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A Working Mechanism for Thermoforming Machine Design - Kinematic Analysis

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Abstract
A working module in contemporary thermoforming machines is generally based on mechanisms, due to the fact that sturdiness, precision and speed are most important characteristics. Structural and kinematical analysis of the module becomes an important issue in a design process, especially if dynamical parameters such as pressure angles and driving torques are accounted from the beginning. In single phased, tool rotation thermoforming machine, working module has two tasks, namely lifting of thermoforming tool and it's rotation. Therefore, it consists of two mechanisms. Lifting mechanism's design is common in contemporary machines. It is a complex planar mechanism consisting of three basic mechanisms connecting to each other. Coordination of these three mechanisms is the main problem. Rotation of the tool is much more complicated to achieve. Implanting a rotation device at the tool itself is a solution that is generally abandoned, due to dynamical characteristics of the tool. Therefore, a 2 DOF mechanism, preferably lever one, becomes imminent. Discussion of the design in detail, describing it and structurally and kinematicaly analyzing, taking into consideration dynamical requirements, is the task of this paper.
Keywords: complex plane mechanism, kinematic and structural synthesis, thermoforming machine

1. Introduction

This analysis is a part of design process for a new mechanical working module for thermoforming machine with rotational tool table.

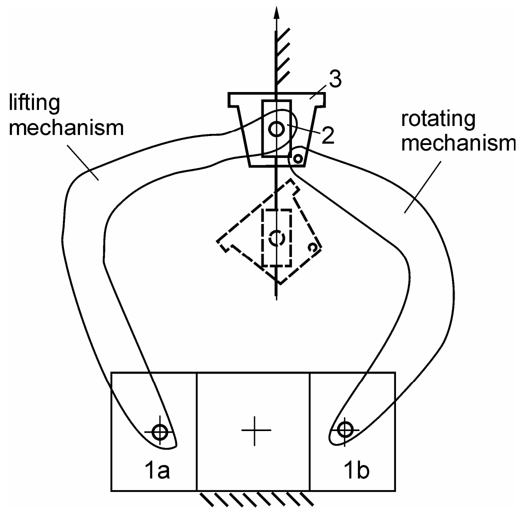

Figure 1. General appearance of working module

The module (Fig. 1.) is responsible for movement of the tool table (3). It consists of two mechanisms that work simultaneously: one provides tool table vertical movement, and the other it's rotation. Both mechanisms are driven via a single shaft, through separate cam mechanisms that provide required motion cycle for working links (Fig. 2.).

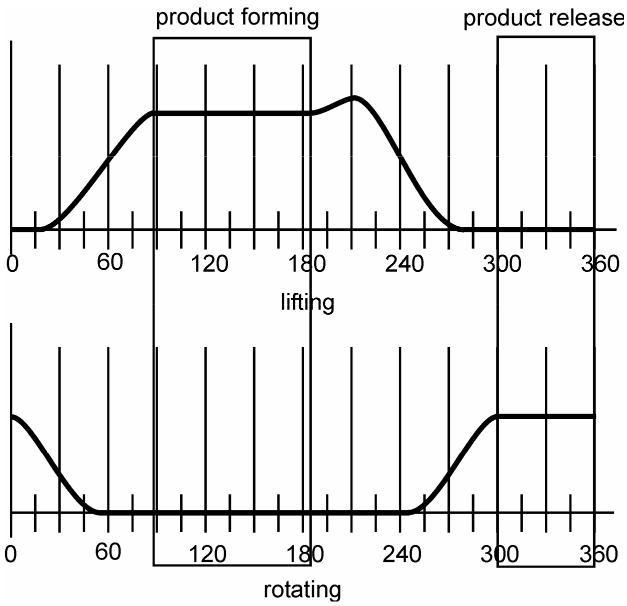

Figure 2. Working module cycle

Instead of classic open conjugate cam mechanisms, that are common in most mechanical systems, the standardized closed cam mechanism with rotational movement of the follower is introduced (1a and 1b on Fig.1). This mechanism, that has appearance of standard gear box, is purchased from Italian company Colombo Filippetti SPA and is known as oscillator. Beside obvious advantages regarding efficiency, precision and functioning in general, oscillator has two important limitations: due to restricted space, output torque and angle are limited to certain values. This was an important constraint for the design. The trapezoidal motion law [1] is adopted for the output link of cam mechanism. It gives modest maximum values for velocity and acceleration of output link.

