

Geometric Analysis of a Dual Clutch Lever-Based Electromechanical Actuator with Application to Actuator Dynamics Modelling

Milan Milutinovic, Vladimir Ivanovic, Josko Deur and Dragutin Scap
University of Zagreb

ABSTRACT

This paper presents a practical and straightforward method of identifying geometry parameters of a cam-like lever-based electromechanical dual-clutch actuator, with application to actuator dynamics model parameterization. The lever-based actuator resembles a cam mechanism in that a movable roller fulcrum, driven by an electromotor through a ball-screw, drives the lever by direct contact along the lever profile. This necessitates the identification of the lever profile geometry in order to accurately model the mechanism dynamics. The identification method is based on the measured basic lever mechanism dimensions, experimentally recorded input-output response of the lever mechanism during unloaded operation, observed geometric constraints satisfied during operation, and common CAD software tools to conduct a CAD-based mechanism synthesis and position analysis. First, the geometry, i.e. shape, of the lever profile in contact with the roller fulcrum is determined by means of kinematic inversion. Next, the geometric parameters of the mechanism motion are determined by geometric analysis. Finally, the method is validated after incorporating results into the overall dynamics model of the electromechanical actuator. Experimental validation confirms the effectiveness of the proposed parameter identification method.

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INTRODUCTION

Dual Clutch Transmissions (DCTs) combine the comfort of torque converter automatic transmissions with the fuel economy and the sporty, responsive and connected drive feel of manual transmissions [1]. The heart of a DCT is the dual clutch, which generally consists of two main subsystems: (i) actuation mechanism and (ii) double clutch assembly [2]. In addition to well established electrohydraulic clutch actuation systems, electromechanical systems have been introduced on the market. The main advantages of the electromechanical actuation solutions include easier integration into the whole transmission system and potentially better total efficiency [3,4,5].

In order to achieve a good accuracy of a dynamic model of a dual clutch lever-based electromechanical actuator [2], which is cam-like in that it comprises an electromotor-driven movable fulcrum roller which drives the lever by direct contact along the lever profile, the lever geometry parameters must be accurately identified [6]. These parameters include those describing (i) the lever motion and (ii) the contact

between the lever and the fulcrum with respect to the fulcrum position. Knowing the lever profile is a prerequisite for determining these parameters.

The objective of identifying the geometry parameters of the given lever mechanism, which is a production component for which no design data is available, can be accomplished in a number of ways. In order to identify the lever profile, one might resort to a photogrammetric or a 3D scanning system. This method might require disassembling the actuator, which is not intended to be disassembled, damaging it and rendering it lost for further use. If pertinent dimensions and the input and output motion of the lever mechanism were known, the lever profile could be determined by using designated motion analysis software. Once the profile is determined, the parameters describing the lever motion during operation can be identified by using designated motion analysis software.

This paper proposes a practical and straightforward method which does not require actuator disassembling or application of specialized motion analysis software. The geometry parameters describing the shape and motion of the lever-based clutch actuator are identified using the measured

basic lever mechanism dimensions, experimentally recorded lever mechanism input and output motions during idling, observed geometric constraints satisfied during operation, and CAD-based mechanism synthesis and motion analysis. In order to record input-output motion response of the actuator, a custom-made test rig [7] is used. The identified lever mechanism geometry parameters are incorporated into the previously developed actuator dynamics model [6]. The model experimental validation results are analysed and compared with the results obtained by using a simpler method that does not involve CAD analysis.

SYSTEM DESCRIPTION

Fig. 1 shows the schematic of the actuator electromechanical system. Electric motor, whose angular position is α_m , drives the system of rollers containing the fulcrum roller, whose displacement is x_f through a precise ball screw-drive with speed ratio i_{sd} . Fulcrum roller in turn drives the actuator lever by direct contact at the lever profile. The lever motion is constrained by the fulcrum, a soft leaf spring attached to the lever and a stiff support, and it is determined by torques produced around the fulcrum by the energy storage springs' force F_s , the engagement bearing force F_b and the support force F_{sup} , where the latter equals zero if there is no contact with the support. The lever-fulcrum contact point, and thus the effective lever ratio $(l_b - l_{cf})/(l_s + l_{cf})$, changes as the fulcrum roller moves (with $l_{ce}(x_f)$ denoting the axial eccentricity of the contact point with respect to fulcrum roller centre). As the fulcrum roller pushes the lever along its profile, the lever is simultaneously displaced radially with respect to screw-drive centre line at the lever-fulcrum contact point by the displacement x_{lf} , and rotated around the lever-fulcrum contact point by the angle α_l . In plain language, the lever is simultaneously lifted by x_{lf} and rotated by α_l at and around lever-fulcrum contact point, respectively. The resulting lever radial displacement at the engagement bearing is x_b , and this is effectively the lever output motion, as it is intended to actuate the clutch through the engagement bearing (not shown). On the other hand, the lever mechanism input motion is given by the fulcrum roller travel x_f . A more thorough system description, focused on detailed dynamics modelling, can be found in [6].

