

# **Functional design and kinematic synthesis of a chain driven high-speed packaging machine.**

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## **Abstract.**

In this paper we show a design procedure for the main layout of a new concept of packaging machine based on a chain drive. The introduction of this type of drive unit allows a significant increase of the capacity of the machine, in comparison with the traditional cam driven mechanisms nowadays in use, but involves some additional complex kinematic and dynamic behaviours that must be carefully analyzed and understood in order to obtain the full functionality of the final design and to avoid unexpected effects during operation. For this reason, starting from basic geometric and functional parameters easily derived from the main specifications, all the steps involved in the design process are performed, without specific reference to any defined geometry of the bodies, using ADAMS kinematic parametric models. This allows a great flexibility and, during the initial stages of the design process, gives the possibility to quickly loop several times over different possible solutions simply changing some user defined parameters without any effort spent to change already existing complex CAD models. Once a kinematically satisfactory scheme has been found, all the relevant information is transferred to the CAD system where the actual parts and assemblies are created using the guidelines defined during the kinematic synthesis. It is then possible to estimate the relevant inertia properties that can be transferred in ADAMS to perform a fully dynamic analysis, allowing both the calculation of the forces acting on the various parts, which can then be used for finite element analyses, and the verification of the system actual performances.

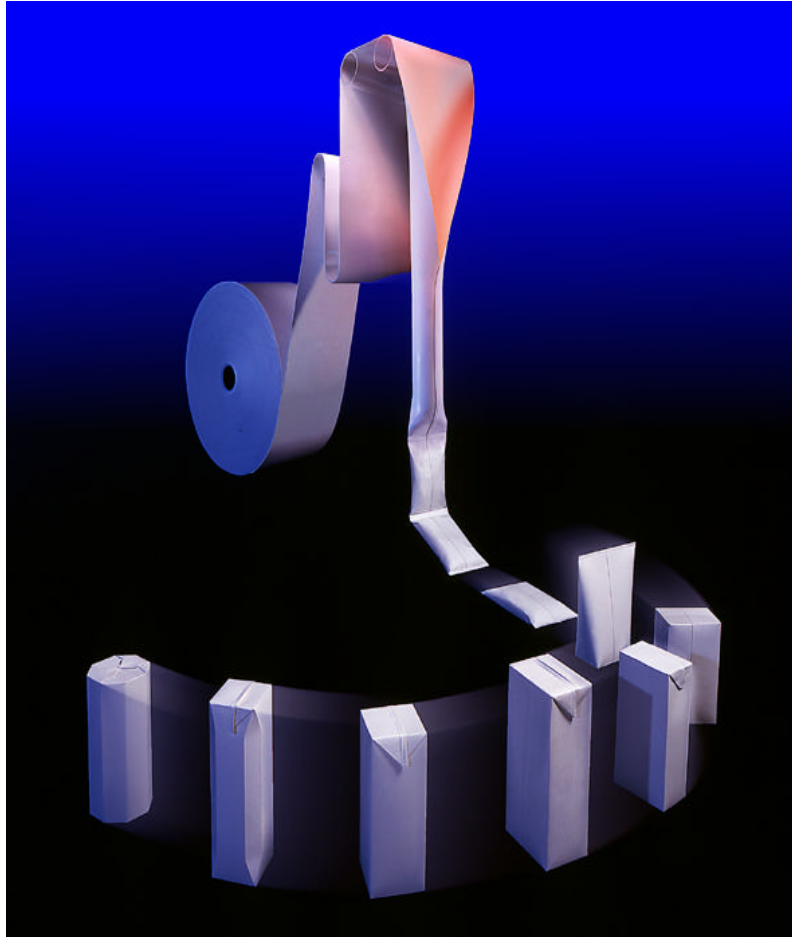
## **1. Introduction.**

Tetra Pak develops, manufactures and markets systems for processing, packaging and distribution of liquid food. Business Unit Tetra Brik is a part of the Carton Division of the Tetra Pak Group and is responsible for the development, production, marketing and support of the Tetra Brik Packaging Systems. The systems include packaging material, package specifications, filling machine and distribution equipment for the filling line.

The fundamental idea behind this type of packaging systems is to form a tube from a roll of plastic-coated paper, fill it with the liquid product and seal it below the liquid level. The entire process is continuous and takes place in a single machine, which both forms and fills the package. This process is schematically shown in fig.1. The main advantages of this roll/tube concept are:

- a. the space savings before and after filling,
- b. it is possible to sterilize the whole surface of the packaging material,

- c. the resulting simple filling system gives high hygiene,
- d. the packages are fully filled ensuring high quality product and good distribution properties.



**Fig. 1 The roll/tube concept.**

It is also possible to see in fig.1 that from the same concept not only different volumes but also different shapes of packages can be obtained.

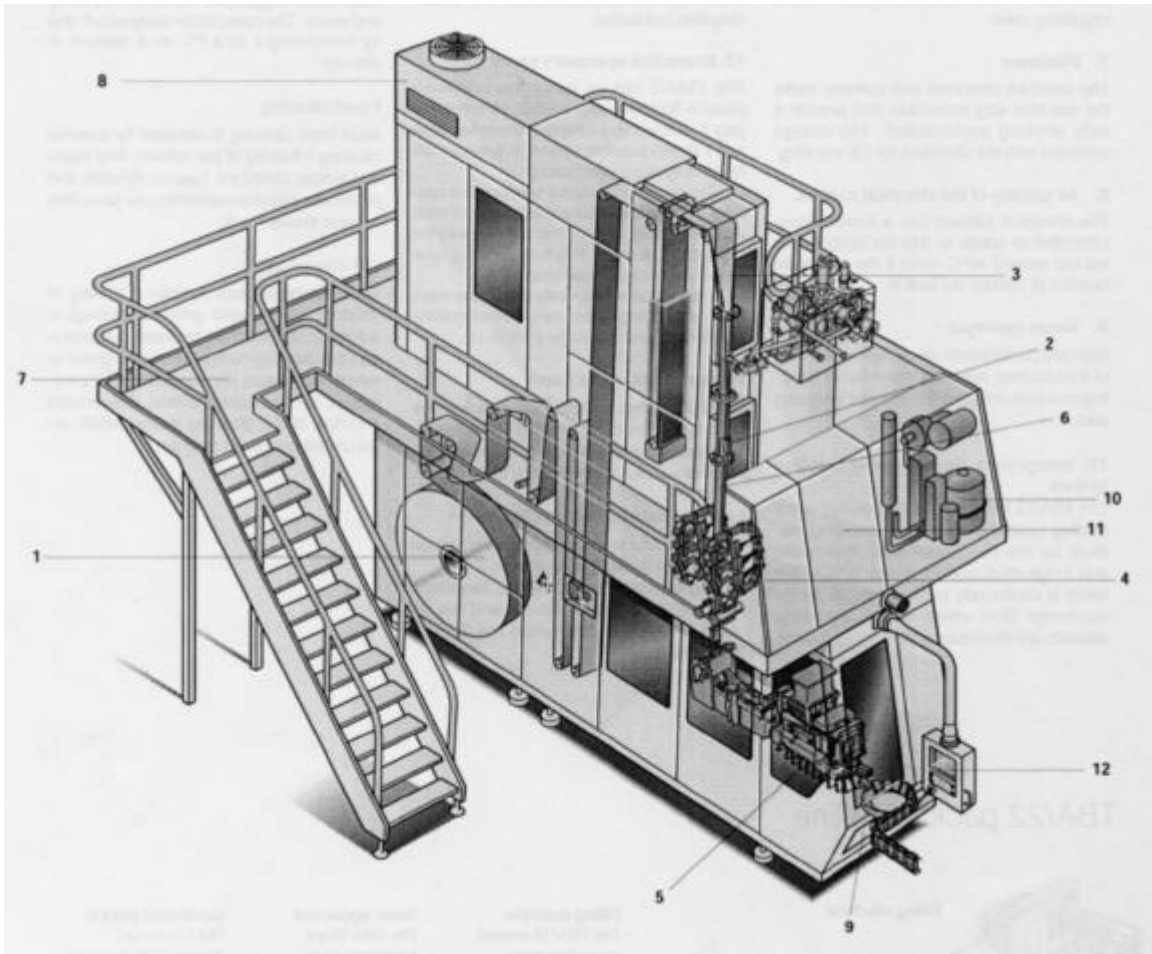
### **1.1 The High Capacity TBA/22 Filling Machine.**

TBA/22 is a new system inside the Tetra Brik Aseptic portfolio and has been designed with the specific purpose of reducing the costs and raising the productivity of the producers. It achieves a capacity of 20000 packages per hour, which is about three times the capacity of a traditional Tetra Brik filling machine. The relevant machine functions are described with the help of fig.2.

1. Packaging material supply. Two reels of packaging material are placed in an integrated section of the machine. The reels are spliced automatically by induction heat. This solution enables the next reel to be spliced before the one currently in use is finished.
2. Longitudinal sealing. Induction heat is used for sealing the longitudinal seam of the packages.

3. Aseptic system. Sterilization of the packaging material is achieved by letting the packaging material pass through a deep bath of hydrogen peroxide. Rollers remove the hydrogen peroxide from the packaging material and residues are evaporated by hot sterile air.

4. Jaw unit. The jaw unit consists of two chains opposite each other driven at constant speed by a servo-motor. Each chain is made of 10 links and these links move against fixed cams. In the jaw system a semi-finished filled package (with a pouch-like shape) is created and cut out from the tube.



**Fig. 2 The TBA/22 filling machine.**

5. Single final folder. After the semi-finished package has been created from the tube in the jaw system, it is folded into its final shape in the final folder. This consists of a single chain driven at constant speed by a servo-motor. During this stage the flaps of the semi-finished package are folded and glued on the side panels.

