WP6: Applications
D6.4: Experimental Validation of non-Smooth Bifurcations
Part III

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Abstract

This deliverable reports experimental results validating the dynamical behavior of a cam-follower system (including time series plots and stroboscopic diagrams). The goal of the experimental research activity is to analyze all the non-linear features occurring at certain velocity ranges where collision phenomena generate switching transitions between different dynamical configurations. Experimental results are compared with both theoretical developments and numerical analysis based on a simulation tool.
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Chapter 1

Experimental Analysis of a cam-follower mechanical system

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1.1 Introduction

A number of dynamical systems contain discontinuities due to the presence of structural components with displacement constraints. Examples include bouncing or hopping systems, vibro-mechanical impacts in machine vibrations, loosely connected members, and gearing systems with fatigue-induced over-tolerances. All represent situations where impacting oscillator models can provide a valuable insight to understand the observed dynamical behavior [2]. Although such systems typically operate in linear regimes over certain parameter ranges, the discontinuities in the force-deflection relationship have been shown to produce a characteristic non-linear behavior, such as amplitude jumps, subharmonics and chaos [3]. The phenomenon of impact could be either desirable, when it represents the base of operation, as for example in pneumatic hammers, impact print hammers and heat exchangers, or undesired when is destructive and should be eliminated, as for instance in gear-boxes [4].

In this context, cam-follower systems can be chosen as a very general and relevant benchmark problem since they are widely used in various machines and mechanical engineering devices [2]. For instance, all types of automated production machines including screw machines, spring winders and assembly machines, rely heavily on this kind of systems for their operation.

The most common application is to the valve train of internal combustion engines (ICE) [5], where the effectiveness of the ICE is based on the proper working of a cam-follower system. For this specific application the presence of discontinuities can be really critical and must be avoided or controlled since the purpose of the valve train is to open and close both the intake and exhaust valves of the ICE. A typical pushrod valve train contains the following components: cam, follower, pushing rod, rocker arm and valve springs. As the camshaft rotates, the cam imparts a translation motion to the follower and pushrod (see Figure 1.1). The pushrod then pivots the rocker arm which opens the valve. The valve springs provide the restoring force to close the valve after the maximum lift is obtained.
During operation, it is important to keep all the mechanical parts in close contact with each other so that the motion of the mechanism perfectly reflects the motion of the cam. Unfortunately, in practice, as the engine speed increases, the valve train motion can be different from the ideal desired kinematic behavior due to inertia of components, and the surge of valve springs. These phenomenon leads to valve floating and bouncing, which can seriously deteriorate the engine performance. For instance, valve floating occurs when the inertial force of the valve train components, exceeds the spring force of the valve springs, thus allowing components to separate and causing the valve to exceed the maximum lift of kinematic motion and close with an abnormally high velocity. Valve bouncing instead occurs when the valve closes against the seat with a sufficiently high velocity, such that it physically bounces off the seat and remains open as the piston begins the compression cycle, thus allowing the air-fuel mixture going out of the combustion chamber. It has been experimentally observed, that at certain engine speed termed limit speed, the valve bounce amplitude increases dramatically, thereby resulting in what seems to be a chaotic valve motion and a loss of engine performance. Then, reduction in valve bouncing and/or valve floating is established as primary goal of valve designs including cam-follower mechanism.

All the applications based on a cam follower mechanism require a deeper insight in the system dynamics, i.e. taking explicitly into account the occurrence of gaps between connecting components. Obviously, embedding possible collisions in the representative model generates discontinuities and nonlinearities in the system equations, reducing the applicability and the effectiveness of traditional modal analysis tools.

In order to put our experimental activity in a theoretical context, we use the theory that has been recently developed to deal with non-smooth bifurcations in discontinuous dynamical systems [6] [7], thus providing the analytical framework to characterize the complex behavior exhibited by mechanical systems with impacts and friction. For example, in [8] such a theory is employed to analyze the dynamics of a cam-follower system by means of numerical bifurcation analysis, carried out on a lumped-parameter single-degree of freedom model reference.

The goal of the experimental activity described in this chapter (carried out at the SINCRO Laboratory in the University of Naples Federico II [9]) is to observe the actual behavior of the
physical system under different environmental conditions, thus validating all the analytical studies performed on the equivalent mathematical description and complementing the simulations results. To this aim an experimental rig composed of a resized and modified cam-follower system was purposely designed and built. The system is complemented by all the motion and measurement hardware necessary to carry out the experimental analysis.

The outline of the report follows: in section 1.2 the physical elements of the experimental setup are described. In section 1.3 a corresponding mathematical representation is derived. In section 1.4 preliminary numerical results of the rig numerical analysis are showed and related in section 1.5 with those obtained under real implementation. Conclusions are drawn in section 1.6.

1.2 Description of experimental setup

Next, a brief description of the experimental prototype of a cam-follower system is developed.

1.2.1 The experimental rig

Figure 1.1 shows a cam-follower system used in automotive valve actuation. This is an overhead camshaft engine. The camshaft operated against an oscillating follower arm that in turn opens the valve. The cam-joint is force-closed by the valve spring. Maximum cam speed in this kind of applications can range from about 2500 rpm in large automobile engines to over 10000 rpm in motorcycle racing engines. From the application viewpoint, the velocity is a crucial parameter to be properly controlled to induce a desired behavior. Essentially, unwanted nonlinear behavior, may be caused by impacts that have to be avoided.

![Diagram of cam-follower system](image)

Figure 1.2: Radial cam with an oscillating flat-faced follower. in Cam design and manufacturing hand book. R. L. Norton, Industrial press. inc.
Inspired by these kinds of problems, a test-bench for a radial cam with a flat-faced follower was realized at the University of Napoli Federico II. A typical geometry for the calculation of the cam contour for an generic oscillating flat-faced follower is in Figure 1.2 (see also [2] for further details), where all the features of the mechanical system are summarized.

Since our attention is focused on the nonlinear behavior of the cam-follower mechanism, it is not necessary, at this stage of the project to explicitly consider the engine in the rig. Without loss of generality, a simple mechanical system composed by two rigid contacting bodies (the cam and the follower) is used as an experimental rig sufficient for the practical visualization of all the phenomena related with the bouncing of a follower over a cam surface. The presence of the spring tied to the follower/rocker arm provides the necessary restoring force. An electric servo motor provides the motion of the cam according to a prefixed velocity profile. In Figures 1.3 and 1.4, the experimental rig is shown and in what follows all the components and their design are described in detail.

