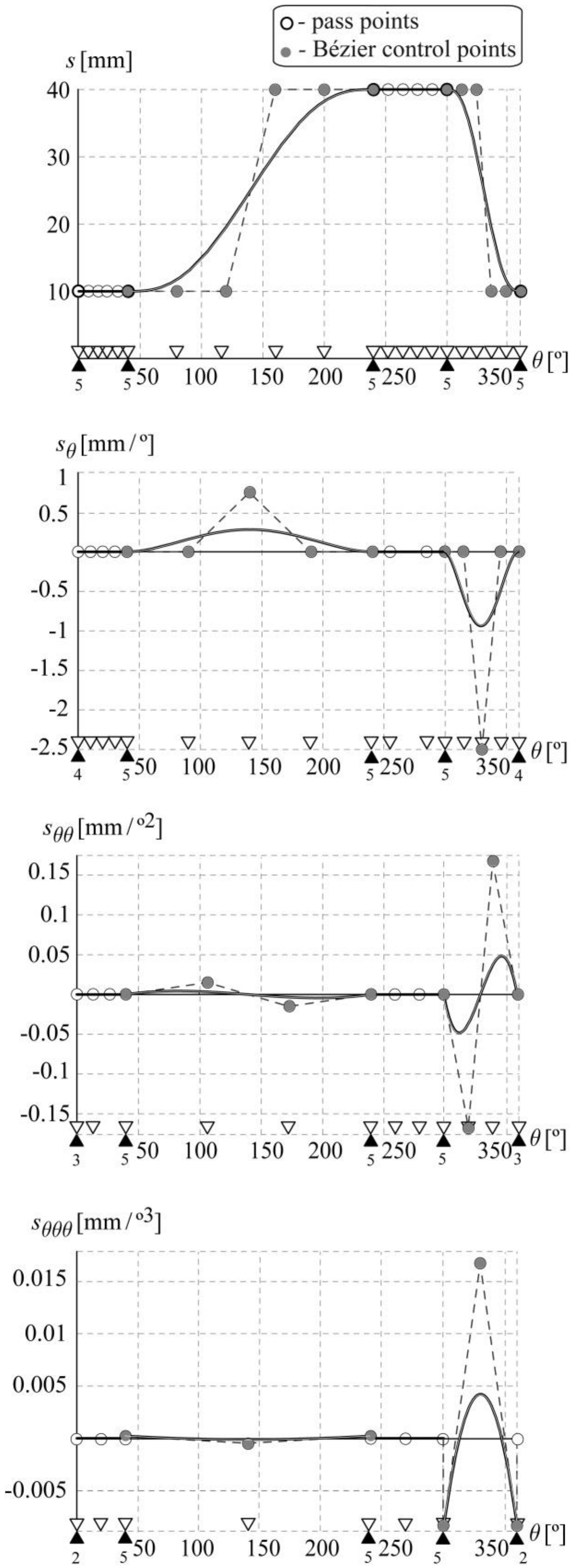

Fig. 3. Displacement function $s(\theta)$ and its first three derivatives with respect to θ, designed with a B-spline curve (in black color) and with Bézier curves (in grey color, overlapped to the B-spline). A degree $n = 5$ and continuity $C = 2$ is used in the design of both displacement functions. The multiplicity of the knots of the B-spline curve is indicated numerically under the knot symbol.