2. Lifting mechanism

Vertical movement of tool table is obtained by combined cam-fourbar-slider mechanism (Fig. 3) that is common in contemporary machines of that kind.

It is a complex planar mechanism with cam (oscillator 2) on the input and lever-slide mechanism (6,7,8) on the output side, with a slider (8) attached to a tool table as a working link. Between them, a fourbar linkage (3,4,5) is generally added, (note that 5 and 6 are actually one link) in order to improve kinematical and dynamical characteristics. In fact, there are two simultaneous parallel lever slider mechanisms, on both sides of the tool.

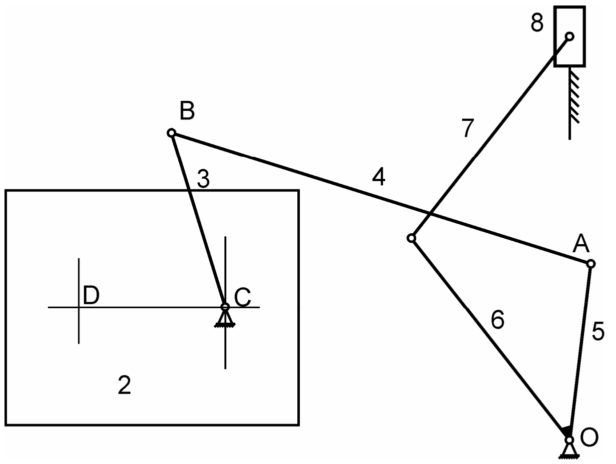

Figure 3. Sheme of lifting mechanism

Coordination of characteristics of these three mechanisms is the main problem. Since movement takes only a part of possible motion cycle, the determination of working interval for lever-slide and fourbar mechanism is most significant issue.

Most important constraints are: prescribed value of oscillator output crank rotation, prescribed movement of tool table (about 150 mm range) and the fact that there are two principal loads, namely inertial force of tool table and cutting force that is applied in the top most position and presents about 90 percents of maximum load.

Having in mind mentioned conditions, it is obvious that lever-slide mechanism is positioned vertically, with levers almost collinear in up most position of the slider in order to facilitate achievement of cutting force.

After initial design that can achieve required motion interval for the working slider is set, an optimization process is conducted in order to minimize pressure angle in fourbar mechanism and required input torque in the oscillator. These requirements define actual position of fourbar links, especially in critical position, namely when the cutting force is applied. Important factor is also design requirement that oscillator box must be far enough from the tool table working space, and lying generally on the same horizontal plane as basic bearing of slider mechanism (point O).

This process is quite complicated due to the fact that pressure angle is a function of positions (link angles), while inertial force is a function of accelerations. So, optimization is obtained by simulation of complete complex mechanism.

Equations for kinematical parameters are quite common since well known mechanisms are in question. Equations for output link displacement, velocity and acceleration in fourbar and lever-slide mechanism can be easily found in literature ([1]), so they will not be discussed in this paper.

3. Rotating mechanism

The real challenge was the design of mechanism that has to control rotation of the tool table. Since the tool table is sliding and rotating simultaneously, the mechanism must have two degrees of freedom, with translatory movement as one input (main), and rotation movement on the other side as the other (control).

The initial choice - common five link 2 DOF mechanism proved to be unsatisfactory, because of the important design requirement: at the beginning of the vertical movement, the tool table must remain in vertical position. Since the rotation of certain link in general 2 DOF mechanisms depends on both input motions - main and control, in order to keep table unrotating, the control movement must compensate for the main movement. It is certainly difficult and expensive to obtain precise compensation, although it is essential for proper module functioning.