6. Product filling system. The product filling system is equipped with a level probe that senses and controls the liquid level by a regulating valve connected to the electronic control system.

7. Platform

8. Electrical cabinet with air cooler.

9. Waste conveyer.

10. Integrated closed water cooling system.

11. Service unit.

12. Interactive operator's panel.

During the concept development of TBA/22 several moving parts of the machine have been studied and analyzed with ADAMS, however, for sake of brevity we concentrate in this paper on the kinematic synthesis of the jaw system only.

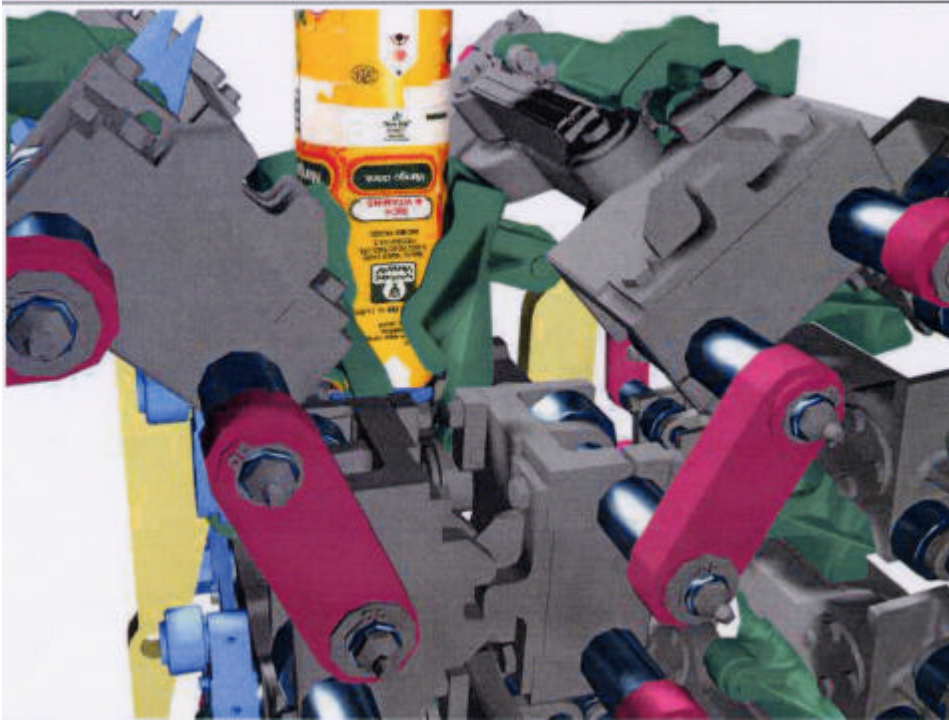
### **1.2 Description of the jaw system.**

The main function of the jaw system in every type of Tetra Brik machines is to form a semi finished package from the packaging material tube filled with the product. In traditional systems the jaw system is a cam driven mechanism made of two moving jaws in alternative motion. The new concept introduced in TBA/22 is to create the jaw movement by a chain carrying a large number of jaws (links) that are moving on predefined paths. Let us now examine the main function achieved by the resulting chain driven mechanism. In fig.3 we present an overview of the final assembly with the two chains carrying the links and the tube of packaging material. The links on one side (left in fig.3) are called sealing links, while the ones on the other side are called pressure links. The reason for these denominations will be clear in a moment.



**Fig. 3 The TBA/22 jaw system.**

The packages are formed at the top of the jaw system. In fig. 4 the detail of the forming phase is shown. It is seen that the packages are formed by the jaws (grey) and by the volume flaps (green).



**Fig. 4 Semi finished package forming inside the jaw system.**

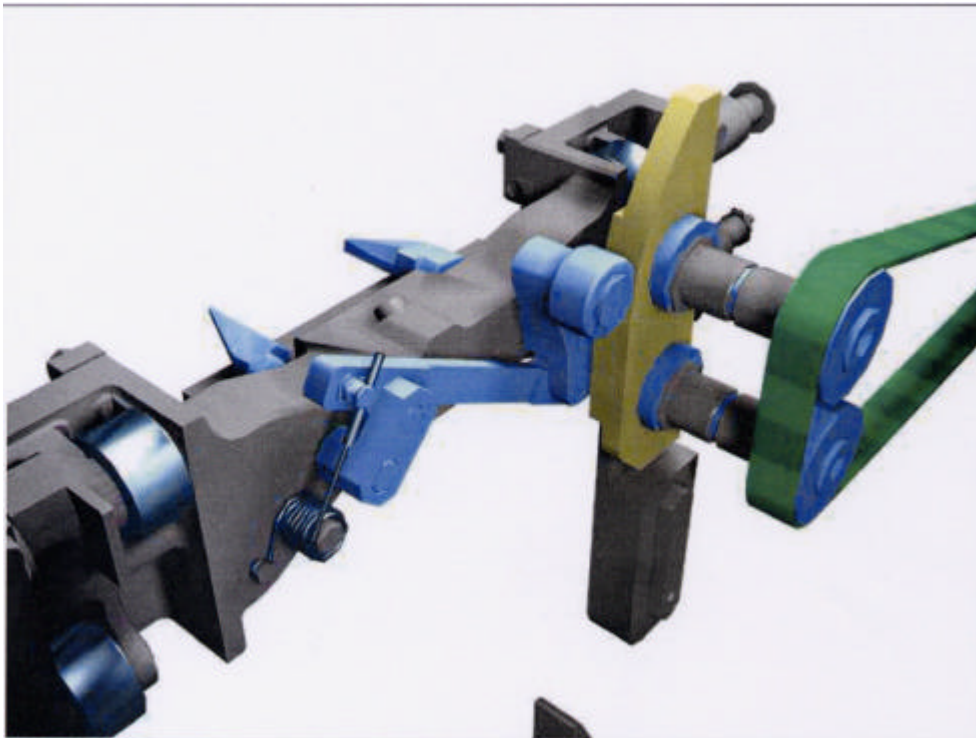
After the jaws are closed together a special mechanism folds the flaps in the bottom of the package in order to keep the printed image on the package synchronized with the overall motion (design correction). This is shown in fig.5 where the end fingers of this mechanism are shown in blue.



**Fig. 5 The printed image is kept in position by the folding flaps (blue).**

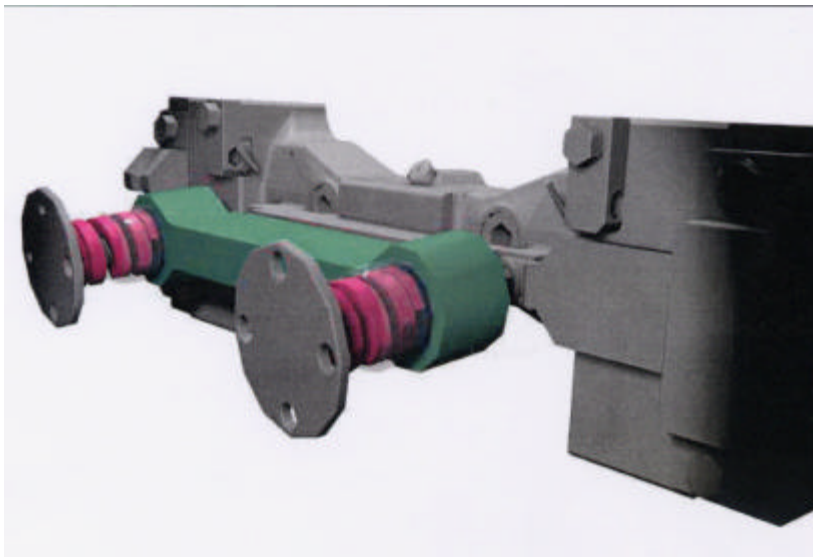
This mechanism is governed by a control system based on optical sensing. This mechanism is responsible for design correction and is called folding flap mechanism.

The complete mechanism is shown in fig.6. The stroke of the folding flaps (fingers) is set by a cam (yellow) which is controlled by a servo-motor. The servo-motor affects the cam via a belt (green) and two eccentric shafts (blue, grey).



**Fig.6 The folding flap mechanism.**

After the jaws are closed and before the folding flaps have started to adjust the package relative position, the jaw are pressed together in order to apply the so-called jaw pressure needed to seal transversally the packages. This jaw pressure is applied by two springs (violet in fig.7) in each pressure jaw. This is shown in fig.7.



**Fig. 7 Jaw pressure applied by spring loading.**

In fig.8 it is shown the package configuration inside the jaw system during the phase of jaw pressure applied. This figure is obtained by virtually opening the jaw system during the motion.

The sequence of operations in this phase is: jaw pressure application, design correction, sealing pulse, and cooling time.



**Fig. 8 Jaw pressure applied.**

Once the jaw pressure is fully applied the sealing pulse starts. To seal transversally the packages we use an induction heating system mounted on each sealing link. This induction system generates an eddy current distribution on an aluminum layer inside the packaging material structure. The aluminum heats up due to Joule effect and transfers heat to the adjacent polyethylene (PE) layers that reach their melting temperature. Due to the applied pressure there is a mixing between the melted PE layers corresponding to the two open opposite sides of the tube pressed together. After the sealing pulse is terminated there is a cooling time that enables the PE to re-solidify as a single layer. This guarantees a strong and tight sealing of the packages and is one of the most advanced and developed technologies in our machines. The jaw must be closed, i.e. the jaw pressure must be kept, during a period long enough to allow a suitable sealing and cooling time, in order to ensure a tight closure of the package. The length of this process (together with the one of the sterilization process) puts physical limits on the capacity of the machine.

After the end of the cooling time a cutting mechanism carried by the pressure link cuts the package in correspondence of the transversal sealing area. The package then falls down in a conveyor to enter the final folder.