![Figure 1.3: Schematic description for experimental rig](image)

Notice that it has been also possible to design a push-road translational cam-follower system, but the choice of a rotational rig seemed to preferable for our experimental purposes, since an oscillating flat-faced follower provides a lower friction, lower wear rates and ease of replacement. Furthermore, our experimental and theoretical effort is based on the detection of multiple impacts, chattering, grazing contacts and so on. This implies the necessity to perform reliable contact measurements. By choosing a push-road translational cam rig, this can not be done easily, since the high frequency impacts do not suggest to employ proximity transducers, such as the piezoelectric ones, forcing to choose optical sensors, like high speed lasers, which are efficient, but expensive. Instead, by designing a ring based on a rotational geometry, measurements of the cam and follower positions are simply performed by incremental encoders placed on the cam-shaft and the follower-shaft.

Finally we remark that, although the complexity of a generic cam-follower mechanism, mainly
due to additional effects acting on the system (such as for example camshaft torsion and bending, backlash, squeeze of lubricant in bearings), there is a general agreement, confirmed in different applicative areas, that a lumped parameter single degree of freedom (SDOF) system is adequate to represent most of the aspects of the dynamic behavior ([10]; [11]; [12]; [13]). In other words a simple kinetostatic model is, in most of the practical situations, sufficient for determining the condition of gross follower jump due to inadequate spring force and/or preload, according with our experimental approach.

![Cam Diagram]

(a) A simple view of the experimental cam-follower test bench

![Distance Diagram]

(b) Distance in mm between cam/flywheel and follower axis

Figure 1.4: Experimental cam-follower test bench

**Cam**

The cam is a specially shaped piece of metal or other material, arranged to move the follower in a controlled fashion. In other words the cam profile can be understood as a control action over follower state [14]. Follower's motion may be either rotational or translational, and often is also coupled to a spring that maintains contact between the components (cam and follower) acting like a closing external force.

The first task faced by cam designers, when presented with a timing diagram, is to select the mathematical functions to be used in order to define the motion of followers. In our experi-
mental context, mainly two different types of cam motions are considered: a continuous periodic sinusoidal-like harmonic motion and a discontinuous second derivative time profile, thus resulting in two different geometries for the cam shape (see Figures 1.5 for details). Both the cams can be mounted on the experiment, thus allowing a comparative experimental analysis.

Sizing

Once the cam profile have been defined, the next step is to size the cam. There are two major issues which affect cam size: the pressure angle and the radius of curvature. Both of these involve the base circle radius on the cam ($R_b$), when using flat-faced follower, or the prime circle radius on the cam ($R_p$), when using roller or curved followers.

The base circle and prime circle centers are the center of rotation of the cam (see Figure 1.5(b)). The base circle of a radial cam is defined as the smallest circle which can be drawn tangent to the physical cam surface. All radial cams will have a base circle, regardless of the follower type used.

In our test bench, the minimum radius of both cams is 30mm and the maximum one is 60mm, as shown in Figure 1.5(a). Both experimental cams are made of low-alloy hardened stainless steel (UNI 38NiCrMo4).

Follower

All the followers can be classified on the basis of the type of contact with the cam as: flat-faced, mushroom (curved) and roller type. The roller follower has the advantage of a lower (rolling) friction than the sliding contact occurring in the flat-face or mushroom case, but it can be more expensive.

Since they have a lesser price and they can package smaller, flat-faced followers are often favored for moving the automotive valve trains. Only recently roller followers are applied in engines due to their lower friction.

All the geometrical features of the flat-faced like follower built for our experimental set-up are shown in Figure 1.6.

Flywheels

In the experimental set-up the motion is ensured by an electrical motor that provides the torque necessary for rotating the cam with a prefixed velocity profile. In the cam-follower system large variation in acceleration, due for example to impacts occurrence, can cause significant oscillations in the torque required to drive the system at a constant or near constant speed. For this reason a very high peak torque may be needed, while the average torque over the cycle, due mainly to losses and external work done, may often be much smaller than the peak torque.

So, unless servomotors are used, we may need to smooth out these oscillations in torque profile during the cycle. In this way it is possible to design the motor so as to deliver only the average torque rather than the peak torque. To this end a purposely designed and sized flywheel is added to the system.

Both the cams are provided with a purposely balanced flywheel that increases the inertia of the whole part and makes its centroid to be on the rotation axis. The flywheel assures a low sensitivity of the cam angular speed to the disturbances, represented by impacts with the
(a) radius measures in millimeters $mm$

Continuous periodic cam shape

Discontinuous second derivative cam shape

(b) rotating axis

Figure 1.5: Cam profiles
Figure 1.6: Design details of the follower
follower, while, balancing the whole part with respect to its rotation axis, reduces the loss of
energy and the bearings stress. Details on the design of the cams with balanced flywheels are
shown in Figure 1.7.

The mechanical set-up

According with the mechanical design of all its components, a mechanical prototype of the cam-
follower system was built as shown here in Figure 1.8.

1.2.2 Motion and measurement hardware

In order to acquire the evolution of all the physical quantities involved in the system and, in the
future, to make the real-time control of the process, it is necessary to embed into the rig sensors,
actuators and specific hardware/software for the signals acquisition and manipulation. In what
follows we describe all the components that complement the mechanical set-up.

The whole experimental set-up developed to analyze nonlinear behavior of the simplified
cam-follower system, is shown in Figure 1.9.

Driving motion system

The driving of cam motion according to a certain angular velocity profile is achieved by using a
SANYO-DENKI® Q series servo system (see Figure 1.10(a)), composed of a 1.5 KW brushless
motor (model P20B10150DXS00M) and an inverter (servo amplifier/controller) model QS1A05.

Sensing

Three main signals are currently measured in the proposed cam-follower system: position and
velocity of cam and position of follower. Obviously, in the future other sensors can be added to
the system to acquire a larger number of different signals, if required.

By employing an encoder sensor located inside the motor (look at Figure 1.10(b) for a graph-
ical description of functional principle) and using the internal processing of signals by means of
the inverter circuit, the measure of cams position is obtained with a resolution of 5000 pulses
per revolution [15].

