

POLITECNICO DI MILANO

MSC IN MECHANICAL ENGINEERING

FUNCTIONAL MECHANICAL DESIGN

Turning pad subsystem

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1 Introduction

The subsystem has the purpose of discharging the soap once the product has been definitively made. The tilting pad motion is a pure rotation and takes place around a fixed axis. The movement is realized through a slider crank mechanism in which the slider is moved by a translating cam. The slider has a mass of 2.3 kg and the tilting pad is characterized by a momentum of inertia (with respect to the axis of rotation) $J = 1.1\text{kg m}^2$. The motion law of the tilting pad is:

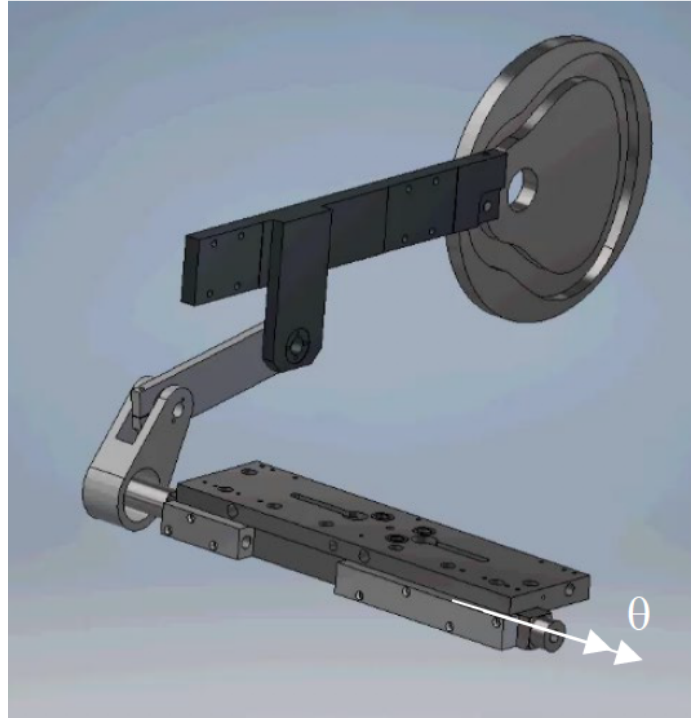


Figure 1: Example of uploading subsystem.

The motion of the system has to achieve the points described by Tab. 1.

$\alpha = 0^\circ$	$\theta = 45^\circ$
$\alpha = 70^\circ$	$\theta = 0^\circ$
$\alpha = 200^\circ$	$\theta = 0^\circ$
$\alpha = 360^\circ$	$\theta = 45^\circ$

Table 1: Motion law

2 Objectives

Considering the driver unit moves at constant speed, we need to

- i Design the motion law of the tilting pad.
- ii Synthesize the slider crank mechanism and the cam mechanisms with a translating follower.
- iii Analyse the mechanism through kinematic parameters (transmission angle, pressure angle and undercut) and compute the motion law obtained with the designed mechanism
- iv Calculate forces transmitted and the motor torque with a multibody model.
- v Critically evaluate the feasibility of the system.

3 Methods and procedure

The motion law is described for θ . Since we are constructing a translating cam whose displacement x is the displacement of the slider on the slider-crank mechanism. We first choose a , b , e and α_0 , accounting for performance parameters as transmission angle μ . From that, we obtain the desired motion law in x and generate the corresponding cam.

3.1 Slider-crank

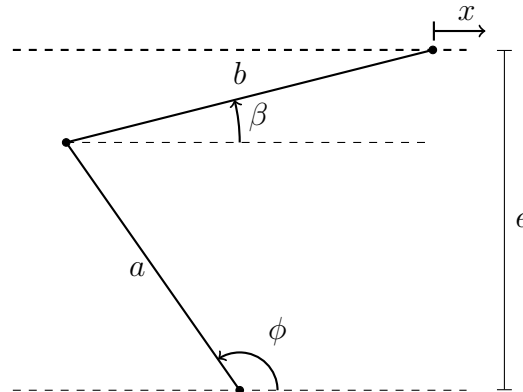


Figure 2: Schematic of the slider-crank mechanism.