In the following we will examine the steps involved in the functional design and synthesis of all the mechanisms and motions involved in the package-forming phase.

## **2. Kinematic synthesis of the main cams.**

The elementary mechanics of chain drives has been known for many decades. There are however some subtle characteristics that require complex analyses to be fully understood due to the advanced mechanics involved. The conventional roller chain drive consists of two sprockets and a chain belt. The chain belt is made up of discrete links. The motion transmission is due to the geometry of the connection and there is no relative sliding. The chain axis, i.e. the line connecting the centers of the axles, is made of a regular polygon inscribed inside the primitive circle of the sprocket. The main consequences of this are:

- a. The axles connecting the links of the chain are lifted and lowered subsequently (polygon effect) during the motion.
- b. The absolute speed and acceleration of the axles vary during the motion. These fluctuations decrease rapidly as the number of teeth of the sprocket increases.
- c. If the two sprockets (drive and driven) have different diameter the transmission ratio varies cyclically. Furthermore the length of the chain span varies discontinuously with time.
- d. The teeth of the sprockets and the axles of the links have different velocities (mainly in direction) when they come in contact. This causes impacts and energy losses that are proportional to the square of the angular velocity of the sprocket.

The chain drive of TBA/22 shows further sources of complexity. First of all it is not made of two sprockets and the chain but, since we need to strictly follow the motion of the links during the forming, sealing and cutting phases, the motion is controlled by a series of cams. Furthermore the chain is not made of simple links but each link is in reality the assembly of a number of sub-systems resulting in a considerable final mass with complex inertia distribution. This forces the minimization of the velocity fluctuations during the motion in order to reduce noise and to avoid unexpected behaviours in operative conditions and the introduction of a specific spring-like chain-tensioning device to reduce possible problems due to inertial effects.

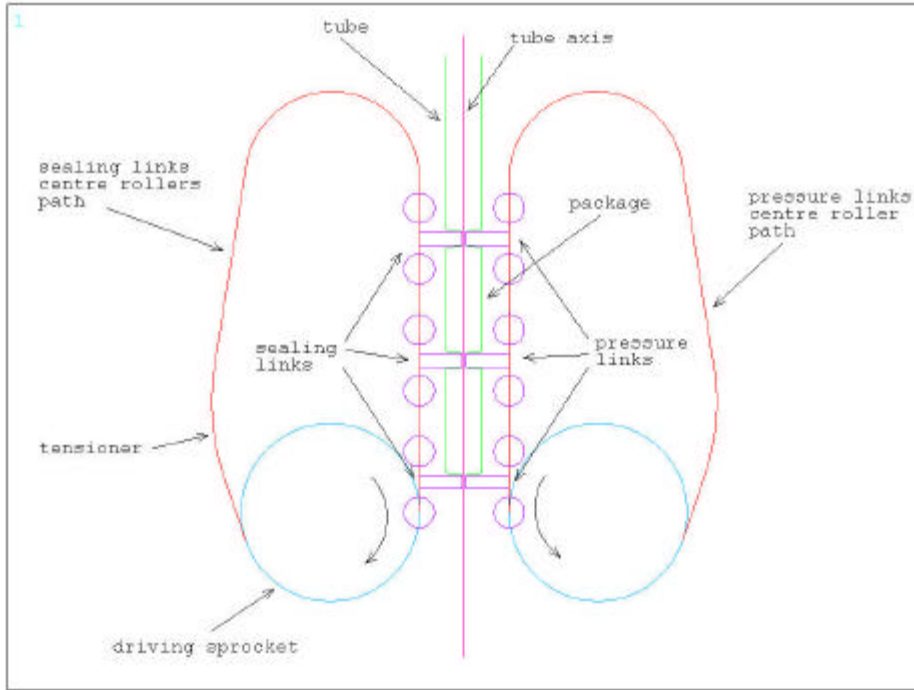
### **2.1 Basic curves definition.**

The starting point for the design of the jaw unit is a simple geometrical layout like the one shown in fig.9. The package dimensions and shape, the capacity of the machine and the sealing time mainly decide the dimensions defining this configuration. In particular the vertical path has to be long enough to allow a suitable sealing and cooling time under jaw pressure applied. External constraints are added by cost, simplicity and size considerations, all being strongly affected by the total number of links that should be minimized. Of course the total length of the path should fit with the length of the chain, small differences being absorbed by the tensioning device.

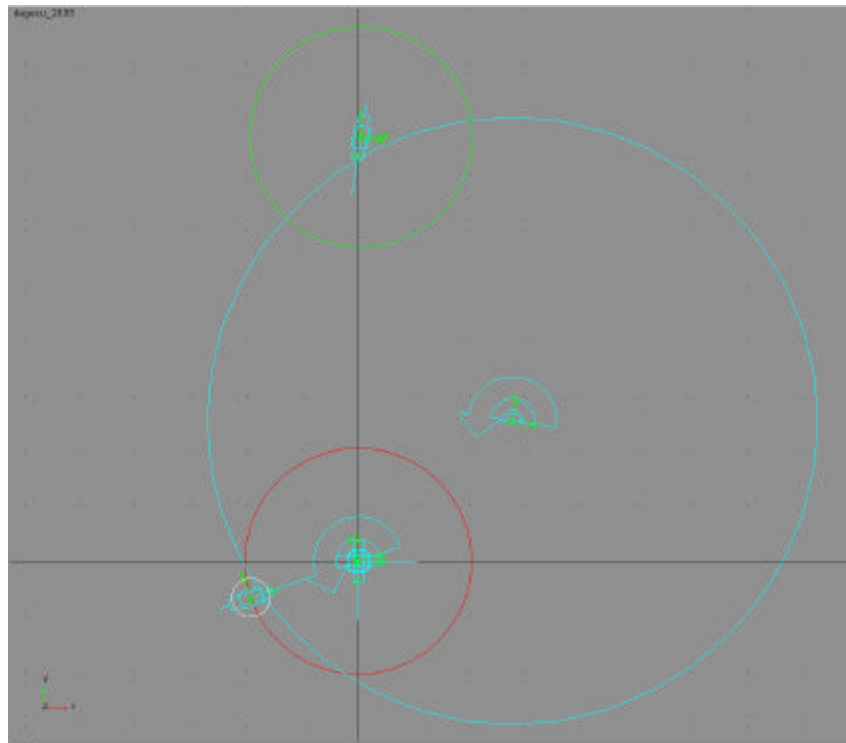
A parametric Adams kinematic model is used to generate the curves that define the basic path of the rollers carrying the links. The following parameters have to be specified as input for this model: the coordinates of the centers of the two sprocket primitive circles, the corresponding radiuses, and the coordinates of the center and the radius of the circle corresponding to the tensioning device. The Adams model is shown in fig.10 and is made of 6 parts (not including ground), 1 cylindrical, 1 revolute, 4 translational joints and 7 motions. The model is kinematically well defined and capable to give as result a curve describing the path of the roller of the links. This curve is shown in fig.11 and is used as



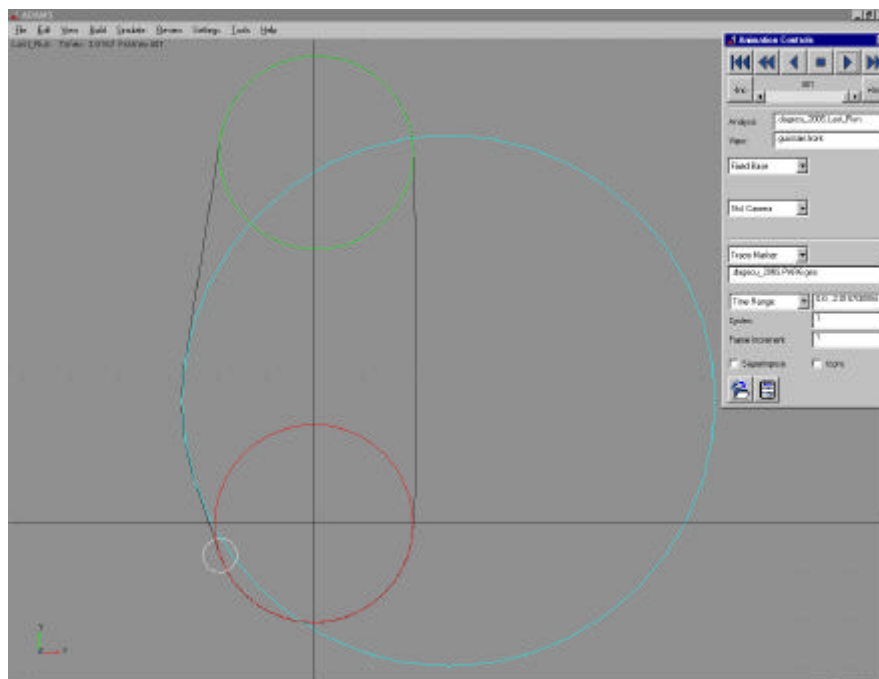
the basis for successive corrections needed to allow all the phases involved in the package forming process to be correctly executed.



