The Design of a Novel Pure-Rolling Transmission to Convert Rotational into Translational Motion

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Introduction

Rack-and-pinion transmissions are broadly accepted means of power transmission between a rotating motor and a translating load. Their technology is well established within the framework of gearing [1]. For decades, this transmission went unchallenged, that is, until the advent of highly accurate manufacturing processes that gave rise to more accurate, smoother, and more reliable alternatives, such as *ball screws* and *linear actuators*. In robotics and mechatronics applications, whereby motion is controlled using a piece of software, the conversion of motion from rotational to translational is usually done by alternative means. Of these alternatives, ball screws are gaining popularity, one of their drawbacks being the high number of moving parts that they comprise, for their functioning relies on a number of balls rolling on grooves machined on a shaft; one more drawback of ball screws is their low load-carrying capacity, stemming from the punctual form of contact by means of which loads are transmitted. Linear bearings solve these drawbacks to some extent, for they can be fabricated with roller bearings, their drawback being that these devices rely on a form of direct-drive motor, which makes them expensive to produce and to maintain. Hence the motivation behind the work reported here.

Upon considering the foregoing alternatives to rack-and-pinions, along with their drawbacks, we decided to try to replace these transmissions with cam mechanisms. In the methodology that we adopted, only line contact is considered, for this methodology is based on the three-dimensional version of the Aronhold-Kennedy Theorem, as described extensively in [2]. Furthermore, the cam mechanisms at the core of the transmissions under disclosure are of the translating roller-follower type. A unified methodology on the synthesis of cam mechanisms was introduced in the foregoing reference. This methodology was exploited extensively to develop the software package USyCaMs [3], which proved an invaluable tool in the design of the transmission reported in this paper.

In USyCaMs we introduced a displacement program called *cycloidal-modified*, which can be varied smoothly from cycloidal to linear motion with the aid of a coefficient that changes from unity to zero. The results obtained with this function motivated us to investigate the possibility of transmitting a *linear*—in the algebraic sense—displacement program as needed in rack-and-pinions. This methodology has already led to the development of friction-and-backlash-free speed reducers with virtually unlimited stiffness, as reported in [4]. The transmission reported in this paper, termed Slide-o-Cam, is currently under development at McGill University's Centre for Intelligent Machines, within a research program aimed at highly accurate transmissions for robotics and mechatronics applications. In the context of this program, we envision applications of Slide-o-Cam as a drive for revolute joints using hydraulic pistons as actuators, for Slide-o-Cam is a reversible transmission.

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Mechanism Synthesis

The transmission introduced here, Slide-o-Cam, has the morphology of an indexing cam mechanism (ICM), i.e., a cam mechanism whose follower bears a *periodic* geometry. The follower pattern repeats itself N times per turn of the cam, the ICM at hand thus being said to have N *stages* or *indexing steps*. An ICM is designed so as to allow the production of a *periodic*, nonreversing speed of the follower when the cam rotates at a constant angular speed.

While Slide-o-Cam stems from the concept of ICM, its distinguishing feature is the type of periodic speed produced on the follower, namely, a translational speed that is proportional to the cam speed. As a matter of fact, Slide-o-Cam is reversible, and hence, can be used to convert the translational motion of a motor, e.g., a hydraulic actuator, into the rotational motion of a load, e.g., a revolute joint.

The procedure adopted to synthesize Slide-o-Cam is an instance of the unified methodology developed by the authors to synthesize ICM [2]. In this vein, we are interested in the synthesis of mechanisms composed of four links, whereby the frame, the cam, the follower, and the roller are numbered from 1 to 4, in this order. Moreover, the cam is coupled to the frame via a revolute pair, while the follower is coupled to the same via a prismatic pair; the roller, in the most general case, is coupled to the follower via a cylindrical pair and to the cam via a higher pair, as depicted in Fig. 1. The contact surface between the cam and the roller is a ruled surface on the cam and a circular cylinder on the roller [2]. In particular, when the direction of translation of the follower is at right angles with the axis of rotation of the cam, the coupling between roller and follower becomes a revolute pair. Most applications of rack-and-pinions call for a geometry of this kind, but rack-and-pinions with skew input and output axes are also available. The prototype of Slide-o-Cam that is currently under development and the focus of this paper is of the orthogonal, revolute-roller type.