For the purpose of modelling, one must have a knowledge of $x_{lf}(x_f)$, $l_{cf}(x_f)$ and the lever-fulcrum contact angle $\beta(x_f)$. The latter is necessary to determine the screw-drive/motor load as a component of the lever-fulcrum contact force F_n . The prerequisite for determining these parameters is identification of the lever profile.

Fig. 2 features a simplified schematic (a kinematic scheme) of the system retaining only those elements that are necessary for the geometric CAD synthesis and analysis presented in this paper. It is assumed that the fulcrum roller is constrained to move horizontally. As illustrated in Fig. 2, the lever can operate in two possible modes: (i) lever leaned to

support (Mode 1) and (ii) lever in see-saw mode (Mode 2). The mode of operation is determined by the balance of torques around the lever-fulcrum contact point (Fig. 1, [6]). If there were no F_b , the lever would always operate in Mode 1 because of the torque produced by the spring force F_s around the fulcrum.

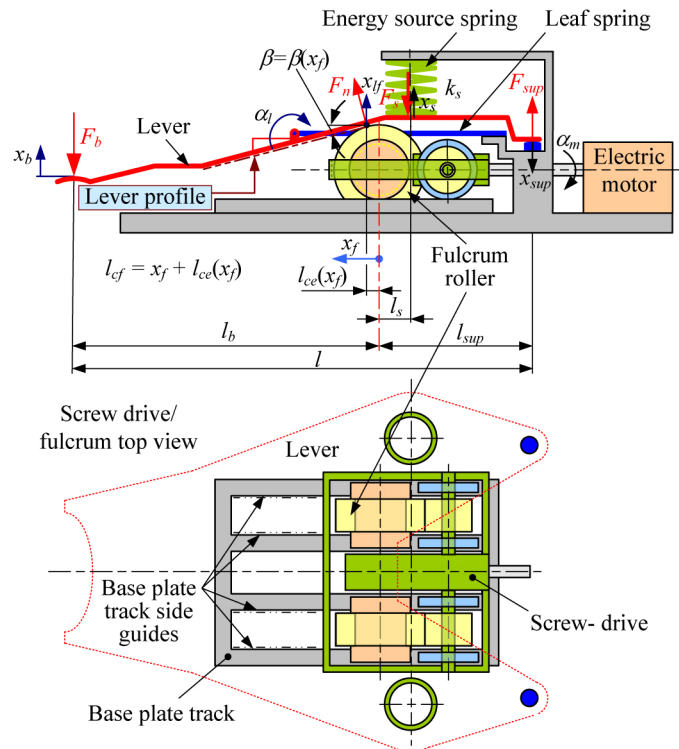


Figure 1. Schematic of the lever based actuator system

Fig. 3 shows the symbols representing the basic actuator mechanism dimensions in its initial position that have to be determined, by measurement or otherwise, in order for the CAD analysis to be conducted. Fig. 3 fully defines initial position of the three movable parts, which are the actuator lever whose motion is already described, the fulcrum roller that travels horizontally and the leaf spring that rotates around point B. The dimensions are shown either explicitly or, for the sake of figure legibility, implicitly. Implicitly shown dimensions are the required relative positions, i.e. horizontal and vertical distances, of the reference points A, B, C and D. A simple rectangular form of the lever is chosen arbitrarily without loss of generality. Namely, it is the relative position of the reference points A, C and D along with lever end radius R_e and that describes the actual shape of the lever as a rigid body (also, part of the lever that leans on the support around point A is indeed flat). This simple kinematic scheme corresponds to the schematic used in subsequent CAD models. This is the initial schematic, and no lever profile is shown yet. All that is currently known about the profile is that its initial point is near the top of the fulcrum roller.

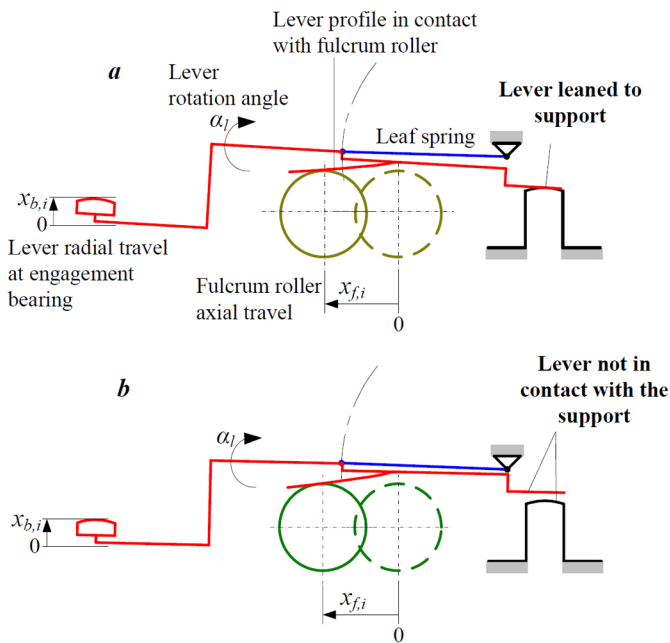


Figure 2. Kinematic scheme of the system illustrating operation in Mode 1 (a) and Mode 2 (b)