With the motion described for θ , we derive the motion for x as (see Fig. 2)

$$x = a \sin(\phi) + b \cos(\beta), \quad (1)$$

where a is the crank length, b is the slider length, e is the height of the slider with respect to the ground and

$$\beta = \arcsin\left(\frac{e - a \sin(\phi)}{b}\right), \quad (2)$$

$$\dots \phi = \phi_0 + \theta. \quad (3)$$

We initially choose all the motions to be cycloidal, so the normalized motion law is

$$\begin{cases} \ddot{y} = c_a \frac{h}{t_a^2} \sin\left(\frac{2\pi t}{t_a}\right), \\ \dot{y} = \frac{h}{t_a} \left[1 - \cos\left(\frac{2\pi t}{t_a}\right)\right], \\ y = h \left[\frac{t}{t_a} - \frac{1}{2\pi} \sin\left(\frac{2\pi t}{t_a}\right)\right], \end{cases} \quad (4)$$

where $c_a = 2\pi$, h is the lift from time 0 until t_a . We can also describe the motion law with respect to the master angle (α) and thus the $\dot{y} = y'\omega$ and $\ddot{y} = y''\omega^2$, where y' and y'' are the geometrical velocities and accelerations, respectively, and ω is the constant angular speed. The advantage of the cycloidal motion is that because the relationship between x and θ are trigonometric, both the motions will be cycloidal (not true if I choose a different motion for θ , let's say, cubic).

The crank a length should be long enough so that low forces are needed to rotate the inertia but short enough to avoid oversizing. For a giving a , the eccentricity e should be close to a to ensure a high transmission ratio. For the case we chose $e > a$; in this way we won't meet a dead center during the working condition. The coupler length b is chosen larger so that a significant displacement x takes place for the given angular displacement.

3.2 Cam design

For the design of the first cam, we use the obtained motion law for X and consider the equations for a translating cam with a roller. First, we find the pressure angle ϕ as

$$\phi = \frac{X'}{R_{b0} + X}, \quad (5)$$

where $X' = \partial X / \partial \alpha$ is our geometrical velocity (remember the abscissa is the master angle α) and R_{b0} is the base radius of the cam. The next step is to find the pitch

profile. The curvature radius of the pitch profile is found by

$$\rho_0 = \frac{[X'^2 + (R_{b0} + X)^2]^{3/2}}{(R_{b0} + X)^2 - (R_{b0} + X)X'' + 2X'^2}, \quad (6)$$

while the curvature radius of the cam profile is

$$\rho = \rho_0 - R_r, \quad (7)$$

where R_r is the roller radius.

Using Carnot theorem, one can find the polar coordinates of the cam as

$$r(\alpha) = \sqrt{R_r^2 + (R_{b0} + X)^2 - 2R_r(R_{b0} + X)\cos\phi}, \quad (8)$$

$$\varphi(\alpha) = \alpha + \arcsin\left(\frac{R_r \sin\phi}{r}\right). \quad (9)$$

After the definition of the cam and of the slider-crank mechanism, their profile was designed through MATLAB routines. The physical object of the cam, that will later be used in ADAMS, was constructed into a CAD software, SolidWorks, from the profiles given by the MATLAB routine.

4 ADAMS analysis

5 Results

We choose the following parameters for the slider-crank mechanism

$$a = 80 \text{ mm}, \quad b = 100 \text{ mm}, \quad e = 90 \text{ mm}, \quad \phi_0 = 125^\circ.$$

The resulting motion law is depicted at Fig. 3. We chose the cycloidal motion law for it has no discontinuities at high derivatives (until acceleration), favoring the smoothness of the system.

With the above motion law, we checked for the transmission angle, given by Fig. 4; it varies from 76 til 84 degrees (a good transmission angle is above 45°).

Thus, we design the translational cam with the equations in (9). We remember that x is positive to the right and that the contact point of our cam is at the left, thus a negative sign in the x-coordinate of the cam has to be added in order to obtain the right cam. Furthermore, the cam should then rotate counter-clockwise. The respective motion law is depicted at Fig. 5. For this corresponding stroke, we choose a base radius $R_{b0} = 300 \text{ mm}$ and a roller radius $R_r = 30 \text{ mm}$, that seemed