In a similar way, by using a RI-58 D type HENGSTLER® incremental encoder (see Figure
1.10(c)) located on the rotational axis of the follower, its position can be measured with a
resolution of 10000 pulses per revolution [16].

Acquisition of data

All signals produced by the sensors and the input/output control commands related with inver-
ter process, are managed using the integrated data acquisition system dSPACE® ACE-kit
ACE1104CLP (see Figure 1.10(d)). The device combines a powerful hardware platform with ad-
vanced software resources (including developers code environment extensible for operation with
MATLAB® Simulink®), thus allowing an interactive way to deal with all the system variables
[17].

1.3 Mathematical model derivation [1]

The experimental activity has to be complemented by the analytical and numerical study of the
dynamics that characterize the system under analysis. For this reason in this section we develop
Figure 1.7: Cam details measured in mm
Figure 1.8: Set-up after mounting all the mechanical components

Figure 1.9: Experimental set-up
(a) Servo system employed to cam motion

(b) Structural schematic of incremental encoder

(c) Incremental encoder attached to follower

(d) Integrated acquisition system

Figure 1.10: Motion and measurement hardware
a mathematical model for the experimental rig.

We are interested in the dynamics of the follower, thus the cam can be modelled as an external input acting directly on the follower. This input is a non linear forcing component. The follower motion is constrained to a phase space region bounded by the cam angular position. The non conservative Newton restitution law is used to model the impacts. Under these assumptions, for the mathematical description of the test bed, we must to model:

- the free-body motion of the follower;
- the equation for the contact;
- and the restitution law.

For derivation of this set of equations, the following notation will be employed accordingly with the schematic on Figure 1.11:

- \( \theta_f \), angular follower position;
- \( d_h \), half height of the follower;
- \( K \), stiffness of the spring;
- \( J \), moment of inertia of the follower;
- \( d_0 \), length of the spring when it isn’t applied any force;
- \( B, P \) and \( d_1 \), respectively extreme points and length of a mechanic element which doesn’t allow the spring to rotate when the follower rotates;
- \( A \), point with coordinate \((x_A, y_A)\) where the spring is hooked;
- \( E \), intersection between the straight line parallel to \( y \)-axes passing through \( A \) and the straight line passing through the origin with slope \( \tan(\theta_f) \);
- \( \rho \), distance between the origin and \( E \);
- \( d_2 \), distance between \( P \) and \( E \).

In order to obtain a description of the system behavior, we use a Lagrange approach defined in terms of the potential and kinetic energies of the system, respectively named by \( U \) and \( T \), as a function of the Lagrange variables \( \theta_f \) and \( \dot{\theta}_f \).

In particular the system equations can be derived by solving:

\[
\frac{d}{dt} \left( \frac{\partial L}{\partial \dot{\theta}_f} \right) - \frac{\partial L}{\partial \theta_f} = 0 \tag{1.1}
\]

with \( L \) relating the Lagrange function, defined as:

\[
L = T + U \tag{1.2}
\]

and the kinetic energy is:

\[
T = \frac{1}{2} J \dot{\theta}_f^2 \tag{1.3}
\]
So, if \( f_e \) corresponds to the elastic force exhibited by the spring, the potential energy of the system could be defined as:

\[
\delta U \triangleq f_e^T \delta B
\]

\[
= K \begin{bmatrix} y_A - y_E - d_0 - d_1 - d_2 \end{bmatrix} \begin{bmatrix} 0 & 1 \end{bmatrix} \delta \begin{bmatrix} x_B \\ y_B \end{bmatrix}
\]

\[
= K \begin{bmatrix} y_A - y_E - d_0 - d_1 - d_2 \end{bmatrix} \begin{bmatrix} 0 & 1 \end{bmatrix} \delta \begin{bmatrix} x_B \\ y_E + d_1 + d_2 \end{bmatrix}
\]

\[
= K \left[ (y_A - d_0) - (y_E + d_1 + d_2) \right] \delta (y_E + d_1 + d_2)
\]

\[
= \delta \left[ -\frac{1}{2} K \left[ (y_A - d_0) - (y_E + d_1 + d_2) \right]^2 \right]
\]

Then, by integrating equation (1.5), the explicit expression of \( U \) is obtained as:

\[
U = -\frac{1}{2} K \left[ (y_A - d_0) - (y_E + d_1 + d_2) \right]^2
\]

where

\[
d_2(\theta_f) = \frac{d}{\cos \theta_f}
\]

Furthermore, notice that the distance \( \rho \) is not constant, but if the spring does not rotate (while the follower is in motion) this implies that \( x_E = x_A \) and, in this case, it can be derived by simple geometric considerations as:

\[
y_E(\theta_f) = x_A \tan(\theta_f)
\]

### 1.3.1 Free-body motion of the follower

From the solution of equation (1.1), having in mind the definitions (1.3) (1.6) and using the expressions (1.7), (1.8) we now have the following free-body motion equation:

\[
J \ddot{\theta}_f + K \left( x_A \tan(\theta_f) + \frac{d}{\cos(\theta_f)} - (y_A - d_0 - d_1) \right) \left( \frac{x_A}{\cos^2(\theta_f)} + \frac{d \sin(\theta_f)}{\cos^2(\theta_f)} \right) = 0
\]
1.3.2 Equation for contact

The contact equation can be obtained by modelling the cam as an external input acting directly on the follower. Let \( \tilde{\theta}_f(t) \) be the angular position of the follower when the two bodies (cam and follower) are in contact, then the torque \( q \) provided by the cam has to be such that \( \theta_f = \tilde{\theta}_f \). Including the external forcing of the cam \( q \) and imposing \( \theta_f = \tilde{\theta}_f \), equation (1.9) becomes:

\[
q(t) = J\ddot{\theta}_f(t) + K \left( x_A \tan(\tilde{\theta}_f(t)) + \frac{d}{\cos(\tilde{\theta}_f(t))} - (y_A - d_0 - d_1) \right) \left( \frac{x_A}{\cos^2(\tilde{\theta}_f(t))} + d \frac{\sin(\tilde{\theta}_f(t))}{\cos^2(\tilde{\theta}_f(t))} \right) \tag{1.10}
\]