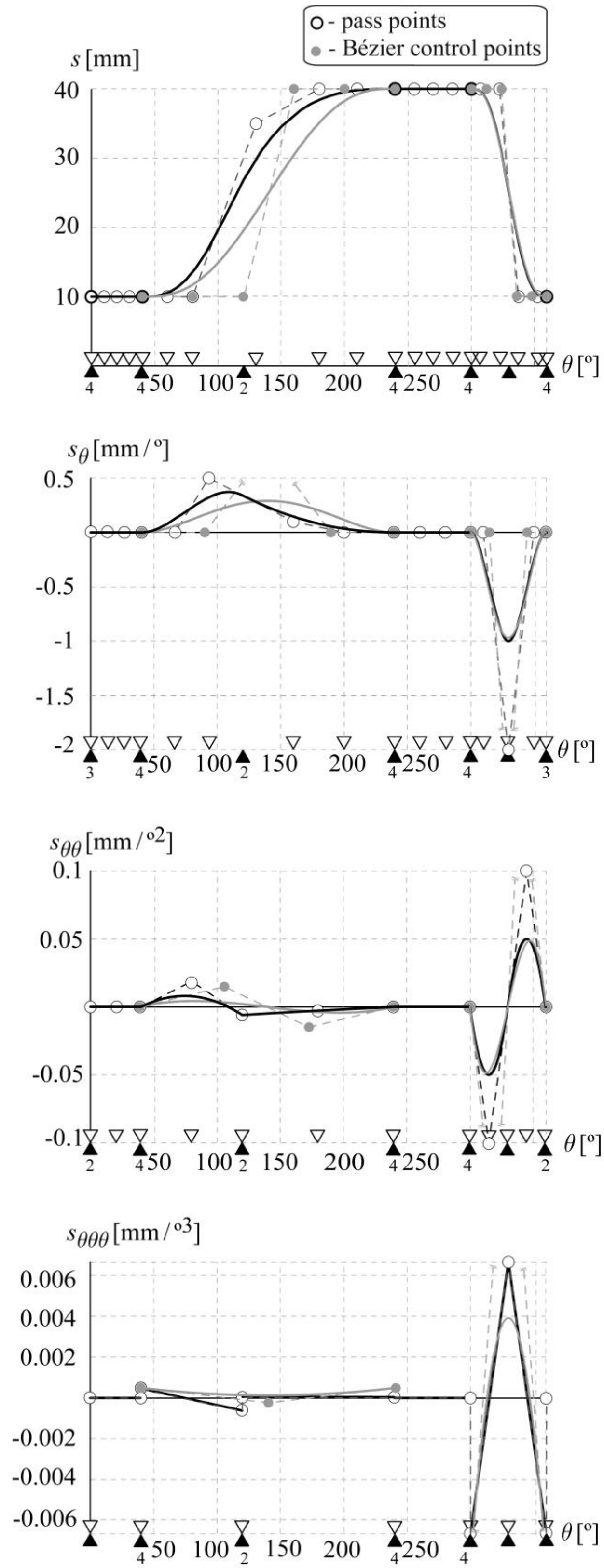

Fig. 4. Displacement function $s(\theta)$ and its first three derivatives with respect to θ, designed with a B-spline curve of degree $n = 4$ and two middle knots (in black color); and with Bézier curves of a degree $n = 5$ (in grey color). A continuity $C = 2$ is used in the design of both displacement functions.

The graphs of Fig. 3 have been obtained with the procedures described in section 2. The continuity of the proposed displacement function and of its three derivatives can be observed in them. Fig. 3 also shows the design of the same displacement function using non-parametric Bézier curves that are drawn in grey color.

From Fig. 3 it can be observed that in the case of using a B-spline of degree $n = n_0$, which does not require additional knots to be used in its definition, both curves –B-spline and non-parametric Bézier curve– coincide, due to that control points of both control polygons are equally spaced and are conditioned by the continuity. The control points of the Bézier curve in the rise and return segments –free segments– are indicated in grey color. The pass points of both curves are indicated with circles drawn with thick line.

The second example, Fig. 4, defines a displacement function with the same requirements as the previously designed function but using degree $n = 4$. In this case, where $n < n_0$, it is necessary to use middle knots to define the B-spline. In order to guarantee the continuity of the function and simultaneously to have the freedom to modify the shape of the curve, a middle knot of multiplicity 2 is used. In the second segment of the curve –non straight line segment–, a middle knot with multiplicity 1 is included to guarantee the global continuity. Thus, the curve is defined by the following compact knots vector:

$\boldsymbol{u} = \{\{0^o,4\},\{40^o,4\},\{120^o,2\},\{240^o,4\},\{300^o,4\},\{330^o,1\},$
$\{360^o,4\}\}$ and consists of 6 non-null polynomial segments of degree 4. The control polygon has 19 control points with ordinates conditioned by the continuity and one with a free ordinate in the third polynomial segment. Fig. 4 also shows the design of the displacement function by means of non-parametric Bézier curves used in the first example – with degree $n = 5$ and continuity $C = 2$. For adequate the graphical representation of control polygons of Bézier curves to the scale used in the first, second and third derivatives graphs obtained using B-spline curve, such polygons have been limited to the size of the grid box.

From Fig. 4 it can be observed that in the case of using a B-spline of degree $n < n_0$, which implies that middle knots must be given in the free segments, additional control points are necessary to guarantee the continuity of the displacement function. Overlapping Bézier curves in these graphs shows the differences between both displacement functions designed and their derivates, and the greater freedom of modifying the shape of the displacement function using B-splines instead of using Bézier curves.

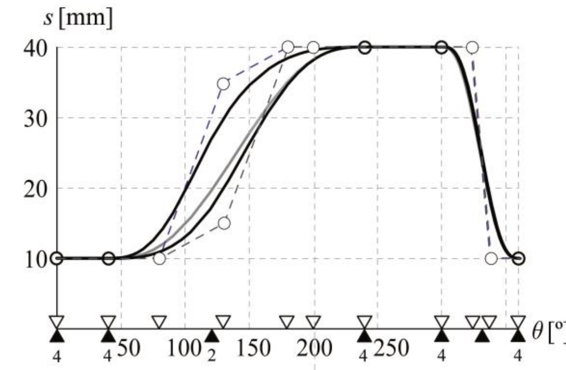

Fig. 5. Comparison between two displacement function obtained by modifying the free ordinate of the previous example.

Fig. 5 shows the overlap of the displacement functions designed by means of B-splines of degree $n = 4$, obtained by the use of different values of the free choice ordinate of the control point located in the rise segment. In this figure, the possibility of locally modifying -local control– the motion function by introducing middle knots is shown. The original displacement curve –which coincides with Bézier curve used–, Fig. 3, is indicated in grey.

4. Considerations about the incidence of the parameter selection of B-spline curves, used in the design of displacement or motion functions

In this section, we present considerations about the parameter selection of the B-splines curve; these parameter are: the degree n of the B-spline to be used, the continuity C desired in the displacement or motion function, the number of knots to use and their multiplicity m. The considerations are:

– To guarantee the continuity C at the pass points and impose on them the derivatives until order C, it is necessary to provide, in principle, at least $2(C+1)$ control points between two consecutive pass points.