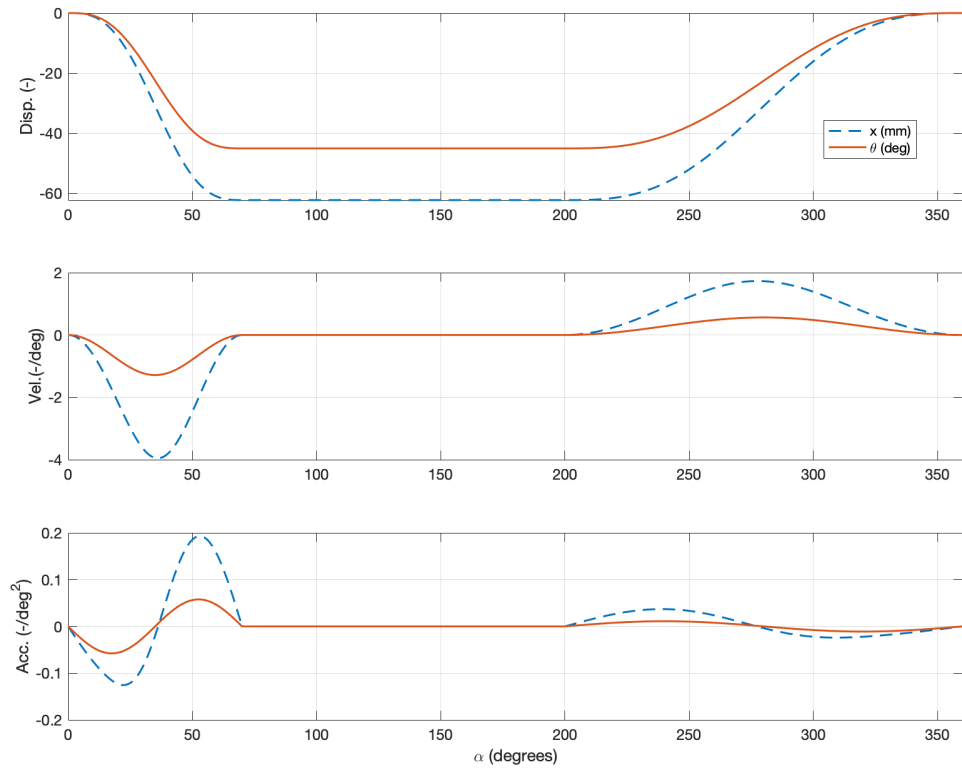


Figure 3: Motion law for both x (in blue) and θ (in red) with cycloidal choices.

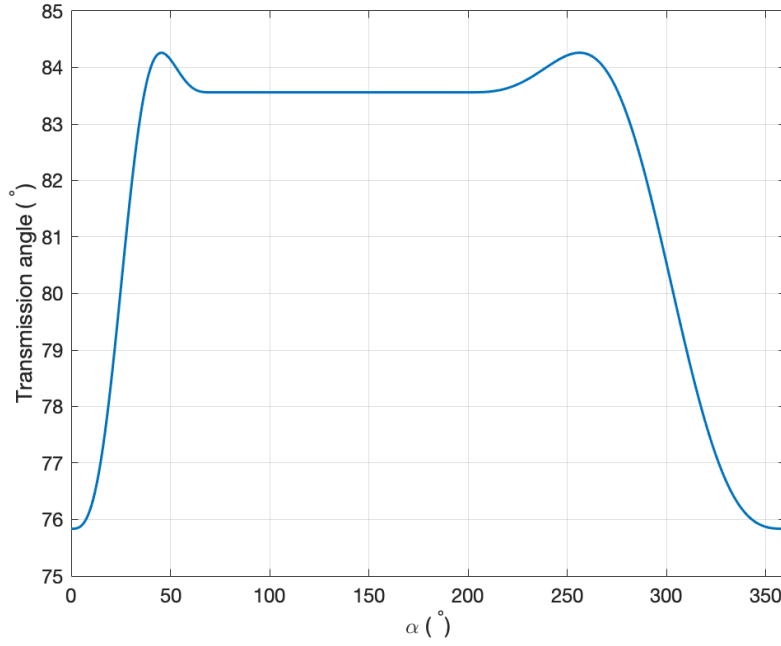


Figure 4: Transmission angle of the slider-crank mechanism.

reasonable to keep the pressure angle under control (for the given motion law) and to avoid undercut. Fig. 6 shows how the pressure angle between the cam and the follower varies as function of the master angle. It is kept bound and its maximum value is less than 45 degrees. Fig. 7.a shows the curvature of the both the pitch and cam profile, while the Fig. 7.b plots $\rho \cdot \rho_0$ vs α . If $\rho \cdot \rho_0 < 0$ the undercut occurs. From the plot, it's clear (any trespassing of the middle axis would have been graphically accused) that no undercut takes place here.