**Fig.9 Model schematic.**

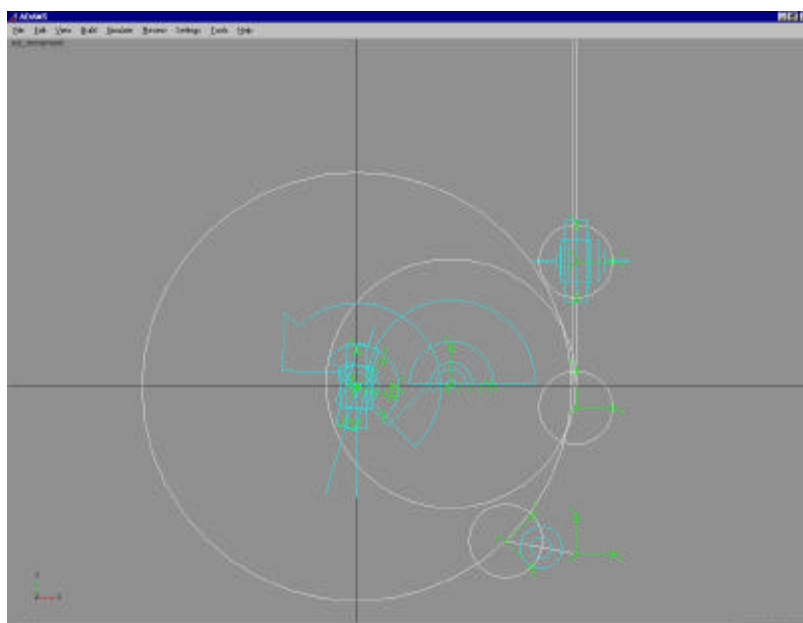


**Fig.10 Adams model for basic curve generation.**



**Fig.11 Basic curve generated by Adams.**

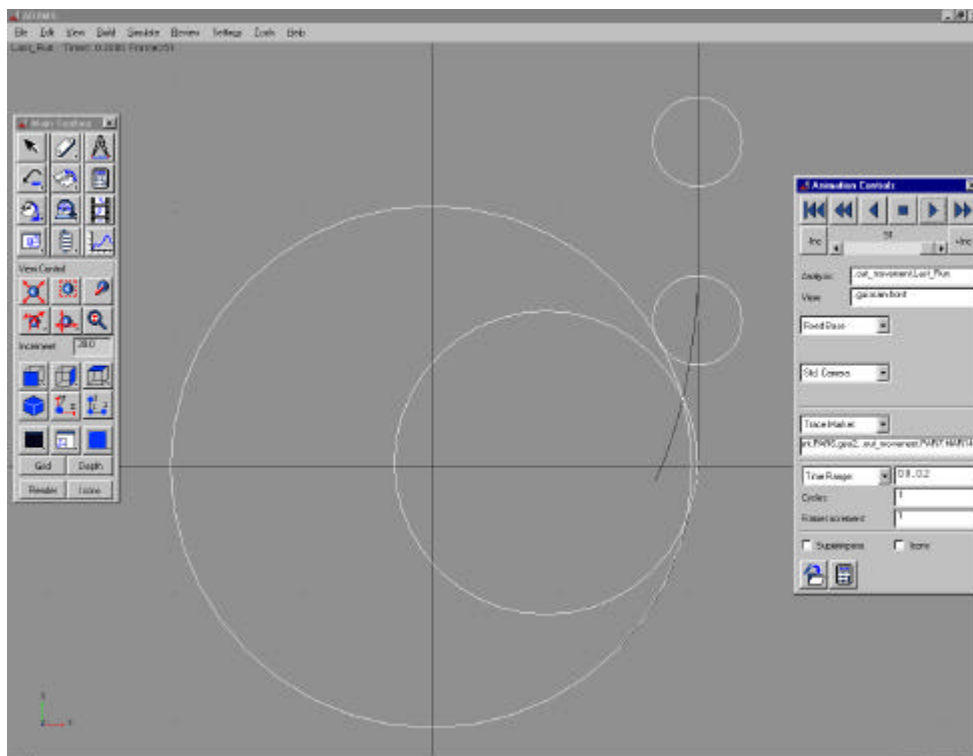
A second parametric Adams model is used to correct the basic curve to allow the link to release correctly the jaw pressure. We proceed in the synthesis process with inverse kinematic analyses and since at this stage we have only the knowledge of when and



**Fig.12 Model for jaw pressure release and out movement.**

where during the motion a certain phase is completed we are forced to drive the motion backwards. The model used is shown in fig.12. The model is made up of 6 moving parts, 1 cylindrical, 2 revolute, 1 spherical, 3 translational joints and 4 motions. Once more the model is kinematically well defined.

conveyor. This motion is fully determined by geometric clearances considerations and depends only on the chosen dimensions of the sprocket, of the rollers pitch, of the jaw pressure stroke (which in turn is defined by the chosen spring stiffness and the sealing pressure), and of course of the package at hand. The resulting trajectory of the roller centers is shown in fig. 13. It must be noted that while the previously defined basic curve was a single curve, now we divide the rollers into two sets: the first and the second of each link, and each set will follow a different path. Two different cams driving one of these sets each will indeed determine the final motion of the links and of the complete chain.

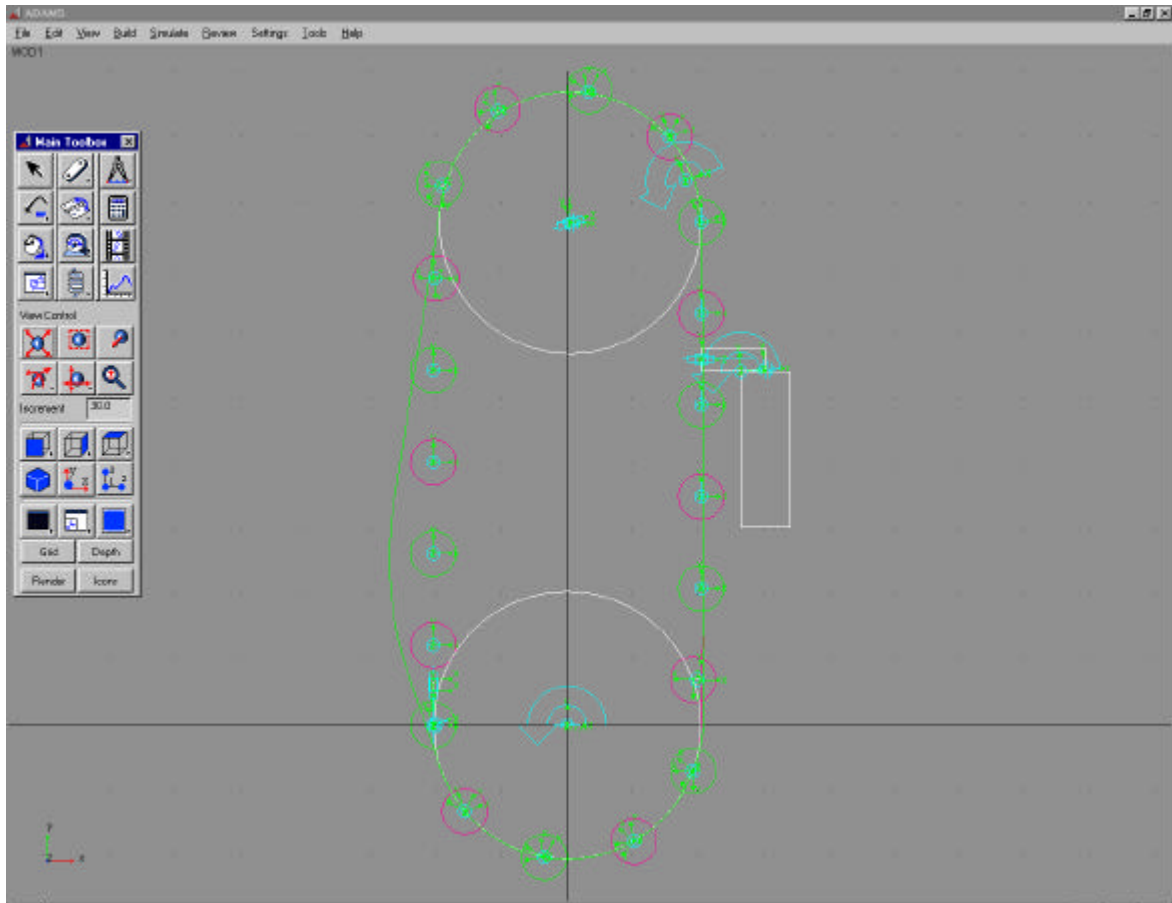


**Fig. 13 Jaw pressure release and out movement curve correction.**

Once the basic curve and the corresponding out movement correction have been generated, a special purpose written procedure creates the basic curves for roller sets 1 and 2 merging them together.

### **2.2 Jaw pressure application and in movement correction.**

A third Adams model is then used to implement the other dimensioning parameters defined by the package and link dimensions. Since the system (and the motion) is symmetric about the tube center axis, we can consider just one side, so only the pressure jaw chain is considered in the following. The model is shown in fig.14.

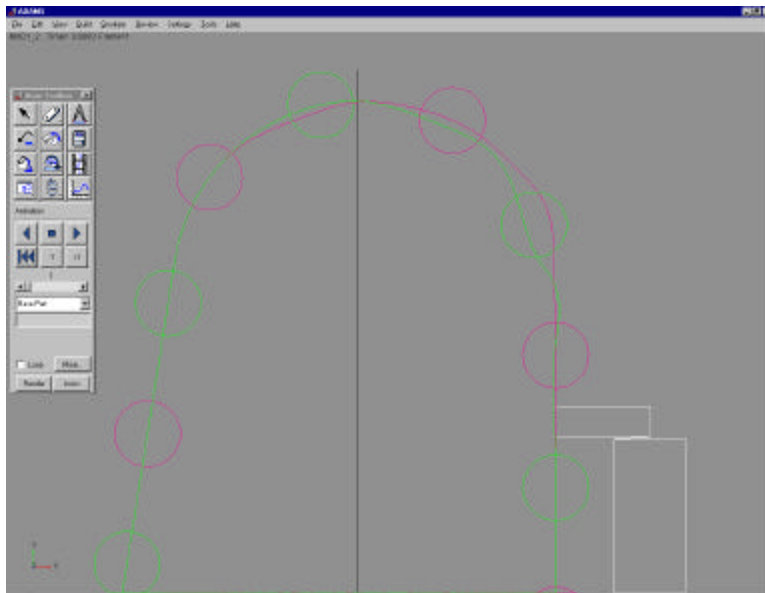


**Fig. 14 Chain model.**

The resulting curves from the previous step are introduced in this model and the corresponding rollers are connected to them by Point to Curve constraints (PTCV). The resulting model is made of 26 moving parts, 22 cylindrical, 1 revolute, 3 spherical and 3 translational joints, 9 motions and 15 PTCV. The main goal of this motion is to introduce into the exiting curves the correction needed to correctly perform the jaw pressure and in movement (as we call the motion during the forming of the package from the tube) phases. The points on the curves where the jaw pressure release starts have been identified in the previous step of the synthesis. From the needed sealing and cooling time we can calculate the length of the curve vertical path, identifying the points where the jaw pressure must be fully applied (as before we proceed backwards). From this point we disconnect a link from the curves and apply (backwards again) the appropriate motion for the jaw pressure application and the in movement. Simply a smooth polynomial motion on a translational joint gives the jaw pressure application. The in movement is instead a more complex motion to define. It is generated by polynomial interpolation and essentially defines the rotation of the package panels due to the action of the jaw carried by the links on the tube. The actual interaction simulation should include impact between a mechanical system (the jaw) and the filled tube. This is a far too complex problem to be handled at this stage (it involves contacts, nonlinear material behaviours and fluid-structure interactions), so we have developed a simpler kinematic model of the forming based on minimizing the deformation (measured as fiber length variations) of the