Therefore, the design team is encouraged to pursuit another solution - a mechanism that will obtain the rotation of the tool table according to control movement only. This presents a typical problem of structural synthesis. The solution is found when the problem is restated in following matter: find one degree of freedom mechanism, which has translatory input motion, in which angle of floating link will not change during the motion cycle. The solution for this problem is mechanism presented in Figure 4.

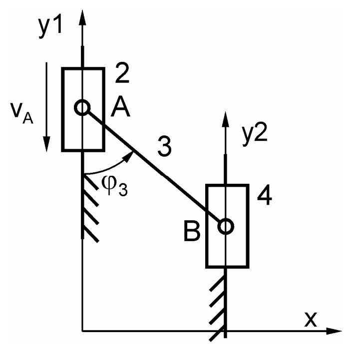

Figure 4. 1 DOF mechanism

This simple mechanism has slider 2 as input link, and it is obvious that angle of link 3 (φ_3) depends only on x-distance between axes y1 and y2.

Including a device that can change this distance will lead to final design of required mechanism. It is presented on Figure 5. As can be seen, a lever-slide mechanism (7,6,5) is added to previously mentioned mechanism.

At the machine itself, slider 2 is connected to lifting mechanism (actually, it is the same slider denoted as 8 at the lifting mechanism), providing vertical slide, link 3 is connected to tool table, while link 7 is an output crank of the oscillator box and provides tool rotation control.

The presented mechanism (Fig. 5) is an adopted one, although some other solutions were also inspected (see [2]).

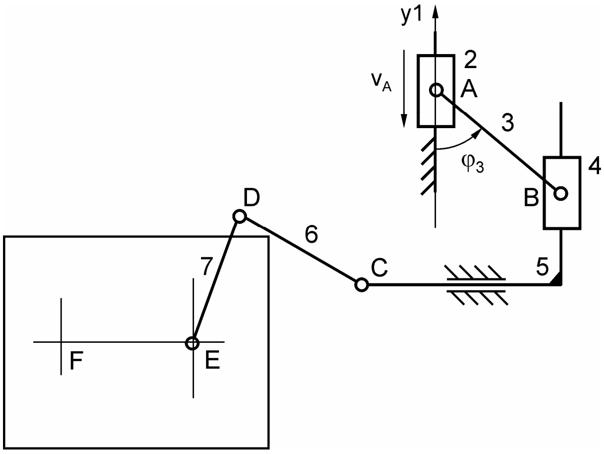

Figure 5. Sheme of rotating mechanism

Equations that are used for calculation of interesting kinematic parameters - angular displacement (φ_3), velocity (ω_3) and acceleration (ε_3) of the tool can be obtained by closed loop vector analysis in the form:

$$\varphi_3 = arccos\frac{x_C + \overline{CG} - L_2}{\overline{AB}} \qquad (1)$$

where

$$x_C = \overline{ED}\cdot cos\,\varphi_7 + \overline{DC}\cdot cos\,\varphi_6 \qquad (2)$$

$$\varphi_6 = arcsin\frac{L_5 - \overline{ED}\cdot sin\,\varphi_7}{\overline{DC}} \qquad (3)$$

All position parameters can be seen on Figure 6.

$$\omega_3 = -\frac{\dot{x}_C}{\overline{AB}\cdot sin\,\varphi_3} \qquad (4)$$

$$\dot{x}_C = -\overline{ED}\cdot\omega_7\cdot sin\,\varphi_7 - \overline{DC}\cdot\omega_6\cdot sin\,\varphi_6 \qquad (5)$$

$$\omega_6 = -\frac{\overline{ED}}{\overline{DC}}\cdot\omega_6\cdot\frac{cos\,\varphi_6}{cos\,\varphi_7} \qquad (6)$$