The cam surface of contact is determined by the position vector $\mathbf{r}$ defining a ruled surface, which is given by

$$\mathbf{r}(\psi,\lambda)=\mathbf{p}(\psi)+\lambda\,\mathbf{e}(\psi) \tag{1}$$

where ψ and λ are the parameters of the ruled surface, $\mathbf{p}(\psi)$ is the position vector of the *generatrix* and $\mathbf{e}$ is a unit vector parallel to the line producing the surface.

Furthermore, the geometric parameters defining the cam mechanism are illustrated in Fig. 1. The notation of this figure is based on the general notation introduced in [2]:

- a_1+a_3 = the distance between the cam axis of rotation and the trajectory of the center of the roller
- α_1 = the angle between the input and output axes
- a_4 = the radius of the roller axis
- ψ = the angle of rotation of the cam
- z_3 = the displacement of the follower
- z_{43} = the translational displacement of the roller with respect to the cam.

The above parameters as well as the contact surface on the cam are determined by both the geometric relations dictated by the Aronhold-Kennedy Theorem in the plane [5] and the relation between the input and the output variables, ψ and z_3, respectively. This methodology is applicable, moreover, to indexing mechanisms, which requires the specification of the number N of index-

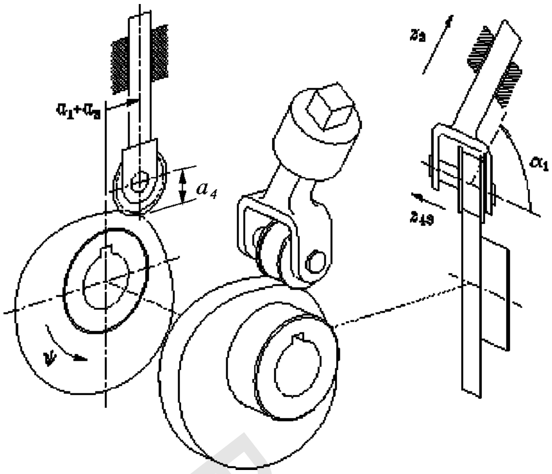

Fig. 1 The geometry of a general cam mechanism with a translating follower

ing steps per turn of the cam. In the case at hand, N is identical to the number of rollers that come into contact with the cam per cam turn. Furthermore, the input-output relation is given by

$$z_3=\frac{\pi-\psi}{N}a_3 \tag{2}$$

The Designed Mechanism

In the discussion below we will make reference to the layout displayed in Fig. 1, which shows a general planar cam mechanism with a translating roller-follower. In that figure, angle ψ is the input variable, the coordinate $z_3=z_3(\psi)$ being the output variable, a prescribed linear function of ψ. Based on Eq. (1), the position vector defining the contact surface of the cam is given as a special case of the general synthesis equations derived in [2].

If we substitute $z_3(\psi)$ as given by Eq. (2) in the above-mentioned synthesis equations, with $\alpha_1=90$ deg, the position vector $\mathbf{r}_c$ of the contact point, for a given value of ψ, becomes

$$\mathbf{r}_c=\begin{bmatrix} k_1\cos\psi+(k_2-a_4)\cos(\psi-k_3)\\ -k_1\sin\psi-(k_2-a_4)\sin(\psi-k_3)\\ \lambda \end{bmatrix} \tag{3a}$$

where coefficients k_i, for $i=1,2,3$, are defined as

$$k_1=-\frac{2\pi a_3}{N}\sin\alpha_1,\quad k_2=\sqrt{(k_1\psi)^2+(a_1+a_3-k_1)^2}$$

$$k_3=\arctan\left(\frac{k_1\psi}{k_1-a_1-a_3}\right) \tag{3b}$$

and, to avoid undercutting, a_3 should be bounded as [2]

$$a_3<\frac{a_1}{1+1/N} \tag{3c}$$

The mechanism thus designed, with $N=8$, $a_1=100$ mm, and $a_3=-81.487$ mm, is displayed in Figs. 2 & 3. Note that the value for a_3 was defined close to the bound given in expression $(3c)$.

The Pressure Angle

The quality of the force transmission in cam mechanisms is measured by means of the pressure angle μ. The pressure angle is defined as that between the direction of the *normal component* of the force applied onto the follower and the direction of the velocity vector of the point of application of the force, on the follower. The expression for the pressure angle of the mechanism at hand is well documented in the literature. A generalized formula, applicable to planar, spherical, and spatial cam mechanisms is derived in [2]. Using these expressions, the pressure angle for Slide-o-Cam is given by $\tan\mu=-(a_1+a_3+z_3')/z_3$, where z_3' is defined as $z_3'\equiv dz_3/d\psi$.