Thus the dynamic equation during permanent contact is:

\[
J\ddot{\theta}_f + K \left( x_A \tan(\theta_f) + \frac{d}{\cos(\theta_f)} - (y_A - d_0 - d_1) \right) \left( \frac{x_A}{\cos^2(\theta_f)} + d \frac{\sin(\theta_f)}{\cos^2(\theta_f)} \right) = q(t) \tag{1.11}
\]

1.3.3 Restitution law

At this stage of the model derivation, the restitution law has to be found. Firstly, we define the following primitives (as depicted in Figure 1.12):

- \((\hat{l}, \hat{n})\) is the reference system attached to the follower;
- \(\Sigma\) is the boundary of the follower that becomes in contact with the cam. The equation of \(\Sigma\) in the \((x, y)\) reference system can be written as:

\[
y = \tan \theta_f - \frac{d}{\cos(\theta_f)} \tag{1.12}
\]

- \(C\) is the point of the cam nearest to the surface \(\Sigma\). It has coordinates \((x_c, y_c)\) in the reference system \((x, y)\);
- \(h\) is the distance between \(C\) and \(\Sigma\). It is straightforward that \(h\) is zero if there is contact (in this case \(\Sigma\) is the tangent to the cam at point \(C\));
- \(F\) is the point belonging to the follower that will impact on the cam. This point is \((l, -d)\) in the coordinate system \((\hat{l}, \hat{n})\) and \((x_f, y_f)\) in the coordinate system \((x, y)\). The point \((x_f, y_f)\) coincides with \((x_c, y_c)\) when the impact occurs.

Let \(t_k\) be the time instant when a generic impact occurs, then the non conservative Newton restitution law at such a time is:

\[
\dot{h}(t_k^+) = -r\dot{h}(t_k^-) \tag{1.13}
\]

where \(r\) means for restitution coefficient.

From equation (1.13), if the velocity of the point \(C\) is continuous, we write:

\[
\nabla \Sigma \cdot \dot{F}(t_k^+) = \nabla \Sigma \cdot \dot{C}(t_k) - r\dot{h}(t_k^-) \tag{1.14}
\]

It is now necessary to rewrite the Newton restitution rule as a function of the follower angular position \((\theta_f)\), the cam angular position \((\theta_c)\) and the angular velocities \((\dot{\theta}_f\) and \(\dot{\theta}_c)\).

To this aim, we first derive the equations that describe \(h\) and \(\nabla \Sigma \cdot \dot{C}\) as a function of \(x_c, y_c, \theta_f\) and the relationship between \(\theta_f\) and \(\nabla \Sigma \cdot \dot{F}\). The distance \(h\) between the straight line passing
through the point of coordinates \((0, \frac{-d}{\cos(\theta_f)})\) in the \((x,y)\) plane with angular coefficient \(\tan(\theta_f)\), and a generic point \(C = (x_c, y_c)\), corresponds to:

\[
h(x_c, y_c, \theta_f) = \sin(\theta_f)x_c - \cos(\theta_f)y_c - d
\]

thus, its time first derivative \(\dot{h}\) follows that:

\[
\dot{h}(x_c, y_c, \theta_f) = \sin(\theta_f)\dot{x}_c - \cos(\theta_f)\dot{y}_c + (\cos(\theta_f)x_c + \sin(\theta_f)y_c)\dot{\theta}_f
\]

Since \(\nabla \Sigma = \begin{bmatrix} -\sin(\theta_f) & \cos(\theta_f) \end{bmatrix}\), it is easy to verify:

\[
\nabla \Sigma \cdot \dot{C} = -\sin(\theta_f)\dot{x}_c + \cos(\theta_f)\dot{y}_c
\]

Note that the point \(F\) has coordinates \((x_f, y_f)\) related to \((l, -d)\) by the following rotation matrix:

\[
R(\theta_f) = \begin{bmatrix} \cos(\theta_f) & -\sin(\theta_f) \\ \sin(\theta_f) & \cos(\theta_f) \end{bmatrix}
\]

From simple algebraic manipulations, it can be shown that:

\[
\nabla \Sigma \cdot \dot{F} = \begin{bmatrix} -\sin(\theta_f) & \cos(\theta_f) \end{bmatrix} \begin{bmatrix} -\sin(\theta_f)l + \cos(\theta_f)d \\ \cos(\theta_f)l + \sin(\theta_f)d \end{bmatrix} \dot{\theta}_f = l\dot{\theta}_f
\]

Then, when an impact occurs, it follows \(F \equiv C\) and thus:

\[
l = \bar{\dot{F}}^T \begin{bmatrix} x_c \\ y_c \end{bmatrix} = \cos(\theta_f)x_c + \sin(\theta_f)y_c
\]

Substituting in equation (1.14) the expressions given by (1.16),(1.17),(1.19) and (1.20), we obtain:

\[
\dot{\theta}_f(t_k^+) = -r\dot{\theta}_f(t_k^-) + \left(1 + r\right)\frac{\cos(\theta_f)\dot{y}_c - \sin(\theta_f)\dot{x}_c}{\cos(\theta_f)x_c + \sin(\theta_f)y_c}
\]

Figure 1.12: Graphical description of the cam-follower system before the impact occurs.
Figure 1.13: Graphical description of the cam-follower system when in contact.

At this point it is necessary to express the coordinates of C in the \((x, y)\) plane, namely \((x_c, y_c)\), as a function of the cam rotation angle \(\theta_c\). The restitution law (1.21) can be easily rewritten as a function \(\Gamma\) of the variables \(\left(\theta_f, \dot{\theta}_f, \theta_c, \dot{\theta}_c\right)\).

In order to find the expression for \(\Gamma\), we refer to Figure 1.13 which gives a graphical description of the cam and follower geometry when they are in contact, and give the following definitions:

- \(\theta_c\), angular position of the cam measured as the relative rotation of the coordinate system \((\hat{x}_c, \hat{y}_c)\) with respect to \((\hat{x}, \hat{y})\);
- \((x_0, y_0)\), coordinates of the rotation center of the cam in the reference system \((x, y)\);
- \((\hat{x}, \hat{y})\), reference system obtained by translating the origin of the axes \((x, y)\) to \((x_0, y_0)\);
- \((\tilde{x}_c, \tilde{y}_c)\), reference system pivoted at the cam with origin \((x_0, y_0)\).