packaging material during the forming process. Despite its simplicity (neglecting most of the real complex physical interactions involved, but trying to keep the essentials), this model has proven to be quite effective and is almost always used during kinematic cam synthesis. Some finite element simulations have permitted the fine tuning of this model giving suitable values for velocities and accelerations (used as conditions for the polynomial interpolation of the motion) that are not creating damages or tears on the final package.

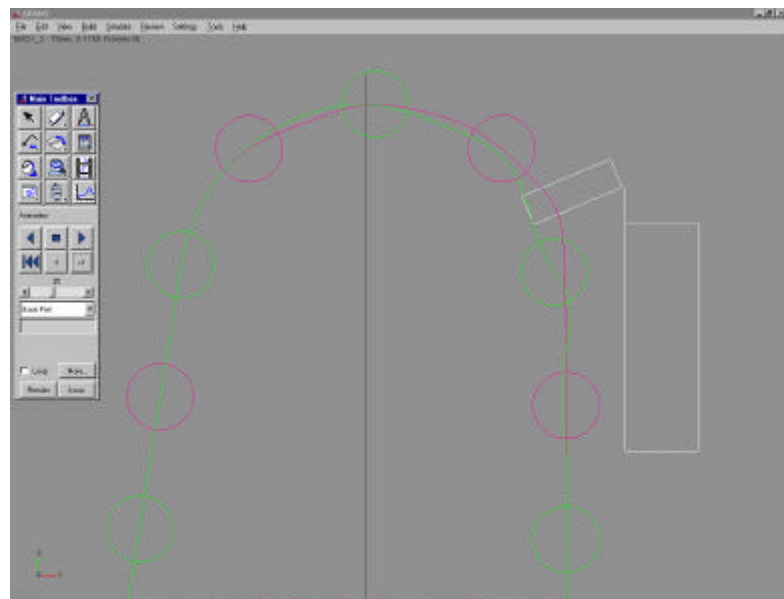
We show in fig.15 the curves resulting after the correction calculated by this model. The



curve and the rollers corresponding to the link set 1 are displayed in green and the ones to set 2 in magenta. The figure shows the configuration frame 1 that corresponds to the complete jaw pressure application; while in fig. 16 we show the last frame corresponding to the tube hitting (we are once more moving backwards).

**Fig. 15 Jaw pressure fully applied.**

It easily seen from these figures that the curves corresponding to the two sets of links have been strongly modified by imposing the correct motions to the links. This means that we have modified the initial layout and we now need to compensate for the chain length variation in order to minimize the polygon effect that can cause unwanted fluctuations and intermittency.



**Fig. 16 The jaw hits the tube.**

An additional Adams macro has been created that automatically calculates the chain length variation and minimizes the polygon effect adding a further correction on one of the curves. Of course this correction should not affect the already defined kinematics.

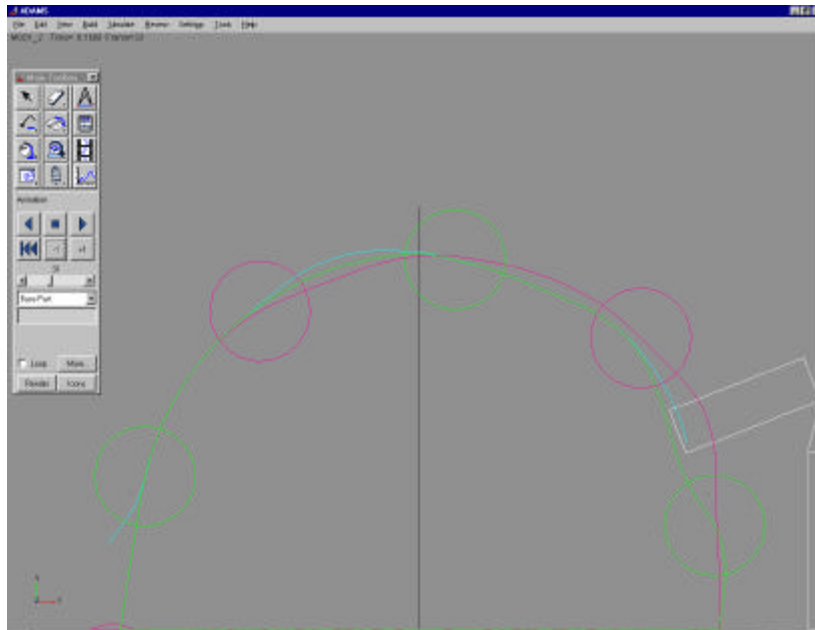


Fig.17 shows the calculated correction (cyan curve). This calculated curve is then trimmed and merged into the curve corresponding to link set 1 (green curve).

**Fig.17 Curve correction for polygon effect compensation.**

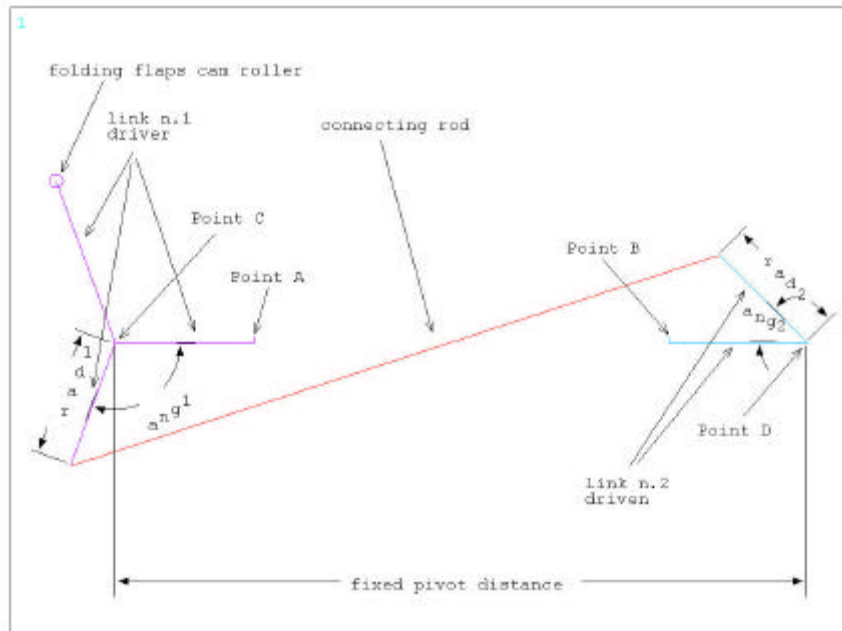
At this stage the kinematic synthesis of the main cams is completed since all the motions needed to form the package are accounted for in the chain path. With a kinematic (this time) forward analysis is then possible to verify the correct timing of the different phases. We will examine this after having completed the design of the mechanisms involved in the jaw systems other than the chain.

### **3. Kinematic synthesis of the design correction mechanism.**

The design correction or folding flap mechanism is responsible for the synchronization of the motion of the jaw system and the one of the packaging material. In addition it helps the forming by pre-stressing the flaps of the package that later, in the final folder, will be folded and glued on the side panels giving the package its final shape. The chosen mechanism for this task is a four bar linkage in the anti-parallelogram configuration, see fig.6. In fig.18 we show a planar schematic of this mechanism together with the relevant dimensional parameters. The points C and D represent pivot points and link1 and link2 rotate about axes, perpendicular to the plane of the drawing, passing through these points. A cam whose roller center is shown in fig.18 then drives Link1. Points A and B are located in the physical position where the fingers pull down the packaging material during the motion. Since the pivot distance is fixed by package specification and is given for a given shape/volume, the mechanism is completely defined by the four variables  $rad1$ ,  $rad2$ ,  $ang1$  and  $ang2$ , the length of the connecting rod being univocally defined by these parameters.