$$\varepsilon_3 = \frac{\ddot{x}_C + \overline{AB}\cdot\omega_3^2\cdot cos\,\varphi_3}{-\overline{AB}\cdot sin\,\varphi_3} \qquad (7)$$

$$\begin{aligned}\ddot{x}_C = &-\overline{ED}\cdot\omega_7^2\cdot cos\,\varphi_7 - \overline{ED}\cdot\varepsilon_7\cdot sin\,\varphi_7 - \\ &-\overline{DC}\cdot\omega_6^2\cdot cos\,\varphi_6 - \overline{DC}\cdot\varepsilon_6\cdot sin\,\varphi_6\end{aligned} \qquad (8)$$

$$\varepsilon_6 = \frac{\overline{ED}\omega_7^2\,sin\,\varphi_7 - \overline{ED}\dot{\omega}_7\,sin\,\varphi_7 + \overline{DC}\omega_6^2\,sin\,\varphi_6}{\overline{DC}\cdot cos\,\varphi_6} \qquad (9)$$

Input parameters are control rotation angle φ_7, velocity ω_7 and acceleration ε_7 , as well as vertical displacement y_2 , velocity v_2 and acceleration a_2. It is obvious that none of interesting parameters is a function of vertical slide of the tool, but only of control motion.

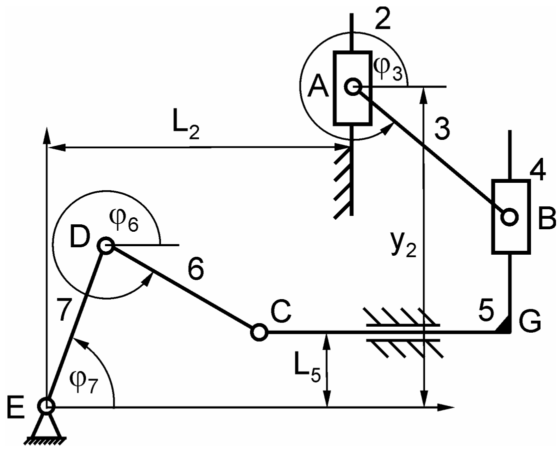

Figure 6. Kinematic sheme of rotating mechanism

Most important conditions for the initial design are: prescribed rotation angle of tool table (80 degrees), prescribed rotation angle of oscillator output crank, the fact that principal load is inertial torque of the tool table and that rotating oscillator must lie parallel to lifting oscillator, since their input shafts are coaxial. The whole mechanism is situated on the far side of the machine, so there were not any constraints about links interference.

The optimization process included efforts to minimize pressure angles, especially on input lever-slide mechanism, and required control input torque.

4. Results

Since the design of machine prototype is completed, together with complete kinematic and dynamic analysis, some of the most interesting results will be presented in approximate values.

Both mechanisms occupy the same space of aprox. 1600x1000 mm in vertical plane, including oscillator boxes. Links have lengths from 185 mm (working link on rotating mechanism) to 875 mm (floating link on fourbar mechanism for lift).

Machine is designed for capacity of 30 cycles per minute. Lifting time is aprox. 0.38 sec, and rotation time aprox. 0.31 sec.

Total lift of the tool table is 165 mm and rotation is 80 degrees.

Maximum cutting force is 200 000 N, tool table weight is about 550 kg, and moment of inertia 17 kgm^2.

4.1 Characteristics of lifting mechanism

On the input side, link 3, as a cam mechanism follower has a skewed trapezoidal motion. This means that interval, and therefore maximum values are not the same for acceleration and deceleration. Because of the specific cycle diagram (R-D-R-F-D) at the point of cutting, some further modifications of trapezoidal motion have to be applied [3]. Total rotation angle is 44.5 degrees, maximum angular velocity is 4 sec^{-1}, and maximum angular acceleration 27 sec^{-2}.

Floating link 4 has total angular interval of less than 6 degrees and it's angular velocity and acceleration are neglectible.