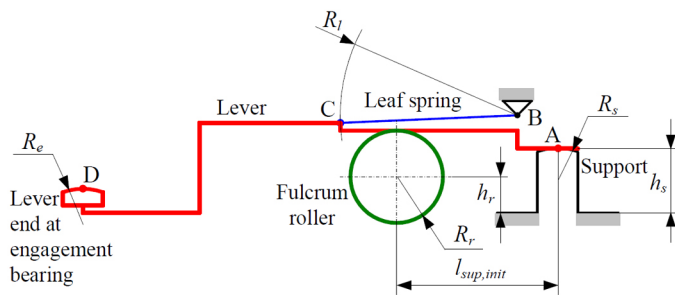


Figure 3. Actuator initial position and lever geometry definition

APPROACH

Fig. 4 illustrates a summarized representation of the proposed geometric analysis-based identification method. Prior to geometric CAD analysis, the actuator dimensions from Fig. 3 have been measured and relation between the fulcrum position x_f and the lever output position x_b experimentally determined for the case of unloaded lever, i.e. for idling operating condition (Fig. 4, left). The idling operating condition has been utilized, because under this condition the lever operates only in Mode 1, for which the lever position is uniquely defined by the geometric constraints only (see previous Section). This facilitates easy reconstruction of the lever position based on the fulcrum position x_f and the engagement bearing lift x_b without a need to directly measure the lever rotation angle α_l , which would also be difficult to practically implement. In the first step of geometric CAD analysis (Fig. 4, middle), the lever profile

$z_p(x_p)$ is determined by using results obtained in the previous phase and the kinematic inversion method [8]. A CAD model that includes main lever geometric constraints is used for that purpose (CAD model 1). In the second CAD analysis step (Fig. 4, right), the parameters $x_f(x_f)$, $l_c(x_f)$ and $\beta(x_f)$ (see Fig. 1), required for the developed dynamic model [6], are determined by analysing the actuator motion with constrained rotation. For that purpose, a full CAD actuator model is used (CAD model 2). This model includes observed geometric constraints during operation and the lever profile results from the first analysis step. The subsequent sections include detailed description of the procedure.

IDENTIFICATION OF BASIC DIMENSIONS AND INPUT/OUTPUT STATIC CURVE

The basic actuator mechanism dimensions in initial position (Fig. 3) were measured using common workshop and laboratory tools, namely digital callipers and a drawing compass. The screw-drive gear ratio i_{sd} was determined by manually turning the actuator spindle (normally driven by the electric motor) while directly measuring corresponding fulcrum wheel travel x_f , again using callipers. The relation between actuator input and output motion, i.e. $x_b(x_f)$ characteristic (Fig. 4, bottom left), was recorded using a custom-made computer-controlled experimental test rig [6, 7] (Fig. 5). The test rig is intended for a more detailed characterization of the actuator system dynamics than was required here [7]. It facilitates a detailed characterization of the actuator dynamics under variable load realized by means of a linear helical spring. Springs of different stiffness can be used, and initial spring load (or freeplay) is adjusted by a preload adjusting mechanism. The lever radial travel at the engagement bearing x_b is recorded directly using a linear variable differential transformer (LVDT) sensor, while fulcrum roller travel x_f is calculated from measured servomotor rotation angle α_m using the determined screw-drive gear ratio i_{sd} . In order to avoid lever deformation during $x_b(x_f)$ recording, the lever was unloaded by removing the helical spring, meaning that the actuator was idling in Mode 1 (Fig. 2). The measured characteristic $x_b(x_f)$, shown in Fig. 4 in the normalized form, constitutes a basis for the CAD lever profile synthesis.