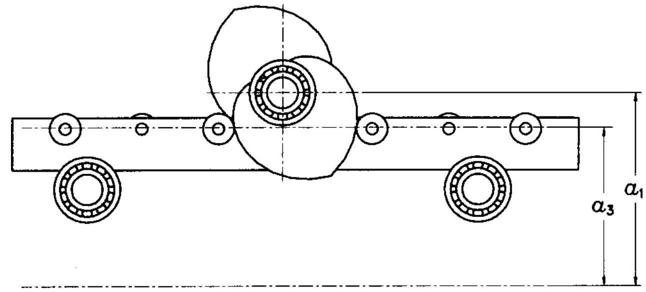

Fig. 2 An instance of Slide-o-Cam

In the design of cam mechanisms, we distinguish two kinds of actuating forces at the contact between the cam and the follower, namely, the force that transmits the motion to the follower and the force that opposes this motion. The action of each of these forces is termed *positive action* (PA) and *negative action* (NA), respectively. Moreover, we call *positive motion* that under which both PA and NA are present in the transmission. Because we designed Slide-o-Cam with positive motion, we have two cams attached to the input shaft and two followers attached to the output slider.

Practical Aspects

The low friction in Slide-o-Cam is to be highlighted as a major advantage of this transmission: low friction means that the life of a transmission with plastic elements can be greatly increased if it is based on Slide-o-Cam. The alternative would be a plastic rack-and-pinion, which is most frequently of the spur type, and hence, exposed to friction and, as a consequence, to excessive wear. Friction losses, of course, cannot be completely eliminated. These are still present in Slide-o-Cam, but confined to the rollers mounted on the follower.

When comparing Slide-o-Cam with rack-and-pinions, notice that the former requires more moving parts than the latter, which undeniably affects the cost of Slide-o-Cam. However, notice that, contrary to rack-and-pinions, which require special machine tools for their cutting, Slide-o-Cam can be machined with general-purpose CNC machine tools. One more item for comparison is the structural stiffness of Slide-o-Cam vs. that of rack-and-pinions: a pinion, or a rack for that matter, is as stiff as the stiffest of its teeth, while the stiffness of Slide-o-Cam is apparent from the es-

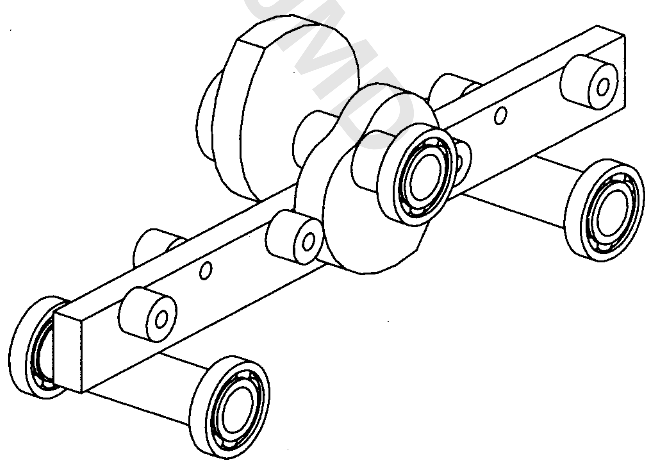

Fig. 3 A 3D rendering of Slide-o-Cam

sentially convex shape of its cams, the flexibility of Slide-o-Cam thus being confined to the roller bearings on the follower. In order to enhance the stiffness of Slide-o-Cam, it is possible to obtain a fully-convex design, if at the expense of a somewhat bigger cam plate. The design shown in Fig. 3 involves a minimum-size cam, which led to a cam with a slight concavity.

Conclusions

We introduced here a novel mechanical transmission, Slide-o-Cam, for the conversion of rotational into translational motion, or vice versa. Hence, this transmission is proposed to replace rack-and-pinions. The morphology of this transmission stems from that of pure-rolling indexing cam mechanisms, which were extensively studied by the authors elsewhere [2]. Hence, Slide-o-Cam is aimed at the elimination of backlash and friction.

Acknowledgments

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