For all \(\theta_c\), if the follower is directly in contact with the cam surface, \(\Sigma\) can be represented in the plane as the tangent straight line to the cam at the contact point. We can determine the angle of the follower in contact conditions \((\theta_f)\) simply by comparing the generic tangent straight line to the cam with a generic expression for \(\Sigma\).

In the reference system \((\tilde{x}_c, \tilde{y}_c)\), it is always possible to express the cam shape in a parametric form as:

\[
\begin{align*}
\tilde{x}_c &= \tilde{f}(\psi) \\
\tilde{y}_c &= \tilde{g}(\psi)
\end{align*}
\]  

(1.22)

with the parameter \(\psi\) chosen such that \(\psi \in [\psi_1, \psi_2]\).

For instance, if the circular cam shape is considered, we have:

\[
\begin{align*}
\tilde{x}_c &= e + \mu \cos(\psi) \\
\tilde{y}_c &= \mu \sin(\psi)
\end{align*}
\]
where \( e \) is the distance between the center of the circle of radius \( \mu \) and the rotation axis of the cam. In this example \( \psi \in [0, 2\pi] \).

Now, in the coordinate system \((\hat{x}_c, \hat{y}_c)\), under variation of \( \psi \), any straight line tangent to the cam can be described as:

\[
\begin{align*}
\hat{y}_c &= m(\psi)\hat{x}_c + w(\psi) \\
\frac{\partial}{\partial \psi}(m(\psi)) &= \frac{\partial}{\partial \psi}(w(\psi)) \\
w(\psi) &= \hat{y}_c(\psi) - m(\psi)\hat{x}_c(\psi)
\end{align*}
\]

or in parametric form:

\[
\begin{aligned}
\hat{x}_c &= \nu \\
\hat{y}_c &= m\nu + w
\end{aligned}
\]

where, for the sake of simplicity, we dropped the dependence of \( m \) and \( w \) upon \( \psi \).

Applying the rotation matrix \((1.27)\) to the vector \((1.26)\), it is possible to obtain after simple computations the equation of the generic straight line tangent to the cam in the reference system \((\hat{x}, \hat{y})\) by:

\[
R(\theta_c) = \begin{bmatrix} \cos(\theta_c) & -\sin(\theta_c) \\ \sin(\theta_c) & \cos(\theta_c) \end{bmatrix}
\]

\[
\hat{y} = \frac{\sin(\theta_c) + m\cos(\theta_c)}{\cos(\theta_c) - m\sin(\theta_c)} \hat{x} + \frac{w}{\cos(\theta_c) - m\sin(\theta_c)}
\]

Considering the offset between the system \((x, y)\) and \((\hat{x}, \hat{y})\), as in Figure 1.13, it follows that:

\[
y = \frac{\sin(\theta_c) + m\cos(\theta_c)}{\cos(\theta_c) - m\sin(\theta_c)} x + \frac{w}{\cos(\theta_c) - m\sin(\theta_c)} + y_0 + \frac{\sin(\theta_c) - m\cos(\theta_c)}{\cos(\theta_c) - m\sin(\theta_c)} x_0
\]

By comparing the generic expression \((1.12)\), for \( \Sigma \) in the \((x, y)\) plane, with equation \((1.29)\), it is straightforward to have:

\[
\tan \theta_f = \frac{\sin(\theta_c) + m(\psi)\cos(\theta_c)}{\cos(\theta_c) - m(\psi)\sin(\theta_c)}
\]

\[
-\frac{d}{\cos(\theta_f)} = \frac{w(\psi)}{\cos(\theta_c) - m(\psi)\sin(\theta_c)} + y_0 + \frac{\sin(\theta_c) - m(\psi)\cos(\theta_c)}{\cos(\theta_c) - m(\psi)\sin(\theta_c)} x_0
\]

It is now possible to select \( \hat{\theta}_f(\theta_c) \) and \( \hat{\psi}(\theta_c) \) that ensure the contact, as the cam rotates, simply by solving equations \((1.30)\) and \((1.31)\). Finally, the expressions of the contact point coordinates \((x_c, y_c)\), as functions of the cam angular position \( \theta_c \) corresponds to:

\[
\begin{aligned}
\{x_c(\theta_c) = \hat{\psi}(\theta_c))\cos(\theta_c) - \hat{\psi}(\theta_c))\sin(\theta_c) + x_0 \\
y_c(\theta_c) = \hat{\psi}(\theta_c))\sin(\theta_c) + \hat{\psi}(\theta_c))\cos(\theta_c) + y_0
\end{aligned}
\]

The model is then represented by equations \((1.11)\), \((1.21)\) and \((1.32)\).
1.4 Simulation results

A preliminary numerical investigation of a simplified cam-follower model, is depicted in Figures 1.14, 1.15 and 1.16 [8]. The main features observed here, are detachment between cam and follower at $\omega_c \approx 150$ rpm, chattering ($\omega_c \approx 200$ rpm) and transitions to seemingly aperiodic behavior (e.g. $\omega_c \approx 250$ rpm).

![Stroboscopic bifurcation diagram](image)

Figure 1.14: Stroboscopic bifurcation diagram obtained by sampling the states at $\Pi_s = \pi/2$. The cam rotational speed is varied $\omega_c \in [118, 500] \text{ rpm}$

1.5 Preliminary experimental results

In what follows we present some preliminary experimental results based on the cam-follower test-bench, aimed to show that the experimental activity is a fundamental tool to understand and characterize the onset of unwanted dynamics.

At this stage of the project, rather than presenting an exhaustive experimental analysis, we shall seek to look for evidence of complex dynamics due to the occurrence of collisions between the cam and the follower. As the angular velocity of the cam increases, we find that the cam-follower system of interest exhibits bifurcations including sudden transitions from periodic solutions to chaos. Many of these phenomena are found to involve multi-impacting behavior and chattering.

The practical methodology used to drive the experiment, acquire the data and the consequent analysis of the experimental data is described in what follows.

1.5.1 Methodology used

We observed that the parameter that mainly characterize the dynamics of the system is the cam velocity $\omega_c$ (according with the experience coming from practical applications). As a con-
Figure 1.15: Example for simulated multi-impacting behavior

Figure 1.16: Example for simulated chaotic behavior
sequence, the experimental framework developed here basically consists of the analysis of the evolution of the follower (periodical position measures) as a function of $\omega_c$.

