Building a conceptual model of the chain harmonic drive

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Abstract. The harmonic drive is a relatively new type of mechanical drives which application increases continuously in different fields of the industry. The rapid growth of application is driven by a number of advantages in comparison to the conventional drives such as gear and belt drives. The effective use of the harmonic drive, however, is limited because of difficulties in designing of such drives. One of the problems is the non-optimal typical design of the harmonic drive that can be improved in different ways. They include designing the harmonic drives in the framework of the closed roller chain using different modelling approaches. The mathematical modelling on the basis of CAD/CAE software appears to be the most effective of them. This article presents the method for building an alternate design of the harmonic drive based on the standard roller chain using commercial software «Siemens NX». Implementation of this method involves the CAD-modelling and multibody simulation to obtain the final geometry of the drive.

1. Introduction

The harmonic drive is a compact transmission mechanism with a high reduction ratio introduced in 1955 by American inventor C. Walton Musser mainly for use in the aerospace industry [1,2]. The main elements of its standard design are an elliptic wave generator, flexible gear and rigid gear. The wave generator is composed with a rigid elliptic race ball bearing and a thin-walled flexible outer race (Fig. 1a). The flexible gear looks like a thin-walled flexible cup with external teeth on its rim. The rigid gear is a thick-walled metal ring with internal teeth.

During the transmission operation the wave generator resiliently deforms the flexible gear along the major axis of the ellipse. As a result, the teeth of the flexible gear mesh with the teeth of the rigid gear in two areas at both ends of the major ellipse axis. By contrast, the meshing along the minor axis of the
ellipse is not engaged (Fig. 1b). During rotation of the generator the meshing areas are rotating with it. Difference between teeth number of rigid and flexible gear (which typically equals two) leads to the relative movement of these gears in a way that the flexible gear is rotated with respect to the rigid gear through the angle equivalent to two teeth after each complete revolution of the generator [3].

The harmonic drives in comparison to the conventional gear drives exhibit many advantages including: capacity to transfer high torques, compact size and low weight, zero backlash, high reduction ratio and high efficiency factor. Their disadvantages are relatively low rotational stiffness, position error (which is small but ubiquitous), resonance vibrations and nonlinear behaviour of the drive [4].

Application of harmonic drives in various practical areas is rapidly increasing. At present, they are used in automobile and aerospace industries, medical equipment, machine tool and robot building. The research on harmonic drives is on-going for many years and is maintained in different directions. One of these directions is the development of alternative designs for harmonic drives which uses various variants of the driving roller chain instead of the flexible gear [5-7].

Such designs benefit higher load capability and an extended range of reduction ratio (due to the offset of its lower limit) comparing to classical harmonic drive [8]. At the same time they are characterized by a more complicated structure which requires the geometry synthesis to provide proper kinematic relations for elements of transmission. Due to the complicated kinematics of the chain harmonic drive this synthesis can be a serious problem demanding development of an appropriate mathematical model. This model can be obtained by different ways. But the use of modern CAD/CAE software appears to be the most effective.

2. Object and tools for modelling
The aforementioned approach can be illustrated by an example of conceptual model building for a harmonic drive in which the standard roller chain is used as a flexible gear. The main components of this drive are rigid gear, driving roller chain (that are composed of external and internal links with different geometry), driven disk and input and output shaft. The interaction of these components is considered on the basis of their representation as a multibody system (MBS). In other word, the system of rigid bodies is interlinked by the non-ideal joints and is subject to consistent movements under the influence of internal and external loads [9]. The modelling is performed in the environment of commercial software «Siemens NX» using its CAD module for solid modelling and CAE module for simulation of kinematics and dynamics of MBS (motion simulation).

3. Modelling method
The modelling of MBS using CAE tools includes the execution of a consequent number of stages. First of them is the building of an assembly model for considered system (mechanism). Therefore the modelling process in the present case starts with the building of an assembly model of the harmonic drive, including parametric solid models of all required components.

Its starting point is the preparation of a parametric model for rigid gear. Because of the specifics of proposed harmonic drive the teeth of this gear must have a round profile defined by two parameters in the model database. These parameters are a nominal chain pitch (P) and a gear teeth number (Z). The further modelling sequence is performed using the pitch equalling to 12.7 mm and the even number of teeth equals 34 pcs. The input of these values in «Siemens NX» is performed using a special program window «Expressions».

The introduced parameters allow defining the main dimensions of gear by either analytical or graphical method. The most convenient and simple method in this case is graphical. The realization involves the sketch creation with the original geometry as shown in Fig.2. This geometry is represented by the set of standard geometrical primitives coupled with proper geometrical and dimensional constraints. These constraints imposed on sketch geometry in strict compliance with Fig.2 uniquely define the addendum gear radius Ra and tooth radius Rr (roller radius of the chain) that are required for building other components of the drive.