– If the degree of the B-spline is n, without middle knots between two consecutive pass points, there are $n+1$ control points in the domain defined by the two pass points. This implies that without middle knots, it is necessary that the minimum degree of B-spline is $n_0 = 2C+1$ (Table 1). It is possible to use middle knots, which increase the number of polynomial segments of the B-spline, to reduce the degree n or to model the shape of the motion function, as has been seen in the examples of the section above. Each middle knot with multiplicity m increases in m the number of control points. The minimum continuity in this knot is, in principle, $n-m$; therefore, it must be $m \le n-C$ (Table 2).

– If a degree $n < n_0$ is used it is necessary to use middle knots between each pair of consecutive pass points (Table 2). The sum of the multiplicity of the middle knots must be, in principle and as a minimum, $n_0 - n$. If the sum of the multiplicity is taken bigger than $n_0 - n$ the additional control points that are generated have free ordinates and this can be used to model the shape of the motion function. In this case, and if there are added knots between the ends of a straight line, the ordinates d_i of the control points not conditioned by the continuity should be calculated in advance; however, it is possible to organize the algorithm to calculate the Greville's ordinates in such a way that defining the middle control points to guarantee the continuity should not be necessary.

– If a straight line between two consecutive pass points without additional calculations is desired, in order to place the middle control points correctly it is necessary that the degree *n* of the B-spline must not be higher than $n_0 = 2C+1$.

– Table 3 shows the case of use a degree $n > n_0$ where appear some additional control points not conditioned. Table 4 summarizes the relation among the desired continuity *C*, the degree *n* of the B-spline and the number of the additional control points to guarantee the continuity. The negative values in the table indicate the number of control points with free ordinates, not conditioned by the continuity.

Table 1. Relation among the number of control points in the domain defined by two consecutive pass points (without middle knots between them), the minimum degree n_0 and the continuity *C*

C	n_0	No. of control points conditioned by *C*
1	3	4
2	5	6
3	7	8

Table 2. Relation among the number of additional control points in the domain defined by two consecutive pass points (with middle knots between them), the degree $n < n_0$ and the continuity *C*

C	*n* ($n < n_0$)			No. of middle knots			Maximum multiplicity of the middle knots			No. of additional control points		
1	2			1			1			1		
2	4	3	2	1	2	3	2	1		1	2	
3	6	5	4	1	2	3	3	2	1	1	2	3

Table 3. Relation among the number of additional control points (not conditioned by the continuity) in the domain defined by two consecutive pass, the degree $n > n_0$ and the continuity *C*

C	n_0	*n* ($n > n_0$)			No. of control points not conditioned by *C*		
1	3	4	5	6	1	2	3
2	5	6	7	8	1	2	3
3	7	8	9	10	1	2	3

Table 4. Relation among the number of additional control points, the degree *n* and the continuity *C*

n / *C*	2	3	4	5	6	7
1	1	0	-1	–2	–3	–4
2		2	1	0	–1	–2
3			3	2	1	0

5. Case studies

Three case studies are presented to show the use of the proposed design process using B-spline non-parametric curves for the motion function and the incidence of the chosen curve parameters is explained. The first case shows the design of a displacement function of a roller follower driven by a disk cam, and the cam profile has been calculated and checked. Also, a prototype obtained by additive manufacturing is shown. In the second case, the design of the motion function corresponding to the cutting unit of a machine that produces continuous cardboard tubes is shown. The third case exposes the design of the displacement function of the bar feeding mechanism in a single-spindle automatic lathe, to produce a screw of hexagonal head.

5.1 Design of a displacement function of a roller follower driven by a disk cam

In the proposed cam-follower mechanism (Fig. 6), the follower must make the following movement: *i)* an initial displacement (rise) with a value of 17,5 mm for the first 90º of cam rotation angle, *ii)* a dwell at this value during the next 45º of the cam rotation angle –an intermediate dwell–, *iii)* a second upward displacement to reach the maximum displacement $s_{max} = 30$ mm while the cam rotates another 90º, *iv)* a second dwell during the next 45º of rotation of the cam –upper dwell– and *v)* finally, a return to the start position. The cam should rotate at high velocity, so the displacement function requires a continuity $C = 3$. The following design parameters have been chosen: *i)* a commercial roller follower with a radius of $R_f = 15$ mm and *ii)* a base radius of the cam $R_b = 50$ mm; these parameters are necessary to generate the profile of the cam without singularities.

Taking the specified design requirements into account, a sketch of the desired motion function is made, but not shown. There are six pass points defined for the displacement function $s(\theta)$: the first and the last points indicate the start and the end of the motion function; the other 4 points are used to define the two dwell segments of the funtcion. Thus there are two straight line segments with null slope and three free segments. The pass points are defined consecutively as follows:

$$P_1 = \{0°, 0 \text{ mm}, 0, 0, 0\}, \quad P_2 = \{90°, 17.5 \text{ mm}, 0, 0, 0\},$$
$$P_3 = \{135°, 17.5 \text{ mm}, 0, 0, 0\}, \quad P_4 = \{225°, 30 \text{ mm}, 0, 0, 0\},$$
$$P_5 = \{270°, 30 \text{ mm}, 0, 0, 0\} \text{ and } P_6 = \{360°, 0 \text{ mm}, 0, 0, 0\}.$$

Taking into account the considerations exposed in section 4, from the requirement of continuity $C = 3$ a degree $n = 7$ $(n = n_0 = 2C + 1)$ has been chosen for the design of the displacement function; thus middle knots are not required to guarantee the continuity and therefore there are no control points of free ordinates. This function is defined by a knots vector with 6 knots, with multiplicity 7, and consequently by 36 conditioned control points.