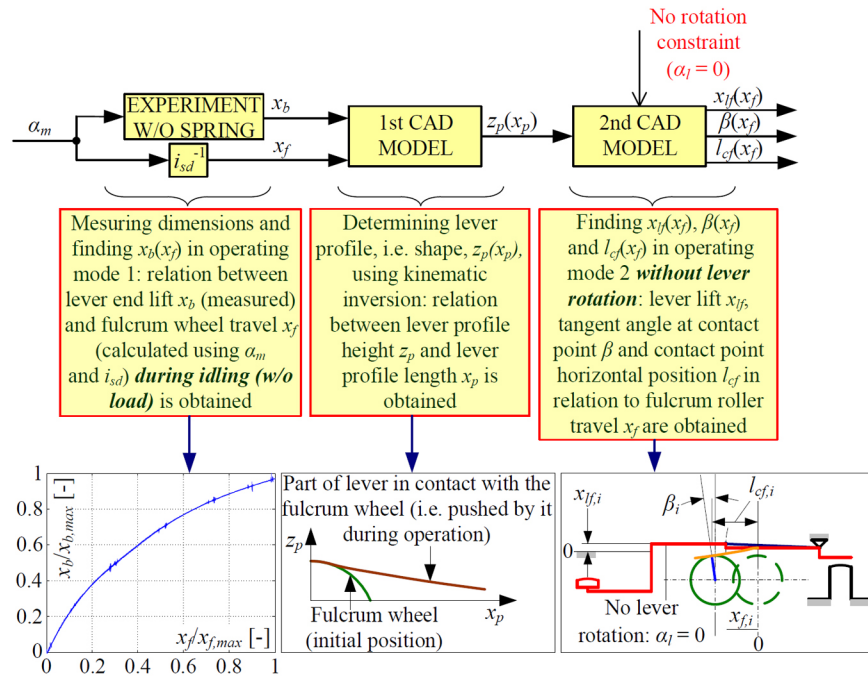


Figure 4. Procedure overview

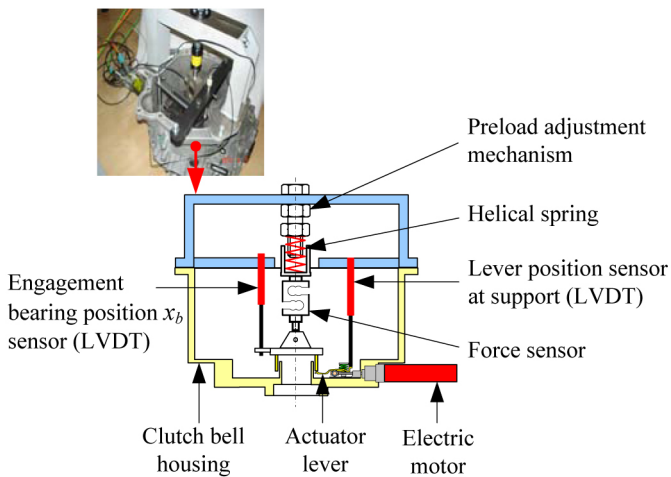


Figure 5. Actuator test rig

LEVER PROFILE SYNTHESIS

In the first step of CAD procedure, a 2D CAD model is made of the actuator in its initial position. The lever is modelled as a rigid body, which is done by properly constraining it using the measured lever dimensions (relative positions of points A, C and D, and radius R_e , all shown in Fig. 3), and by assigning length to its line sections and the perpendicularity constraint between them. The latter two define the shown perpendicular form of the lever, which is arbitrary, and a different set of constraints can be chosen that ensures that the lever is rigid, as long as the reference points

retain their relative position. The fixed base of the mechanism comprises the support and point B.

In Fig. 6a, a model of the mechanism in initial position is repeated for clarity. When the fulcrum roller travels by x_f horizontally, the lever end at engagement bearing lifts by x_b according to the recorded $x_b(x_f)$ characteristic in operating Mode 1 (see Fig. 4).

Fig. 6b shows a characteristic step of the procedure. Observed constraints satisfied during operation in Mode 1 are as follows: (i) the lever is constrained to remain in contact with, i.e. tangent to, the rounded support (e.g. at point A in initial position), (ii) the lever is attached to the leaf spring at point C, and is thus constrained so that the path of point C is an arc defined by the leaf spring length, with the leaf spring modelled as a rigid rod rotating around point B, and (iii) according to the $x_b(x_f)$ relation, the lever end is always tangent to horizontal lines marking successive lever end lifts x_b for corresponding roller travel x_f steps. When the fulcrum roller is in the position $x_{f,i}$, the lever end lifts to reach the corresponding lift $x_{b,i}$, with all of the described constraints satisfied (Position 1). In Position 1, the lever end-lift line tangency is cancelled and instead the roller is constrained to be geometrically fixed to the lever (i.e. to remain in the same relative position with respect to the lever). The lever is then brought back to initial position with the roller fixed to it (position 2, shown dashed).

Repeating the process for x_f ranging from 0 to $x_{f,max}$ in arbitrary increments results in Fig. 1c. The lever profile is represented by the envelope of the shown succession of roller positions, which was obtained by drawing a spline curve tangent to the successive roller positions. The lever profile