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Figure 2. Building a profile of the rigid gear

The obtained values of these dimensions are saved into corresponding parameters Ra and Rr for the convenience of their further use. In «Siemens NX» this is performed through interactive measuring of current dimensions of the existing geometrical element. This function is also available in the aforementioned window «Expressions». In this example the rounded values of parameters Ra and Rr determined in this way equal 68.821 and 3.769 mm, correspondingly.

Then on the basis of generated interdependent geometry the full profile of gear rim is formed as a continuous closed curve using corresponding «Siemens NX» tools. This curve along with an added circle that outlines the external contour of gear is used for building of its solid model through extrusion operation.

At the next stage of assembly building the geometrical model of elliptic cam (wave generator) is formed. This is achieved with creation of the parametric links of the previously generated gear model, i.e. the parameters P, Ra, Rr defined in the model database and their corresponding values are assigned to the values of identical parameters in the gear model. This procedure is performed using program tools of «Siemens NX» that are available in its window «Expressions». In addition, another (integer) parameter (n) is introduced to model for setting the number of rollers in the chain (teeth number of the flexible gear). In this example the number of rollers is chosen to be 32 pcs.

The required dimensions of the elliptical profile for the cam are determined on the basis of the following conditions:

- The major semiaxis of ellipse must be equal to the difference between the addendum radius of the rigid gear (parameter Ra) and the roller radius (parameter Rr).
- The minor semiaxis of ellipse shall be such that the ellipse perimeter is equal to the perimeter of the circle circumscribed around the regular polygon (with n number of sides) obtained by the equidistant offset (inside) of a polygon with the size of side equal to the chain pitch (parameter P).

Exact analytical calculation of minor semiaxis size in the environment of «Siemens NX» appears to be extremely difficult because it requires the use of elliptical integral. Moreover, the existing approximation formulas, as shown by the preliminary calculations, provide the significant error of the cam dimensions. Therefore, the required elliptical profile of the cam is obtained by the graphical method.

For that purpose the sketch containing the typical elliptical curve with position and orientation determined by the construction geometry is build. The dimension equal to the difference of radiuses (Ra – Rr) is assigned to the major semiaxis. The dimension of the minor semiaxis is determined through the building of auxiliary circles with the radius Rr interlinked by the dimensional and geometrical constraints and tangent to the elliptical curve. The example of sketch obtained in such way is presented in Fig.3.

The required dimensions of the major and minor semiaxes of ellipse obtained in this example as a result of the described scheme equals 56.845 mm and 65.052 mm, respectively. After the dimensions of ellipse have been determined it is converted to the solid model of the cam using symmetrical extrusion (the sketch remains in the plane of symmetry of the cam). The model is finalized with addition of the geometrical features in the form of the cylindrical mounting bore and chamfers on the circular and elliptical edges. Then the completed model is saved to file.
Further the parametric models of external and internal chain link are assembled to serve as subassemblies in the model of the harmonic drive (Fig. 4). Here the models of an external plate, pins (short and long) and short roller are included in the model of external link. At the same time the models of internal plate, bush and long roller are included in the model of the internal link. The presented approach allows associating the characteristic dimensions of the specified components with previously prepared drive components through the parametric dependencies.

The motion transfer from the chain elements to the output shaft in this variant of the mechanism is performed through the driven disk rigidly connected to the output shaft. Therefore, the geometric model of such disk is prepared as the next component in the assembly model of the harmonic drive. The disk geometry obtained at this stage is preliminary and represents the actual disk with a cylindrical mounting bore. This is because the complete geometric configuration of the driven disk model is synthesized using the results obtained from motion simulation.

At the final stage of assembly model preparation, the geometric models of input and output shafts, drive housing and housing cover are being built. The shaft models are composed with identical smooth cylinders, the first of which (bundled with the cam) defines the input link and the second (bundled with the driven disk) defines the output link in the prospective model for motion simulation. The models of housing and cover that serves as shaft supports are performed arbitrary with respect to correct arrangement of other drive components.

The obtained geometric models of drive components are used for building its assembly model. The assembly file is created in «Siemens NX» following activation of the relevant program environment with sequential addition of the model components. The required relative position of the components is enforced by the proper assembly constraints. The most time consuming operation in this case is formation of the chain model consisting of separate links and closed around the elliptical cam.

The complete assembly model is then converted to model for motion simulation (multibody system). Relevant CAE-module – module «Motion Simulation» of «Siemens NX» is used. The conversion starts with the manual defining the movable and fixed links (rigid bodies). The single fixed (grounded) link includes the housing, its cover and gear. All other model components are used to define the movable links in the quantity of 82 items. From this number the 16 links are defined by the pair of external chain plates connected by the pair of bushes, 48 links are defined by the long and short rollers, one link is defined by the cam and the input shaft and one link is defined by the disk and the output shaft. It should be noted that the links, including the cam and long rollers also contain the sketches of these parts (elliptical and circular profile) what is needed for further formation of kinematic joints.
After definition of all models links the formation of standard joints between the links is performed. In this case all standard joints in the model are represented by the rotational joints, which are created:

- Between the pair of links, the first of which includes the input shaft and the second includes the housing.
- Between the pair of links, the first of which includes the output shaft and the second includes the housing cover.
- Between the pair of links, the first of which includes the bush and the second includes the pin.
- Between the pair of links, the first of which includes the long roller and the second is defined by the short roller.