Link 5(6), as the output link in fourbar linkage, has generally trapezoidal shape motion but with variable acceleration instead of constant. Total rotation angle is 40 degrees, maximum angular velocity is 3.5 sec^{-1}, and maximum angular acceleration 28 sec^{-2}.

Output slider 8 has alike cycloid motion, with maximum velocity of 1 m/sec and acceleration of 11 m/sec^2.

Looking at the dynamical parameters, beside cutting force, maximum inertial force (including gravity) of the tool table is about 12 000 N. However, these forces can be, in some extent, balanced by a hydraulic cylinder force acting against the load. Cylinder of 7500 N is adopted.

Due to the favorable positioning of mechanism, especially in the critical moment, the actual required torque at the link 5 is about 6000 Nm in peak and 1000 Nm in average. At the output of the oscillator, these values are 4500 Nm and 1000 Nm.

Pressure angles are specially examined at the later end of fourbar linkage (links 4 and 5). The maximum value is about 42 degrees, which is quite a lot but still acceptable. In critical interval of cutting, that angle is 1 to 4 degrees that is very satisfactory.

4.2 Characteristics of rotating mechanism

Generally speaking, kinematic characteristics of the mechanism are quite severe, because of very short time in which substantial rotation angle of tool table has to be achieved. On the other hand, restricted oscillator output crank rotation angle brings to unfavorable torque transmission through the mechanism.

Link 7, an output oscillator crank, has common trapezoidal motion. It's total rotation angle is 45 degrees, maximum angular velocity 5 sec^{-1}, and acceleration 40 sec^{-2}.

Link 6, a floating link of input lever-slide mechanism, has a rotation interval of less than 5 degrees and it's angular velocity and acceleration are neglectible.

Horizontal slider 5 has movement interval of some 137 mm. It's motion is alike trapezoidal, with maximum velocity of 1.6 m/sec and acceleration 13 m/sec^2.

Vertical slider 4 has complex motion because it is a function of both input movements. It's interval is about 43 mm, with maximum velocity of 1.3 m/sec and acceleration 24 m/sec^2. These values are quite substantial, but it's mass is much smaller than for other elements, so it is much of disadvantage.

Link 3, that is firmly attached to the tool table is much more influenced by control than by sliding movement, so it's motion is alike trapezoidal. As said before, it's rotation range is set to 80 degrees, symmetrically to y-axis. Values of angular velocity and acceleration are the most problematical in both mechanisms. They are 9 sec^{-1}, and 78 sec^{-2}. Having in mind mass of tool table, this obviously presents the critical point of the design, but this fact is not consequence of the module solution, but design specifications themselves.

Intensive angular acceleration of the rotating table results in massive inertial torque that is about 1200 Nm. Greatest transmission force in sliding pair 4-5 reaches about 9000 N that has to be accounted in the design of slides. Because unfavorable ratio between input and output rotation angles, as stated before, required torque at the output side of the oscillator is about 2500 Nm. That calculation denies the common opinion that it is easier to obtain rotation than lifting.

Interesting pressure angles are between links 3 and 4 and 6 and 5. In the first case there is not a possibility to change a lot because there are two critical positions – one at the acceleration and one at the deceleration interval. Therefore, the rotation interval is positioned symmetrically to y-axis, and pressure angle in case of maximum torque is about 38 degrees. In second case, the problem is much easier and critical transmission angle is about 7 degrees.

5. Conclusion

The paper presents problems that the designer encounters in the process of developing a mechanical system including complex mechanisms. Some of key issues appear to be:

- Although it is advisable to use simple well-known mechanisms, sometimes it is necessary to perform structural synthesis and develop a mechanism of your own, that can fulfill the mission.
- Equations for performing kinematic analysis are generally available for simplest mechanism. Nevertheless, in complex mechanisms a significant effort has to be made to combine known expressions, and sometimes to add some analytical knowledge.
- In optimization phase it is necessary to add basic dynamical considerations, specially regarding maximum loads, pressure angles and required torques

All of these issues have been dealed with in the design process for new working module for thermoforming machine, and quite satisfactory results are obtained.

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