Fig. 7 shows the follower displacement function that complies with the specified design requirements, and its three first derivatives. The continuity until the third derivative can be observed.

From the designed displacement function $s(\theta)$, the cam profile according to the procedure presented by Zayas et al [10] is generated. Fig. 8 shows the CAD model of the obtained cam and the photo of the materialization of the cam by means of a manufacturing additive in a 3D printer.

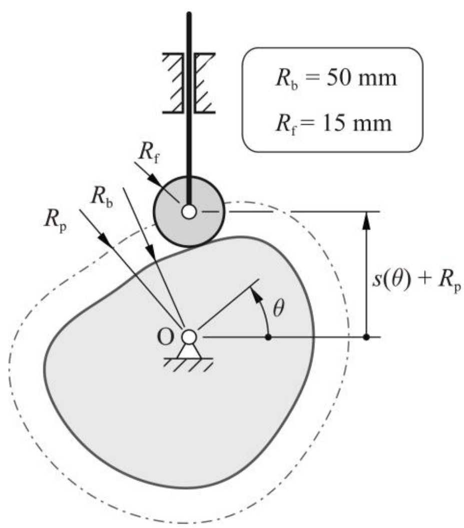

Fig. 6. Cam-follower mechanism. Design geometrical parameters.

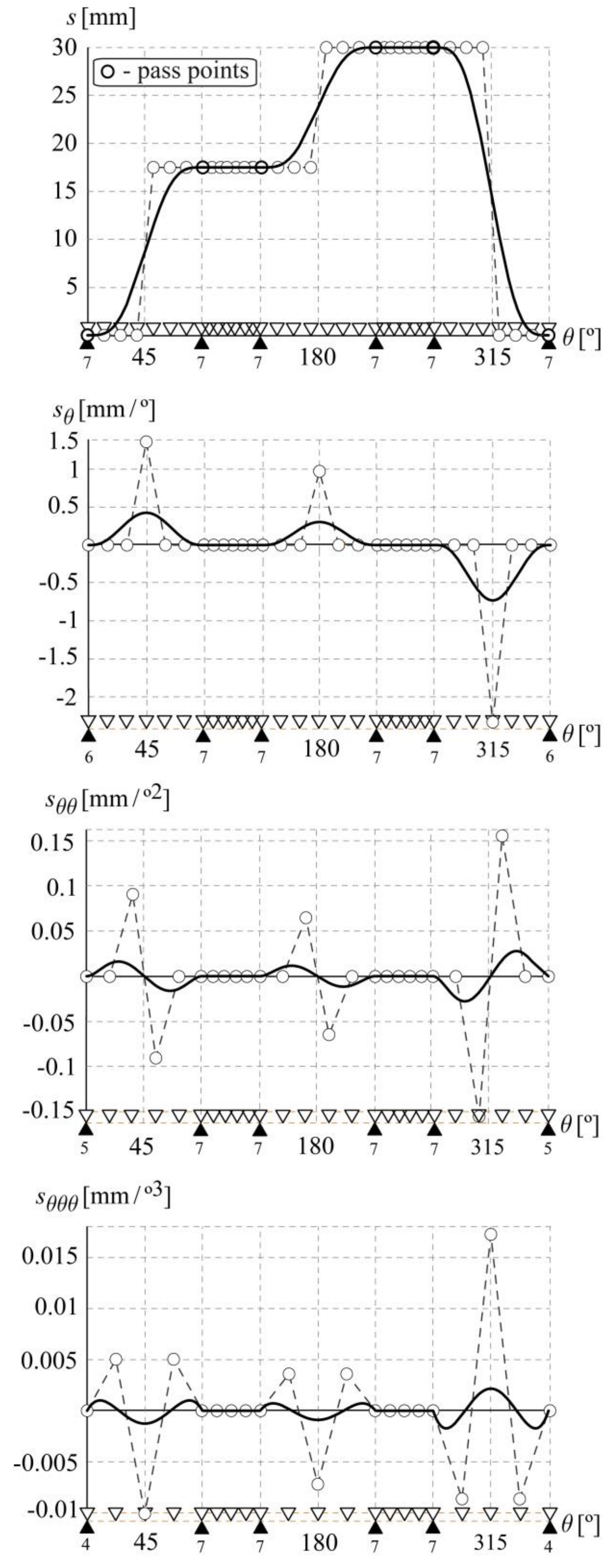

Fig. 7. Graphs of the displacement function $s(\theta)$ and its three derivatives with respect to the parameter θ.

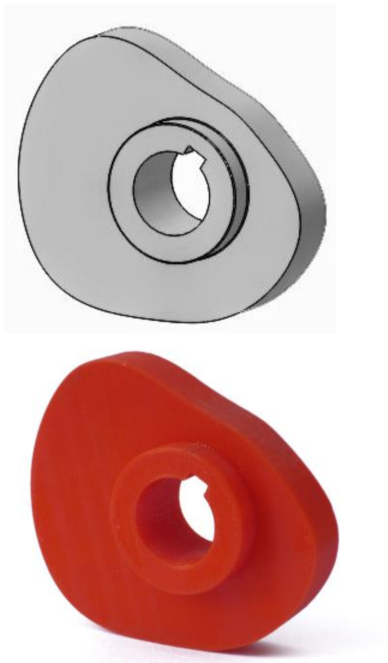

Fig. 8. 3D Model and its prototype, whose profile is obtained from the displacement function presented in Fig.7.