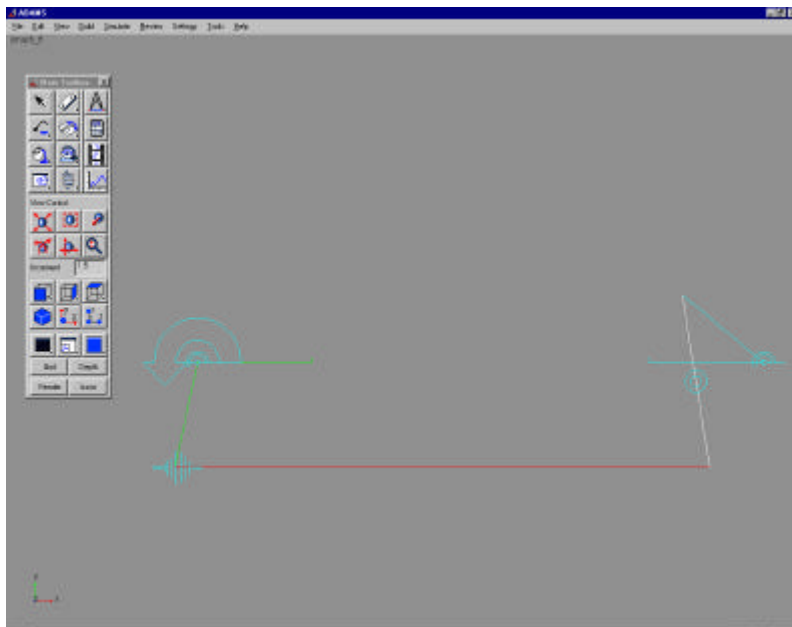
The problem to be solved is to define a suitable combination of the four dimensioning parameters in order to minimize the error between the vertical positions of points A and B during the motion. Practically speaking this means that since we need to pull down a certain amount of packaging material by two different fingers, we need to avoid uneven pulling, that can cause twisting of the tube and forming problems to the final package.

This problem is in reality what is commonly defined a function generation problem in kinematics. In fact, given a certain motion of link1 we want to define a suitable mechanism such that link2 performs a motion that is a function of the motion of link1.



**Fig. 18 Schematic of the folding flaps mechanism.**

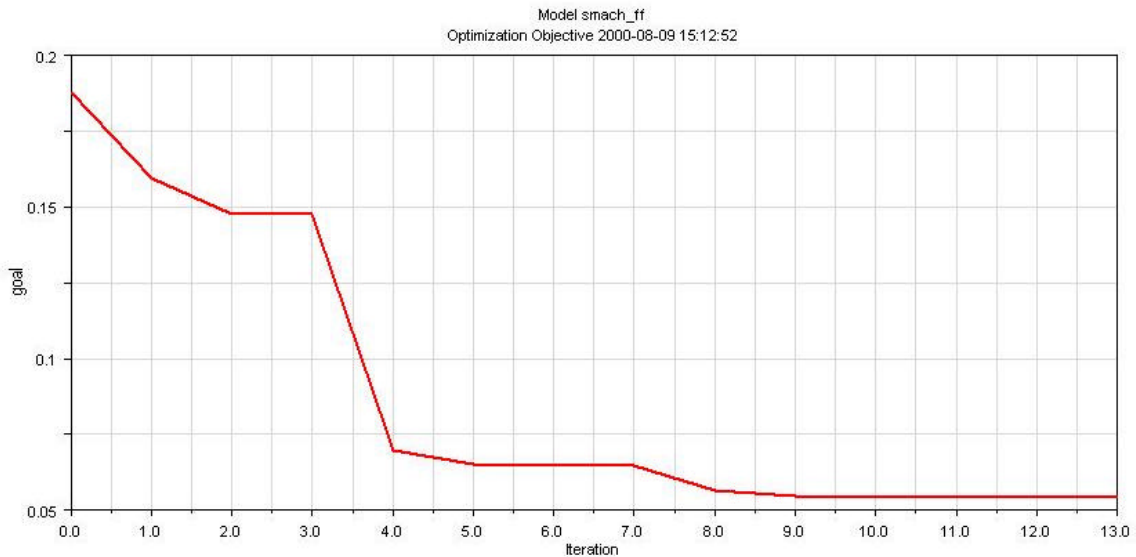
In this particular case the function is simply the identity function. This requirement will of course be satisfied within a certain tolerance. The problem is easily adapted as an optimization design study. In fig. 17 it is shown the simple Adams model used to solve



this problem. The model is defined with just starting approximate values for the driving parameters and with all the relationships needed to drive all the dimensions by these free parameters. Then a goal function is defined as the maximum absolute error between the vertical positions of the end point of the fingers. The goal function is then minimized with a constraint on angular velocities.

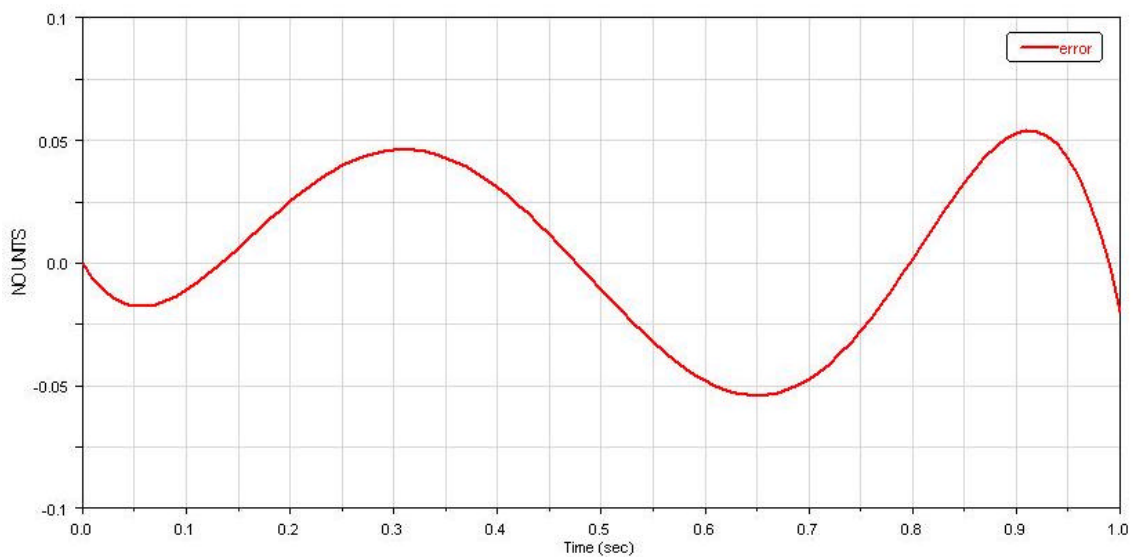
**Fig. 19 Adams model of the folding flaps mechanism.**

This constraint is set up in order to avoid reverse motion of the mechanism. In fig.20 we show the result of the optimization run in terms of the goal function versus the number of iterations.



**Fig. 20 Maximum error vs. optimization iteration number.**

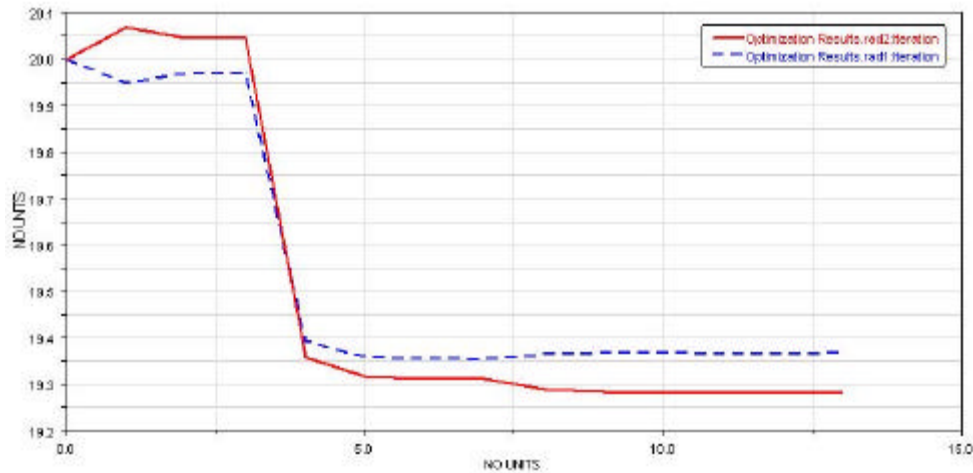
As can be seen from the figure the maximum error between the vertical positions has been reduced from about 0.2 mm to about 0.055 mm using 13 iterations. It should be noted here that the maximum allowed by the design specifications is 0.1 mm. Fig.21 shows the position error in the resulting configuration that corresponds to a rotation of the driving link1 of 90 degrees (the time on the x-axis is not the absolute physical time but a normalized one).



**Fig. 21 Position error during a 90 degrees rotation.**



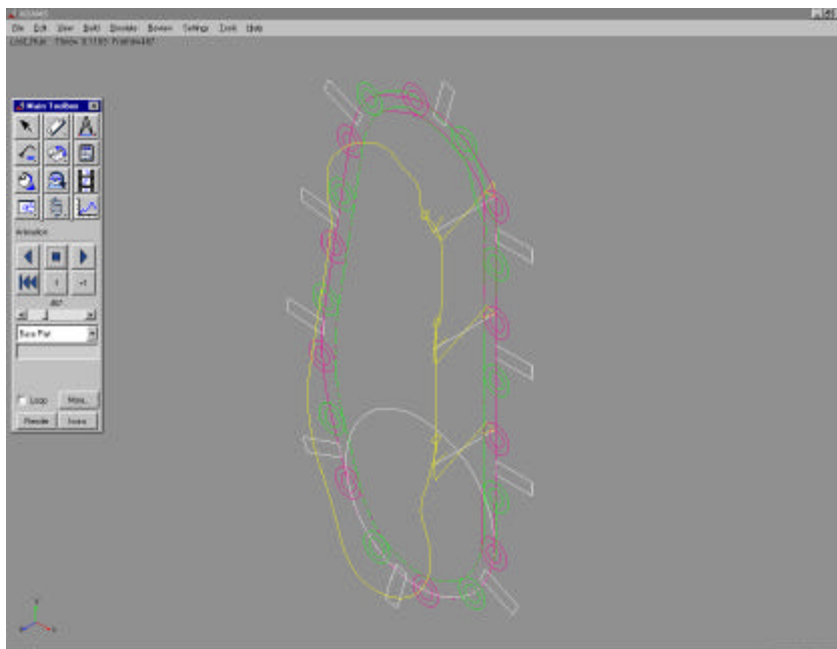
In fig. 22 we show the variation of the parameters rad1 and rad2 vs. iteration number.