![Cam velocity graph](image)

**Figure 1.17:** Example of a cam velocity input driving the experiments

Taking advantage of the software integrated tools (section 1.2.2), a routine consisting of a predefined waveform for cam velocity variation (see Figure 1.17) is applied to the system, while at the same time the positions of the cam and the follower are measured in real-time and acquired by linking Simulink® with the dSPACE® Control Desk platforms (Figure 1.18).

Finally, a postprocessing analysis, that takes into account all the theoretical implications, generates, as preliminary experimental results, time series and bifurcation diagrams. The whole procedure is schematically summarized in Figure 1.19.

1.5.2 Cam profile approximation

The cam profile that drives the follower motion has been derived by an automated numerical calculation that returns the cam profile as a function of the angular position identified on experimental data.

The identification has been carried on by using the curve fitting toolbox available in MATLAB and working with a subset of data corresponding to one revolution of the cam at steady steady low velocity. This condition obviously ensures the continuous contact between the cam and the follower. The estimation procedure provides the cam profile, $f(\cdot)$, as a function of the cam angular position, $\theta_c$, as

$$ f(\theta_c) = \theta_f = \sum_{i=1}^{7} a_i e^{-\left(\frac{\theta_c - b_i}{c_i}\right)^2} $$

(1.33)

where $\theta_c$ and $\theta_f$ corresponds respectively to the cam and the follower position measures, and the coefficients $a_i$, $b_i$, and $c_i$ are estimated numerically. As an example, for the circular shaped cam, the estimated coefficients values are related in Table 1.1.
(a) Simulink® interface

(b) dSPACE® monitoring environment

Figure 1.18: Software tools used to manipulate motion and measurement devices.
Figure 1.19: Graphical description of employed methodology.

<table>
<thead>
<tr>
<th>i value</th>
<th>$a_i$</th>
<th>$b_i$</th>
<th>$c_i$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
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<td>5.943</td>
<td>9.409</td>
</tr>
<tr>
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<td>1.229</td>
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<td>2.32</td>
</tr>
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<td>3</td>
<td>0.2088</td>
<td>1.19</td>
<td>1.735</td>
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<td>4</td>
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<td>1.844</td>
<td>0.4205</td>
</tr>
<tr>
<td>5</td>
<td>0.0009277</td>
<td>2.345</td>
<td>0.4772</td>
</tr>
<tr>
<td>6</td>
<td>0.03371</td>
<td>2.807</td>
<td>1.22</td>
</tr>
<tr>
<td>7</td>
<td>-0.0002147</td>
<td>0.1693</td>
<td>0.1285</td>
</tr>
</tbody>
</table>
1.5.3 Experimental time series

The experimental time series reported below shows the time history of the cam and the follower angular positions at different values of the cam velocity input. The most representative cases obtained experimentally, are shown below.

Permanent contact - low velocity regime

In Figure 1.20 the time series plot for a constant angular velocity $\omega_c = \dot{\theta}_c = 110$ rpm is shown. In the picture the recorded measures of the follower and the cam follower positions (respectively named as $\theta_f$ and $\theta_c$) are compared. As it can be seen from the plot, the force generated by the tangential acceleration of the cam at the contact point (cam joint [2]) is lower than the one produced by the weight of the follower at the opposed direction, thus ensuring the contact between the cam and the follower [18] and allowing consequently the visualization of perfectly overlapping waveforms for both the time evolutions.

![Time series analysis](image)

Figure 1.20: Time series under contacting condition at 110 rpm of $\omega_c$

It is important to note that permanent contact is experimentally detected up to approximately 125 rpm of $\omega_c$. For this reason, the set of values $\omega_c < 125$ rpm will be denoted as the low velocity regime.

Detachment and impacting behavior

Past the low velocity region, the detachment of the mechanical components is observed. Detachment occurs when the force excited by the cam over the follower at a contact point does not balance the restoring force. The follower then exhibits a free fall motion that could be described by means of traditional kinematic equations. After the first detachment condition, as the velocity increases, the response is characterized by multi-impacting behavior between the cam and the follower which is illustrated in the time evolutions of Figures 1.21(a), 1.21(e), 1.21(e) and 1.21(g).

One way to detect impacts, is by plotting the difference between the cam and follower positions ($\theta_f - \theta_c$), searching for a zero value when an impact occurs with a non-penetration condition [14]. These differences are shown in Figures 1.21(b), 1.21(d), 1.21(f) and 1.21(h).
Figure 1.21: Results for $\omega_c = 135 \ (a, b) \ 140 \ (c, d) \ 145 \ (e, f)$ and $150 \ (g, h)$ rpm
In Figure 1.21 (where \( P(m, n) \) denotes \((nT)\)-periodic orbits with \( m \) impacts per period and \( T \) is the period of the cam rotation [19]), it is possible to note that a bouncing motion of fixed amplitude can be detected at 135 rpm. Notice also that the amplitude of the bouncing is not constant anymore for angular velocities above 150 rpm. For \( \omega_c \in [135, 150] \) rpm, the system instead exhibits a stable oscillation mode. Also, from these results, it is easy to see that the bouncing amplitudes increase ramping up the cam surface, according to Newton restitution law [14] (see for example Figure 1.22).

### Aperiodic solutions

A transition to seemingly aperiodic complex behavior, is observed for velocity values \( \omega_c > 152 \) rpm. Examples of pseudo-chaotic trajectories are given in Figures 1.23 to 1.26 showing aperiodic solutions for \( \omega_c > 155 \).

![Graphs showing aperiodic solutions](image)

(a) pattern repeated each 6 periods  
(b) position differences related  
(c) zoom for pattern  
(d) position difference for zoomed region

Figure 1.22: Results for \( \omega_c = 155 \) rpm

#### 1.5.4 Experimental bifurcation diagram

In order to develop a bifurcation diagram, with \( \omega_c \) chosen as the bifurcation parameter, a set of sampled versions for time series must be derived using a sampling interval of one period at the corresponding velocity for which data was taken; i.e. one sample per revolution [19]. This feature has justification in the fact that experimentally was not possible to obtain an impact
Figure 1.23: Results for $\omega_c = 156$ rpm
Figure 1.24: Results for $\omega_c = 157$ rpm
Figure 1.25: Results for $\omega_c = 158$ rpm
Figure 1.26: Results for $\omega_c = 159$ rpm
period greater that the fundamental period of the cam signal.