5.2 Design of motions functions of a cutting unit of a cardboard tube manufacturing machine

This case is an example of producing individual items from a product obtained in continuous form.

Cardboard tubes are used for a variety of things, for example, as supports of paper coils, supports of fabric coils, in packing, etc. The cardboard tube manufacturing machine consists of two modules: the main module that forms the cardboard into a continuous object, and the cutting unit, whose function is to cut the cardboard tube to the required length. Fig. 9 shows a simplified illustration of the cardboard tube cutting operation. The second module has a circular saw that is located on a sled with a displacement in the same direction as the axial axis of the cardboard tube. While the sled is performing the cutting operation, it keeps moving at the same velocity as the tube and then begins the return manoeuver to the start position to make a new cut. This manoeuver must be done smoothly, without abrupt changes of velocity. So, continuity C^1 is taken for the velocity motion function. In the manufacturing of a particular model of cardboard tube a velocity forming $v_{\text{tube}} = 0.4$ m/s is used and the tube must be cut to a length $l_{\text{tube}} = 800$ mm. The cutting time is $t_{\text{cutting}} = 0.5$ s and, while the saw is making a cut, the tube and the saw are simultaneously moving forward a distance of 200 mm. Fig. 9 shows the saw, which is located on the tool holder, the velocity expressed in mm/s, its derivative –the acceleration– and its integral –the displacement function.

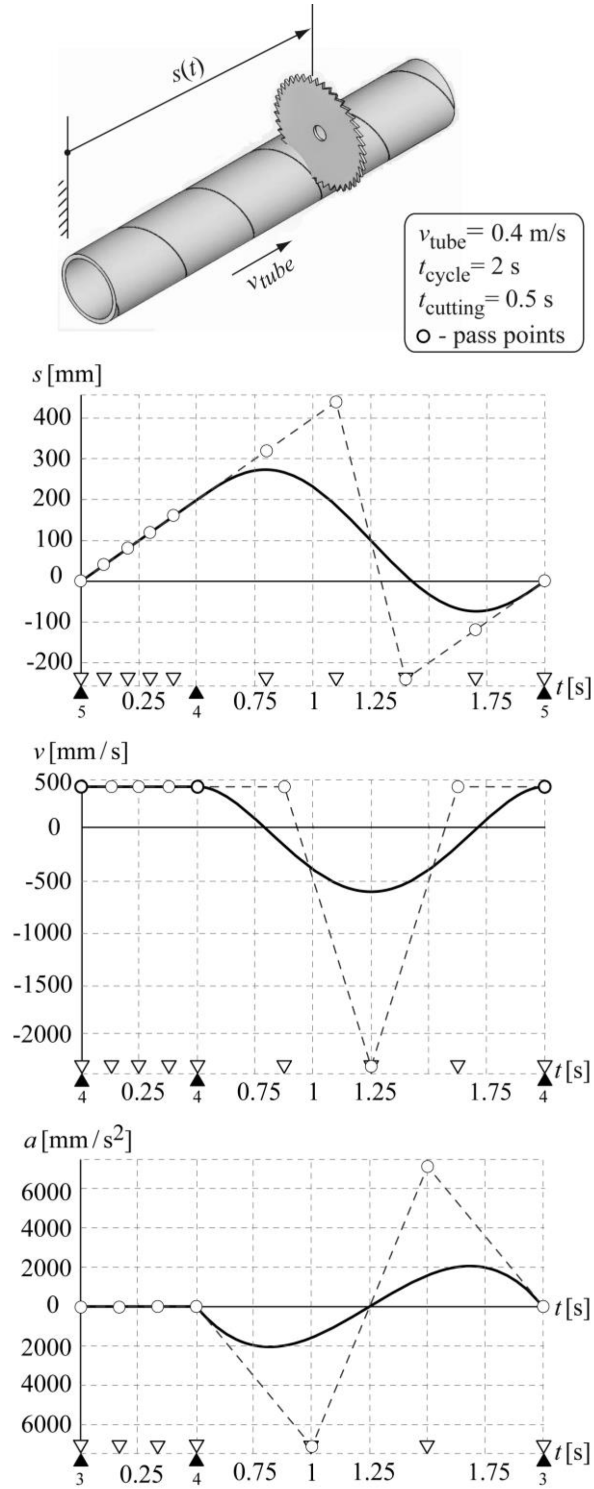

Fig. 9. Cardboard tube cutting operation. Graphs of the velocity function $v(t)$, its integral –the displacement $s(t)$ – and its derivative –the acceleration $a(t)$.