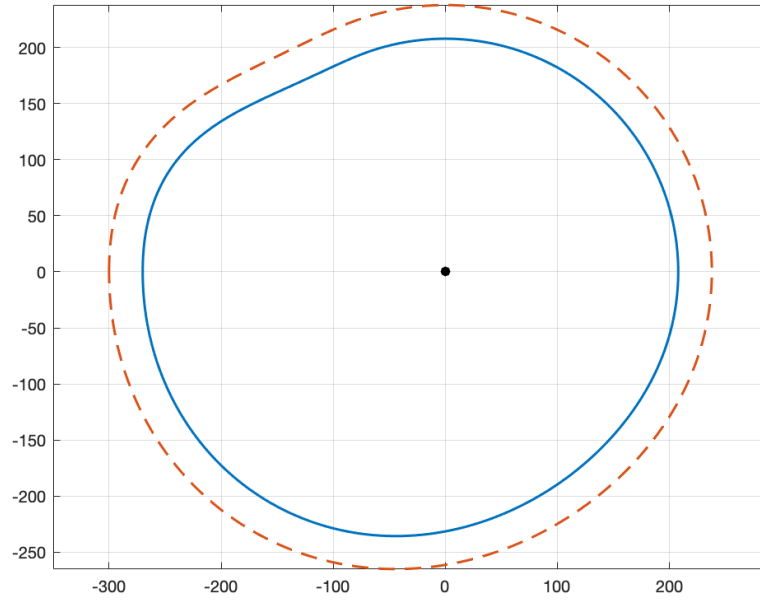


Figure 5: Translational cam generated for the first motion law (cycloidal motions).

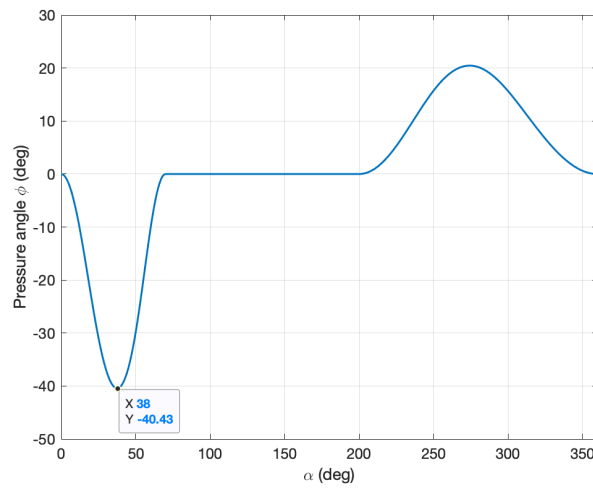


Figure 6: Pressure angle related to the first cam-follower system.

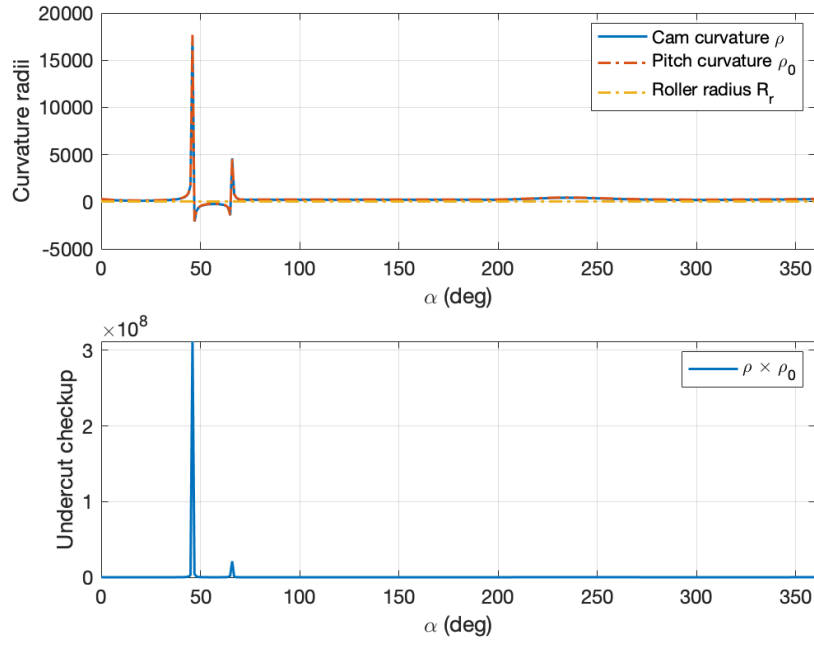


Figure 7: Curvature radius and undercut condition check.

6 Discussion