The total number of rotational joints created in the model with the use of specified parameters accounts for 82 pcs.

The interaction between the components of the roller chain and the cam is realized with a set of non-standard joints «curve on curve» which represent the kinematic pair simulating the cam mechanism with the roller follower. The curve direction in each pair is determined with the elliptical profile (initial sketch) in the middle plane of the cam and with the circular profile in the middle plane of the long roller.

Contact pairs between rollers links and the gear link provides interaction between the chain components and the gear. The direct contact between the chain components and driven disk at this modelling stage is not defined.

After the creation of all described joints the synthesis of the final geometry for driven disk is produced (profiling the slots for short rollers). Two drives are defined in the current model: the first at the rotational joint between the housing and the input shaft; and, the second, at the rotational joint between the output shaft and the housing cover. In the first joint the constant angular velocity is specified by the arbitrary value (100 rpm for this example). In the second joint the angular velocity is specified by the product of the velocity in the first joint and the nominal reduction ratio of harmonic drive that is calculated through the given number of gear teeth and the number of long rollers in the chain ([32–34]/32). Accordingly, the speed calculated for the second joint in this example is equal minus 6.25 rpm.

On the basis of obtained model including described joints and drives the kinematic analysis of mechanism is performed and its results are visualized. It is clear that in the present case the required geometry of the driven disk will be defined by the motion path of the axis of short roller relative to rotating disk. To define this motion path, the tracing of specific points is performed during the animation of the results. These points are the centres of short rollers end faces located near the disk model. In this representation the tracing base is disk connected to the output shaft.

Executing the tracing procedure results in arrays of points represents the position of the specific roller relative to the disk at any given solution step. An example of the tracing results for arbitrary chosen roller is presented in Fig. 5a. As shown in the figure the locus of obtained points is with a high degree of confidence a closed elliptic curve.

Therefore the required path is obtained by LSQ fitting of traced points with the ellipse function that available in «Siemens NX». In order to increase fitting accuracy, all traced sets of points (16 sets) are combined to a single set performing consecutive angular displacement to the arbitrary chosen fixed position.

![Figure 5. Defining the profile of slot: (a) traced points; (b) fitted curves](image)

The fitted elliptical curve is then used for generating an equidistant curve with dimensions defined by the radius of short roller that is applied as normal offset to the points of the original curve. The fitted and
equidistant curves generated in this example are presented in Fig.5b. The former curve, in fact, represents the required profile of the slot, which provides the meshing of chain elements with the driven disk.

Further, this profile is used to create the circular array of elliptical through-holes in the solid geometry of the driven disk. After this the geometry of driven disk is rebuilt to its final variant as shown in Fig.6a.

![Figure 6](image)

**Figure 6.** Modelling and simulation results: (a) assembly model of chain harmonic drive; (b) variation of rotational velocity for input and output shaft

Validation of the model is performed with calculation of angular velocity of output shaft as a function of velocity of the input shaft. In this process of verification the previously prepared model is modified as follows:

- Drive previously created at the rotational joint between the output shaft and housing is deleted.
- 3D contact pairs are introduced between the driven disk and short rollers with the actual contact interface defined by the cylindrical faces of rollers and elliptical faces of slots in the disk.

The calculated results obtained using the modified simulation model are presented as a diagram in Fig.6b.

### 4. Simulation results

The presented diagram shows the variation of calculated rotational velocity for the output shaft depending on the rotational velocity of the input shaft. As is seen, the calculated velocity value varies around the same constant level due to characteristics of the model for contact interaction. The line fit of the velocity values obtained at the each solution step allows determination of this constant level as negative 6.065 rpm. Therefore, when the nominal velocity of the output shaft equals 6.25 rpm, the averaged simulation error is about 3%. This validates the developed model and the possibility of its use for the advanced analysis of kinematic parameters in the chain harmonic drive.

### 5. Conclusions

The obtained simulation results allow making the following conclusions:

- Design of the considered chain harmonic drive is characterized by the complicated kinematics with parameters defined by the special geometry of drive components.
- Parametric solid modelling using CAD software allows automation of basic drive dimensions determination without engaging the time-consuming computations and to prepare the foundation for subsequent computer aided analysis.
- Motion simulation of the drive with use of CAE tools allows synthesis the drive geometry where shape and dimensions are determined by relative motion of drive components.
- Simulation model obtained with the CAE software in multibody approximation allows estimating the kinematic and dynamic parameters of the drive.
• Motion simulation results can be employed for the subsequent analysis of stress-strain state of the drive components using finite element method such as, for instance, presented in the references [2,10-14].

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