Three pass points are taken for designing the motion function in velocity; the first point indicates the start of the motion function and simultaneously the beginning of the constant velocity segment; the second one defines the end of this segment and the beginning of the return manoeuver and the third indicates the end of the function. These pass points are defined as: $P_1 = \{0\ \text{s}, 400\ \text{mm/s}, 0\}$, $P_2 = \{0.5\ \text{s}, 400\ \text{mm/s}, 0\}$ and $P_3 = \{2\ \text{s}, 400\ \text{mm/s}, 0\}$. According to Table 1, if continuity C^1 is wanted and a degree $n = 3$ is taken, a solution can be obtained that does not require additional control points and, therefore, requires no additional knots. This is not an adequate solution for the present case. Here, at least one additional freely-chosen control point must be used to guarantee the value of the integral –the displacement– at the end of the cycle. If a degree $n = 4$ is used then one free additional control point will appear in each segment defined by two consecutive pass points. The ordinate of such points in the first segment is fixed by the fact that this segment must be a straight segment –of constant velocity. The ordinate of the additional control point in the second segment should be chosen by trial and error, or using Eq. (8) so that the displacement be null at the end of the cycle; it only requires calculating the ordinate of the last control point from the curve passing through it. Hence, the curve is defined by a vector of three knots with multiplicity 4 that generate 9 control points; 8 points with ordinates imposed by the continuity and the straight segment; and 1 point with a freely chosen ordinate to guarantee the null displacement at the end of the cycle.

Alternatively, the displacement function might be designed first, afterwards proceeding by derivation to obtain the velocity and the acceleration laws.

5.3 Design of the displacement function of the bar feeding mechanism in a single-spindle automatic lathe.

The third case study is an example of producing individual screws obtained from a raw bar with hexagonal cross-section in a single-spindle automatic lathe. In automatic lathes all movements of cutting tools, their sequence of application, feeding of raw material, parting off and unloading the finished part, are done by the machine, without the operator's interference. In those lathes, automation of the movements is done by means of cams.

The machining operations to produce a partial thread screw of hexagonal head consist of three steps (Fig. 10): 1) cylindrical turning, 2) threading and 3) cutting-off.

Fig. 10 a) shows a simplified sketch of the front view of the screw obtained from the hexagonal bar and also the tools arrangement for the operations above mentioned. Fig. 10 b) exposes the dimensions of a screw of metric M8, according to DIN 931, in order to produce it from the raw bar and the axial movement of the bar and it's clamp that makes possible to machining the parts of the screw (the bar also rotates around its axis while the tools are machining the metal; this rotation is not considered in the present study). Both the radial translation of the cutting tools and the axial translation of the bar are driving by means of cams (not showed in the simplified sketch), which stablish the intervention sequence of each tool

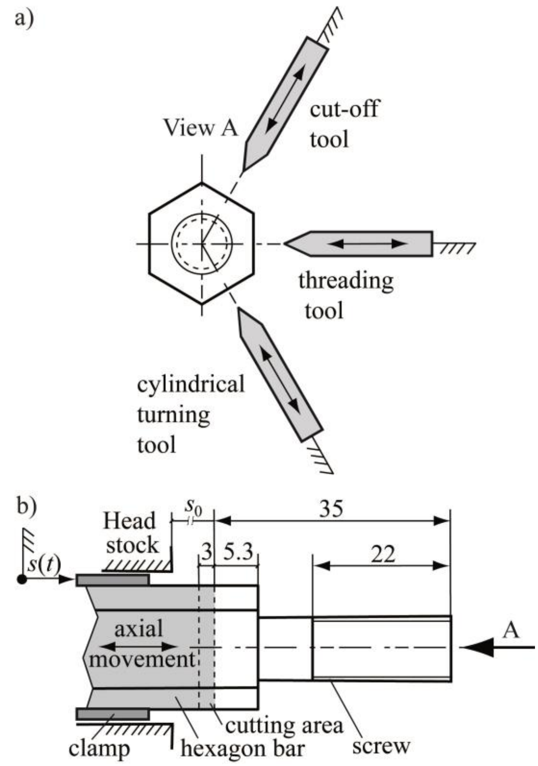

Fig. 10 a) Simplified sketch of the front view A of the screw obtained from the hexagon bar and the tools arrangement for machining operations, b) Dimensions of the screw to be produce and, the axial motion of the hexagonal bar and of the clamp that hugs it.

(by means of three radial cams) and the corresponding axial movement sequence of the bar (by mean of a cylindrical cam that drives a clamp that hugs and releases the bar according to the automatic machining process). The machining steps to produce the screw require to stablish an adequate sequence of intervention of each tool and also the corresponding sequence of movement of the bar.

The last mentioned sequence requires to design a complex displacement function of the clamp that hugs and moves the bar. This sequence is selected to apply the proposed design process of section 3. From Fig. 10 b) the principal dimensions of the screw are: the total length $l_{tot} = 35$ mm, thread length $l_{thread} = 22$ mm, thickness head $k = 5.3$ mm. The normal length is calculated as $l_{norm} = l_{tot} - k = 29.7$ mm. The initial position of the bar with respect to the head stock of the lathe is $s_0 = 25$ mm; from here the manufacturing cycle starts, according to the phases described in Table 5. The cycle duration of the screw machining process is $t_{cycle} = 24$ s, corresponding to an angular velocity of the cam $\omega_{cam} = 0.2618$ rad/s.

The displacement function $s(t)$ of the clamp (Fig. 10 b) to be designed must fulfil the sequence of movement of the bar according to the manufacturing phases established in Table 5. Furthermore, it must guarantee the requirement of keeping a constant translation velocity of the bar and also a smooth displacement of it while turning and threading operations are taking place. Thus, continuity $C = 2$ is taken for the displacement function, guarantying continuity up to the second derivate. Taking into account the requirements above mentioned, an initial sketch of the desire displacement function is made, but not shown.