In this way, the bifurcation diagram of Figure 1.27 taken with an interval of $\omega_c \in [130, 160]$ rpm, shows the transition between stable and chaotic regimes with a bottleneck region close to 153 rpm.

Examples of trajectories belonging to different regions of this diagram, are given as listed below:

B, Figures 1.21(a) and 1.21(b)
C, Figures 1.21(c) and 1.21(d)
D, Figures 1.21(e) and 1.21(f)
E, Figures 1.21(g) and 1.21(h)
F, Figure 1.22.

The bifurcation diagram, confirms experimentally the transition to aperiodic behavior detected numerically.

![Bifurcation diagram with $\omega_c$ as parameter](image)

Figure 1.27: Bifurcation diagram with $\omega_c$ as parameter

1.6 Discussion

Taking into account the main contents and results included in this chapter, it is possible to conclude that:

- The experimental set-up developed gives useful information to analyze real nonlinear dynamic behavior of a cam-follower system.

- The variation of the cam velocity in the system, induces due to the presence of impacts, the occurrence of nonlinear phenomena (from $\omega_c = 125$ rpm) and chaos (at approximately $\omega_c = 153$ rpm).
- The cam-follower experimental prototype can be an effective test-bench for the design and validation of all the theoretical conjectures on impacting mechanical systems aimed both at the analysis and control.

1.7 Final remarks and ongoing work

To conclude this chapter, some observations and comments related to the experimental tasks performed are summarized:

- The maximum practical value for the cam velocity $\omega_c$ available in the ring, corresponds to 160 rpm. Above this velocity, with the selected spring, a double-impact undesired behavior appears.

- There is a clear need to automatize the generation of stroboscopic diagrams, in order to reduce the postprocessing time increased by amounts of data.

- In order to develop the phase plane analysis and to take into account that the chosen sensors just give information in terms of position, a numerical algorithm for generating the related velocities must be implemented trying to avoid, as it can be possible, the eventual amplification of errors generated by discretization process and/or the electromagnetic induction.

More precisely, ongoing works includes:

- Impact detection
- Phase plane plots
- Poincaré maps
- and experimental study of cam-follower system with the discontinuous second derivative cam-shape
Chapter 2

Mechanical friction oscillator experimental rig

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2.1 Experimental set-up

The mass-spring-friction system is housed in a 800x800x450mm mild steel box-section frame supported by four combi anti-vibration mounts. These mounts consist of bonded metal to rubber in a low profile compact design ideal for mobile applications due to an added “Fail Safe” feature. The mounts are rated to 100kg and have a vertical screw adjuster to allow for flexibility in height adjustment/alignment. Two aluminium swing-arm masses of Xkg are supported via a fixed 70mm diameter aluminium column rigidly connected to the outside frame in a central position at both the top and the bottom. The swing-arm masses are each connected to the column via two SKF single row deep groove ball bearings which are in turn supported and constrained vertically on the fixed column by mild steel thick walled sleave sections. The lower mass is connected directly to the outside frame via four identical linear extension springs under preload. Two milled aluminium low profile box sections are rigidly attached to the top face of the lower mass symmetrically around the central hub such that the top mass can be connected to the lower mass via four more identical springs. The spring constants for the system can be adjusted by altering the springs or adjusting the preload. The springs are designed to always be in tension such that the nonlinear effects of a spring becoming slack is never encountered during experimental testing. Angular position of the swing-arm masses is provided by two independent high precision Renishaw optical angle encoders. These encoders consist of a one-piece low inertia stainless steel ring with axial graduations marked directly onto the periphery and a digital readhead containing interpolation electronics and integrated filtering optics which provide excellent dirt immunity. The large internal diameter allows for flexible integration with the test rig while the low inertia of the ring means minimal effect on torque and speed. Eccentricity and ovality are also corrected by robust and fine adjustments. Specifically, two RESR20USA150 rings with a pitch resolution of 20μm and a nominal internal diameter of 130mm (resulting in an exact line count of 23,600) have been selected with two RGH20X50D00A readheads with output resolution of 1μm. The RS422 industrial standard output configuration has been selected for direct implantation with the digital incremental inputs (inCD of the SubD panel) of the dSpace 1104 expansion board which we use for data acquisition.
On one side of the top mass the sprung-loaded friction contact is connected. Two LBBP (double lip seals and raceway plates) linear ball bearings are used to ensure smooth travel of the vertical alignment shaft. The top of this shaft is connected rigidly to a load cell which then holds the replaceable screw-headed friction contact tip. The forces at the contact point are measured using a Kyowa 3-component force transducer. This allows the forces to be experimentally measured in their resolved components of normal force $F_z$, frictional force $F_x$ and tangential force $F_y$ (should be nominally zero to indicate continuous linear motion). Specifically the LSM-B-200NSA1 model is used for its rated capacity of 200N (safe overload rating of 150%) for all degrees of freedom with a natural frequency of 2.0kHz (far outside the experimental range). These strain-gage transducers are most suitable for model experiment due to their compact and lightweight design and simultaneous measurement of the force components and high performance (nonlinearity of $\pm 0.5\%$ and hysteresis of $\pm 0.5\%$ of full stroke excitation). The unit is powered via an RDP Electronics modular 600 system providing an excitation voltage of 5V DC and amplifying the analogue outputs from their rated 0.5mV/V (1000 x10$^{-6}$ strain) to the 16bit analogue input of the dSpace 1104 expansion board. The sprung-loaded system is to ensure that a consistent level of normal force is applied throughout regardless of rig misalignment and contact point wear.