**Fig. 22 Rad1 and rad2 values vs. optimization iteration number.**

#### **4. Phase diagram.**

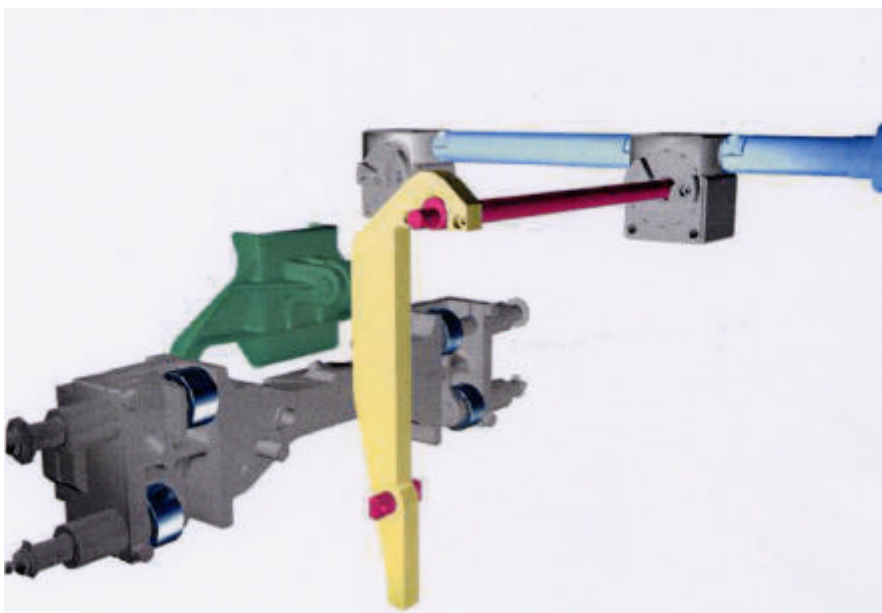
In the previous paragraph we have seen how to define the dimensions of the folding flap mechanism by optimization. The next task is to define the motion of this mechanism and synchronize it with the overall motion of the link that is actually carrying such a mechanism. The correct event sequence is such that first the jaw has to be closed, the jaw pressure applied and only when these two phases are completed the flaps can be folded to



reach the wanted position. A cam drives this motion, as can be seen in figs.5-6. Since the task for this mechanism is to pull down the packaging material by folding the flaps, it is important that the interaction between the mechanical system and the material is not causing any tear or damage.

**Fig.23 Folding flaps mechanism and cam assembled.**

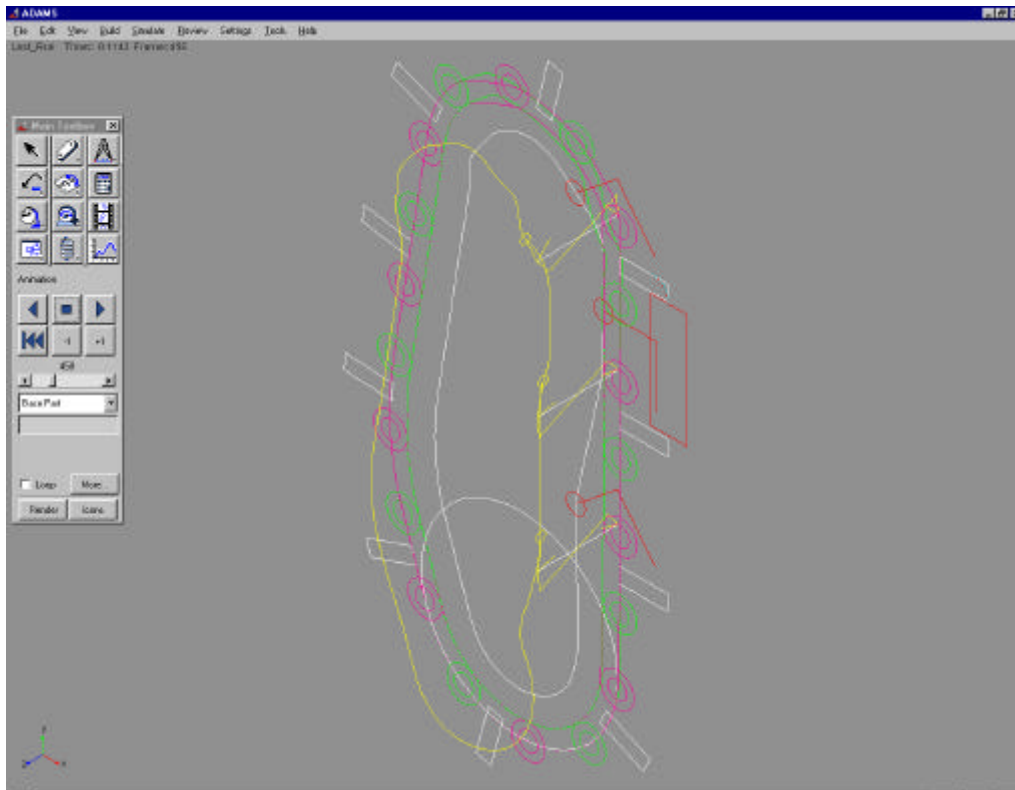
The motion should be a ramp-like function of time, and we use a standard modified trapezoid motion law to minimize the acceleration peaks. The resulting trajectory curve of the roller center (see fig.18) is then used to define a suitable cam for the folding flap mechanism. This cam is shown in fig. 23 together with the chain and the main cams. It is also shown how the folding flap mechanism looks like once assembled. It is worth to note that the cam is now generated by forward motion, since we are performing the synthesis of the motion of a mechanism that is related to the forward motion of the links. The same consideration applies for the next step in our process: the definition of the volume adjustment cam. This cam drives the motion of the volume box (or volume flaps) carried by each link. The volume box has the function to form the body of the package and at the same time it should give the package the correct volume content. This sub system is shown in fig.24. The volume box is pin-jointed directly on the corresponding link, while the position of a cam (yellow in figure), affects the volume box to close more or less. The cam position sets the volume of the packages. A stepper motor via gears and an eccentric shaft controls this position. Once the control system has defined the cam position, the cam shape defines the rotational motion of the volume box, and this has to be correctly synchronized with the motion of the link and the one of the folding flaps.



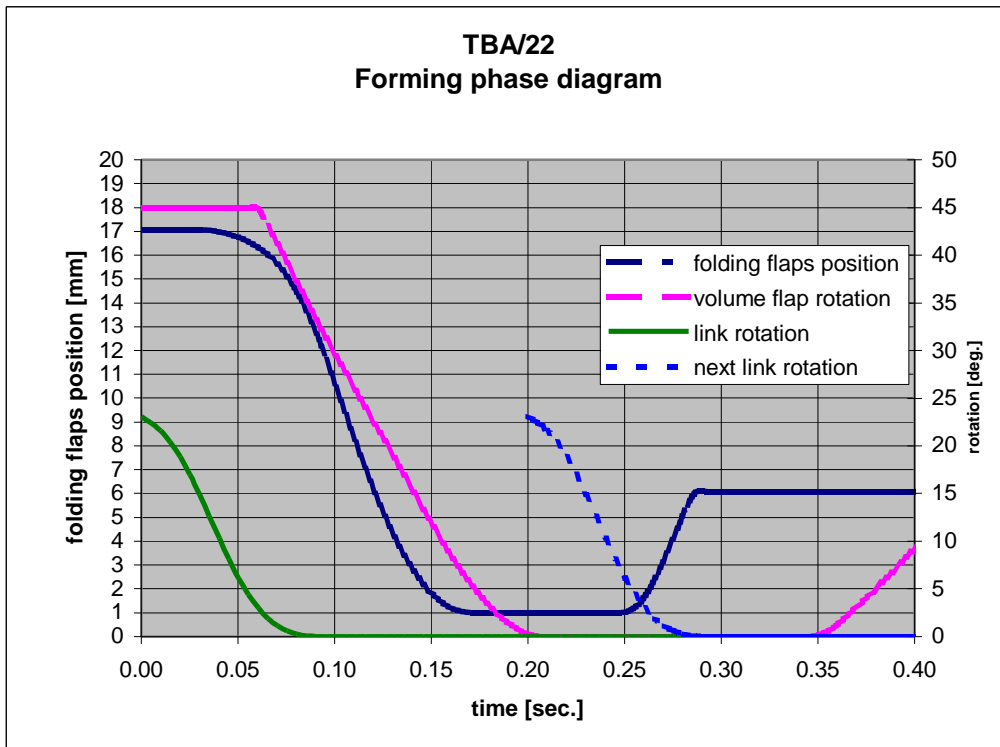
**Fig. 24 Volume adjustment function.**

At this stage it is quite simple to mount on our model the volume box parts and give them the appropriate motion, since all the other needed reference motion profiles are already well defined. In fig.25 it is shown the final assembly.

Having defined all the motions and subsystems involved in the package forming we are now in position to check the synchronization of all the motions involved in order to obtain a suitable phase diagram. Since there are some overlaps in the motion profiles this phase diagram is fundamental information that allows an early determination of possible forming problems due to inadequate timing. If this is the case then some design specification has to be accordingly redefined and the complete procedure should be re-run. An example of such phase diagram is shown in fig.26.