There are 22 pass points defined for the displacement function $s(t)$ that are highlighting in Fig.11; thirteen of them are used to define 7 dwell segments corresponding to phases where the tools are approaching to their working position, are returning to their initial position and to the cutting operation –indicated with the capital letter C in Fig.11– in which the bar (and the clamp) remains without axial motion; the last dwell segment corresponds to the moment when the clamp releases the bar, after it has been cut and separating the screw obtained.

Other six pass points permit to stablish straight segments with constant slope, contained in the turning and threading phases –indicated with the capital letters A and B in Fig.11. It must be mentioned that, in these phases, the free segments that permit the transition between two consecutive segments are also considered. The last three pass points permit to define the three free segments located just before and after the cutting phase to obtain the whole displacement function. The pass points above mentioned are defined as:

$$P_1 = \{0\ \text{s}, 25\ \text{mm}, 0, 0\},\quad P_2 = \{1\ \text{s}, 25\ \text{mm}, 0, 0\}$$
$$P_3 = \{1.5\ \text{s}, 26.5\ \text{mm}, 10\ \text{mm/s}, 0\},$$
$$P_4 = \{4.5\ \text{s}, 53.2\ \text{mm}, 10\ \text{mm/s}, 0\}$$
$$P_5 = \{5\ \text{s}, 54.7\ \text{mm}, 0, 0\},\quad P_6 = \{5.5\ \text{s}, 54.7\ \text{mm}, 0, 0\}$$
$$P_7 = \{6\ \text{s}, 25\ \text{mm}, 0, 0\},\quad P_8 = \{6.5\ \text{s}, 25\ \text{mm}, 0, 0\}$$
$$P_9 = \{7\ \text{s}, 26.5\ \text{mm}, 10\ \text{mm/s}, 0\},$$
$$P_{10} = \{10\ \text{s}, 53.2\ \text{mm}, 10\ \text{mm/s}, 0\}$$
$$P_{11} = \{10.55\ \text{s}, 54.7\ \text{mm}, 0, 0\},\ P_{12} = \{11\ \text{s}, 54.7\ \text{mm}, 0, 0\}$$
$$P_{13} = \{11.5\ \text{s}, 25\ \text{mm}, 0, 0\}, P_{14} = \{12\ \text{s}, 26.5\ \text{mm}, 7\ \text{mm/s}, 0\}$$
$$P_{15} = \{15\ \text{s}, 45.5\ \text{mm}, 7\ \text{mm/s}, 0\}, P_{16} = \{15.5\ \text{s}, 47\ \text{mm}, 0, 0\}$$
$$P_{17} = \{16\ \text{s}, 47\ \text{mm}, 0, 0\},\quad P_{18} = \{16.5\ \text{s}, 60\ \text{mm}, 0, 0\}$$
$$P_{19} = \{20.5\ \text{s}, 60\ \text{mm}, 0, 0\},\quad P_{20} = \{21.5\ \text{s}, 60\ \text{mm}, 0, 0\}$$
$$P_{21} = \{23.5\ \text{s}, 22\ \text{mm}, 0, 0\},\quad P_{22} = \{24\ \text{s}, 25\ \text{mm}, 0, 0\}$$

Taking into account the considerations exposed in section 4, from the requirement of continuity $C = 2$ a degree $n = 5$ has been chosen for the design of the displacement function; thus middle knots are not required to guarantee the continuity and therefore there are no control points of free ordinates. This function is defined by a knots vector with 22 knots, with multiplicity 5, and consequently by 106 conditioned control points.

Table 5. Description and duration of the phases of a screw manufacturing process.

Phase duration t(s)	Phase description
0 - 1	Cylindrical turning tool approaching to its 1st working position (bar in its initial position s_0 = 25 mm)
1 - 5	1st cylindrical turning pass (bar moves forward 29.7 mm at constant velocity)
5 – 5.5	Cylindrical turning tool move away of the bar (bar remains without axial motion)
5.5 - 6	Cylindrical turning tool is approaching to its 2nd working position (bar returns to its initial position)
6 – 6.5	Cylindrical turning tool keeps getting closer to its 2nd working position (bar remains without axial motion)
6.5 – 10.5	2nd cylindrical turning pass (bar moves forward 29.7 mm at constant velocity)
10.5 - 11	Cylindrical turning tool returns to its initial position and remains there (bar remains without axial motion)
11 – 11.5	Threading tool approaching to its working position (bar returns to its initial position)
11,5 - 15,5	Threading operation (bar moves forward 22 mm at constant velocity)
15.5 - 16	Threading tool returns to its initial position and remains there (bar remains without axial motion)
16 – 16.5	Cut-off tool approaching to its initial working position (bar moves forward 13 mm, reaching the total length of the screw)
16.5 – 20.5	Cutting operation: the tool moves reaching the center of the bar and separating the screw from the bar (bar remains without axial movement)
20.5 – 21.5	Cut-off tool returns to its initial position (bar remains without axial movement)
21.5 – 23.5	Clamp release the bar, and returns 38 mm (35 mm of the screw and 3 mm of the cutting area).
23.5 - 24	Clamp hugs the bar and moves 3mm forward its initial position (a new cycle begins)