A top frame, consisting of a 800x800mm mild steel box-section square, is attached to the lower (main) frame via four M12 stud sections at the corners for fine horizontal alignment of the rotating disk contact section. The rotating disk is powered by a Bosch Servodynic AC motor connected to a self-contained Alpha gearbox and is supported over the centre of the top frame by an aluminium plate with $x$ and $y$ axis adjustment. This allows the centre of the disc to be precisely aligned with the centre of the fixed column of the lower frame (the alignment can be checked by observing the magnitude of the tangential force measured by the load cell connected to the friction contact). An aluminium milled hollow shaft section housing two BTV double row deep groove ball with polyamide cages is used to guide and constrain the drive shaft connecting the rotating disk to the motor in order to remove any off-axis loading. The servo motor is the SG-B1.016-060 model defining brushless contacts and Samarium Cobalt (SmCo) magnets with a nominal speed of 6300rpm, torque at standstill of 1.65Nm and nominal power of 0.95kW. The rare-earth materials used to provide the permanent magnetic field also have low inertia providing a very high dynamic response. A high load carrying capacity is achieved by means of direct heat dissipation to the outside and the brushless configuration. An integrated resolves is provided for speed regulation with low torque ripple being reduce by means of sinusoidal motor EMF. Additionally, the compact design of the S series gives a high protection standard (IP 65 housing construction) and a long service life due to the prestressed closed bearings which are permanently lubricated giving maintenance free operation. The motor is powered by a Bosch SM4.7/20-G16 servo module (standard interface) operating at a clock frequency of 10kHz. Electric supply for the servo module is provided by a Bosch ACS 220-C1D 1280 240V power module via a smoothing filter. All modules are then installed in a switch cabinet conforming to IP 54 (with dust filters preceding air entries and exits). The Alpha gearbox is the SP 075-M2-20-010 axially motor mounted model and provides a two stage gear reduction. The unit achieves a reduction ratio of 5 in the first stage (at an efficiency of 97%) and 20 in the second stage (at an efficiency of 94%) to achieve an output torque of 70Nm. The gears are case hardened and finished ground high carbon alloy steel to reduce backlash (up to 4arcmin in the first stage and up to 6arcmin in the second) and ensure long service life. The unit is self contained with IP 64 housing protection and lubricated with Renolin synthetic oil (viscosity ISO VG220). The gearbox output shaft is connected to drive shaft via a Huco-Flex M double stage coupling. Thin pressed steel membranes act as the pivotal members in these couplings resolving torque into simple tensile
stresses in opposing segments to allow for any misalignment or out of balance effect. The speed of the rotating disc is controlled in real-time via the resolver output of the motor via the dSpace DSP (see Section 2.2).

With respect to the fictional contact specifications, the rig has been designed to allow for a varying degree of frictional contact force and for a wide range of coefficients of friction. Both fictional faces can be interchanged — a removable layer is held on the underside of the rotating disc via temporary adhesive while the contact tip connected to the load cell is simply unscrewed and another inserted. This will allow repeatability to be achieved by ensuring correct test conditions prior to experimentation or allows us to observe the change in frictional characteristics as each material wears.

2.2 Real-time control and monitoring

When people talk about real-time, they generally mean “right away” or “fast”. A standard programme, such as Microsoft Word, that a general computer user might use, or a text editor that a system programmer might use needs to be fast and responsive, but if it is delayed now and then it is not that important. Generally, we just want things to happen fast, in development terms this is usually described as low-latency. For general-purpose computer systems, “fast” translates to average case performance. However, fast does not imply real-time.

A real-time system is one that has deadlines that cannot be missed. For example, consider the control of a robot arm that lifts partially assembled automobiles from one assembly station to another. In order to position the arm correctly, the computer must monitor its movement and stop it precisely 5.2ms after it starts. These timing constraints make this a hard real-time system, where average case performance will not do, stopping the arm 7.1ms after it starts one time and 3.4ms after it starts the next is just not acceptable. Even software that should usually meet timing deadlines, such as video drivers, can afford a hitch now and then. A missed video frame will not cause the damage of a missed robot arm control message. In real-time systems literature (for example, [20]) the text editor is considered to be non real-time and the video display would be called soft real-time. Only the robot controller would be called a hard real-time system. The distinctions are as follows:

**Hard real-time**: An operating system is considered to be hard real-time if all time constraints imposed by the external world, so-called deadlines, are strictly met within a predefined tolerance, both for a priori deadlines which can be scheduled, and for sporadic deadlines such as interrupts, i.e. the worst case must be within the tolerance, e.g. a real-time process is scheduled within a tolerance of 1ms any time when it ought to and the interrupt response time for any interrupt issued by a pre-selected device is less than 100µs.

**Soft real-time**: An operating system is considered to be soft real-time if all time constraints imposed by the external world, so-called deadlines, are met in a statistical sense, i.e. the mean value of schedule time deviation is less than a predefined tolerance, e.g. a process is scheduled within a mean tolerance of 1ms, but it may happen, that sometimes the scheduling delay is bigger.

In the 1980s, hard real-time applications were simple enough to be controlled by dedicated, custom, isolated hardware. However, modern real-time applications must control highly complex systems which are far more general and diverse in purpose. Furthermore, as demands for speed and quality of service increase, applications that have never required it before have begun to
require hard real-time support. A CD player that makes a popping sound once in a while is okay for casual listening but is not acceptable for a professional music editing system. The problem is that to deliver the tight worst-case timing needed for hard real-time, operating systems need to be simple, small and predictable. But delivering the sophisticated services that modern applications need is beyond the capabilities of simple, small, predictable operating systems. When you try to put real-time inside a general-purpose operating system, or try to put complex services in a small real-time operating system, you end up with something that does neither task well and where non real-time services can interfere with the execution of real-time services.

Therefore, to implement real-time control we have used a DSP (Digital Signal Processing) processor, specifically the dSpace DS1104 R&D Controller Board running on hardware architecture of MPC8240 (PowerPC 603e core) at 250 MHz with 32 MB synchronous DRAM. This allows the “best of both worlds”; all the development of the experimental algorithms can be done on an existing non real-time PC with all the benefits of a general-purpose computer system and all the hard real-time applications handled separately on the DSP processor. Explicitly, the block diagram-based modelling tool MATLAB/Simulink (which is fully integrated into dSpace architecture) is used; during the compiling phase the algorithm is stringently validated for programming violations before being built into the DSP processor where it is checked for any hard real-time violations. The dSpace companion software ControlDesk is used for online analysis, providing soft real-time access to the hard real-time application on the general-purpose computer. In this way the developer has all the advantages of using a standard PC but can ensure hard real-time constraints.
Bibliography