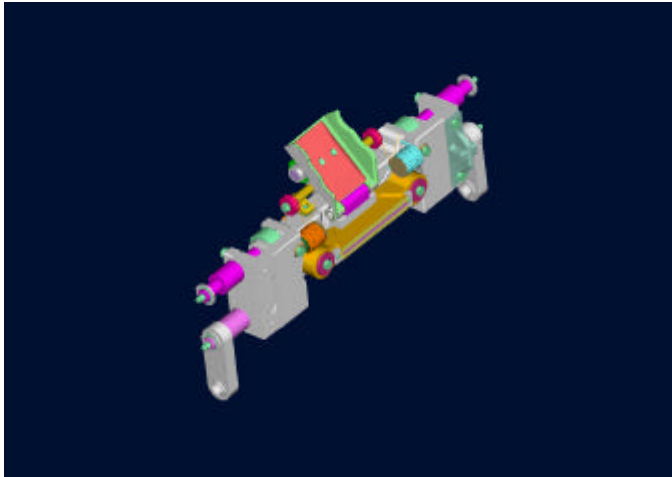
**Fig. 25 Complete chain assembly.**



**Fig. 26 Forming phase diagram.**

## 5. Cutting mechanism design.

Having completed the design of the forming section of the package we have now to define the last function performed by the jaw system, i.e. the cutting of the semi-finished package from the tube. In fig.27 it is shown the complete assembly of a pressure link.

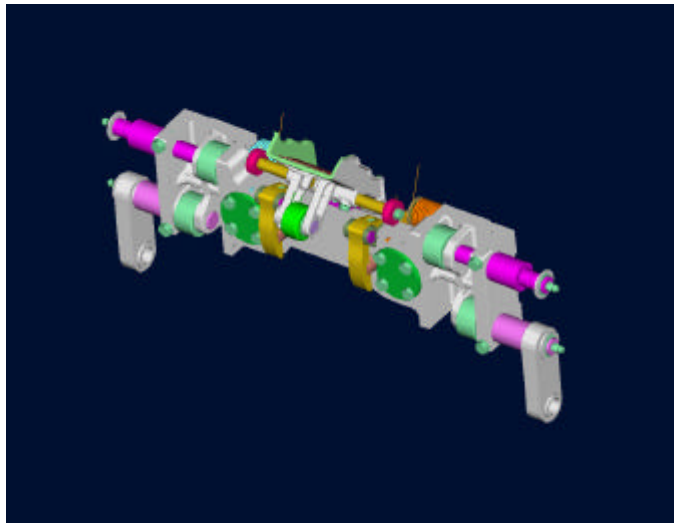


It is possible to see in the picture the volume box and, below it a protrusion with a rectangular hole (pink). A blade is located inside this hole. Once the link has reached a position during its movement where all the previously defined functions have been completed, two identical cams located in the back of the link, see fig.28, come in contact with fixed pins located on the frame and rotate around pin joints on the top of the link.

**Fig.27 Pressure link – front side view.**

The motion of these cams drives the motion of the blade (that is mounted with springs) pushing it until comes out from the hole and cuts the packaging material. The motion profile of the cutting device has been defined by experimental investigation, and is used to define the cam shape by inverse kinematic analysis.

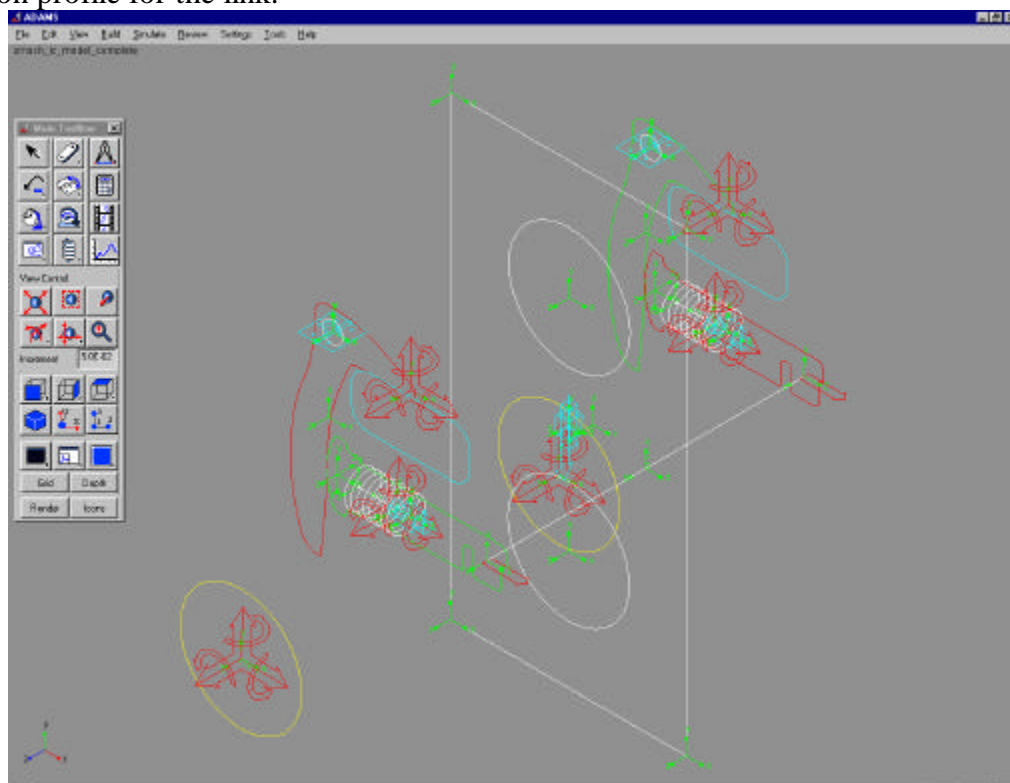
Since in this case the performance of the system is strongly affected by the interaction between the cams and the fixed pins, a dynamic analysis is in order to obtain information about the contact forces, needed to dimension the parts, and the actual motion profile of the blade and the corresponding cutting force.



**Fig. 28 Pressure link – back side view.**

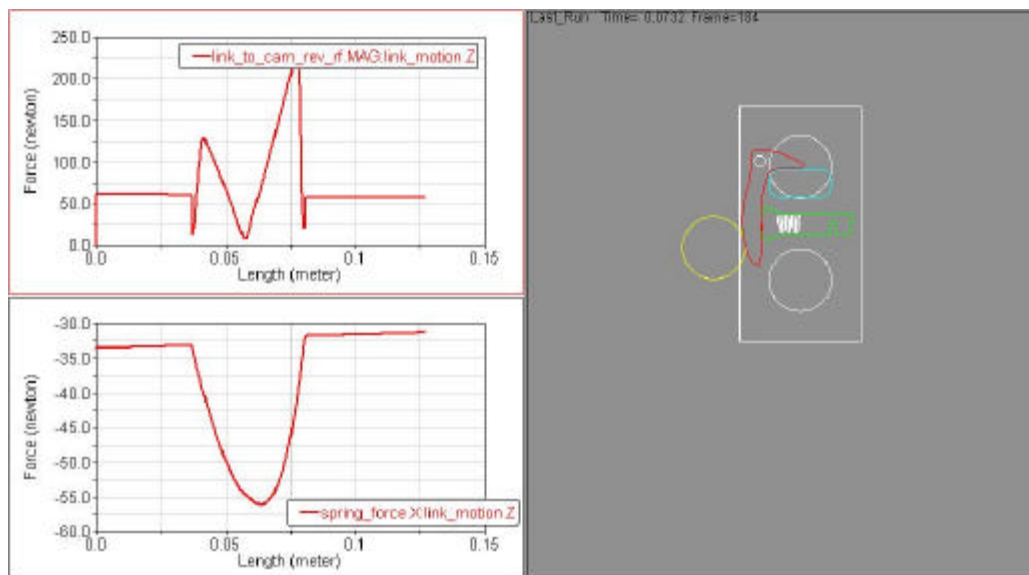
The total mass of the link is quite easy to estimate from the main dimensions given by the design specification, and the motion of the link in this stage should be a pure translational motion, so the inertia properties other than the mass play a little role. The mass of the cams is a direct consequence of their shape calculated by the inverse kinematic analysis already performed, so a fully dynamic Adams model can be easily constructed. This

model is shown in fig.29. We use the data from the previous steps to define a suitable motion profile for the link.



**Fig. 29 Cutting device model.**

The simulation results are shown in fig.30 in terms of spring reaction force and of reaction force on the pin joint. This information is needed for a correct dimensioning of the corresponding axle.



**Fig. 30 Simulation results.**

## **6. Conclusions.**

We have described a design procedure for the kinematic synthesis of a chain driven mechanism used to form packages inside a new concept of filling machine. The main results of this procedure are the set of points of the main cams, the folding flaps cam, the volume adjustment cam and the cutting cam, and the definition of the general layout of the system. Furthermore it is easy to extract useful information for the detail design of the links such as overall size, forces and so on.

The above procedure has several main positive features:

- a. It is fully parametric and is driven by macros that allow a quick evaluation of different solutions.
- b. The geometric description is kept at a minimum level, leaving out any complex manipulation of existing CAD models and reducing to a minimum the design constraints (at geometrical level) at this stage.
- c. It allows the evaluation of all the quantities needed to start the design work on the real parts and assemblies.
- d. It allows an early evaluation of the performances of the final design and in particular it is possible to have a good estimate of the actual phase diagram, highlighting possible timing/sequence problems during forming.
- e. Once the CAD models have been defined, it is quite easy to transfer back the inertia properties to Adams and run new analyses to estimate the forces. This estimation, although not very accurate since many dynamic aspects are not included, is however enough for dimensioning the parts and in particular the links.
- f. A fully dynamic model has been created based on the existing kinematic model introducing all the complex contact interaction among parts. This model enables us to perform a final verification of the design and an accurate calculation of the forces acting on the system. These results are then used in finite element analysis and fatigue evaluation. With this model it is also possible to check possible excitation of resonance frequencies.

Finally it is worth to note here that this parametric procedure has drastically reduced the development time needed to define the layouts corresponding to new volumes/shapes of packages.

## **7. References.**

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