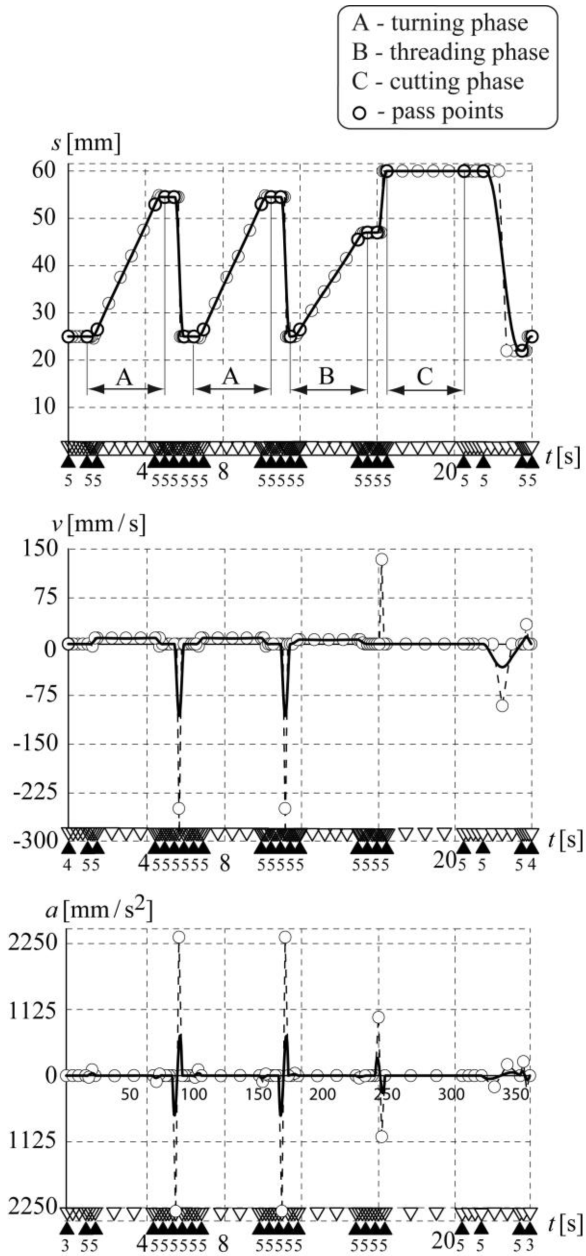

Fig. 11. Graphs of the displacement function $s(t)$ and its two derivatives with respect to the parameter t.

6. Conclusions

A procedure is proposed that allows displacement and motion functions to be designed, by means of non-parametric B-spline curves that are evaluated by the de Boor algorithm. The procedure permits to respond to a set of required requirements guaranteeing the global continuity of the displacement and motion functions.

The paper analyzes the incidence of the selection of the parameters of the curve –degree n, continuity C, number of knots and their multiplicity m– on its degree of freedom and shape. It also establishes the convenience of selecting them from the point of view of the specific application of the above mentioned functions and allows to use a reduced number of parameters and usually simplify the optimization procedure.

In order to make easy the design and use of motion function, the proposed design procedure includes the algorithms for the derivation and integration of the B-splines used; disposing of these algorithms permits, for example, to design the velocity function first and then obtain the displacement and the acceleration functions.

Three case studies have been presented in which the displacement and motion functions obtained satisfy the established design requirements and which validate the correctness of the proposed procedure, and the importance of the analysis of the curve parameters to be use.

The proposed procedure has advantages with respect to definition of the displacement and motion functions using traditional functions and cubic splines, because: i) the function is defined by pass points and the order of the derivatives in them is equal to the desired continuity C; ii) straight segments can be easily defined; iii) it is possible to choose the number of degree of freedom –free choice parameters– that allow the shape of the function to be adjusted to comply with the imposed specifications.

The possibility of locally modifying –local control– the motion function by introducing middle knots in a B-spline curve, shows the greater freedom of modifying the shape of such function using B-spline instead of using Bézier curves.

Nomenclature

m	: multiplicity of a knot inside of knots vector $\boldsymbol{u}$
n	: degree of the B-spline
n_0	: minimum degree of a B-spline without middle knots
$\boldsymbol{u}$	: knots vector of a B-spline
$\boldsymbol{u}'$	: derivative of the knots vector of a B-spline
$'\boldsymbol{u}$	: integral of the knots vector of a B-spline
u_i	: knot i of a knots vector of a B-spline
C	: continuity of the motion function
d_i	: Greville's ordinates of a B-spline
d_i'	: derivative of the Greville's ordinates of a B-spline
$'d_i$	: integral of the Greville's ordinates of a B-spline
d_i^k	: intermediate points in the de Boor algorithm
$d(u)$	: value of a point on the B-spline curve corresponding to the parameter u
$\boldsymbol{d}_i$	: control points of the polygon of control
k	: level of the de Boor algorithm; thickness head of the screw
L	: polynomial segments of a B-spline
l_{norm}	: normal length of the screw
l_{tot}	: total length of the screw
l_{thread}	: thread length of the screw
$s(\theta)$	: displacement function with respect to the cam rotation
$s(t)$	: displacement function with respect to the time
s_0	: initial position of the bar with respect to the head stock of the lathe
t	: time

t_{cycle} : cycle duration of the screw machining process
$v(t)$: velocity function
$a(t)$: acceleration function
R_r : follower radius
R_b : base circle radius of the cam
R_p : prime circle radius of the cam
ξ_i : Greville's abscissae
ξ_i' : derivative of the Greville's abscissae
$\acute{\xi}_i$: integral of the Greville's abscissae
θ : cam rotation angle
ω_{cam} : cam angular velocity

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