
On The Motion of a Skew Conveyor Module

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Abstract: - The interest in simple but specialized conveyors, capable of laterally transporting objects, is motivated by the ever-increasing use of package sorting facilities. The present study investigates the motion of a platform with a skew-hinged roller, the platform being pulled in translation with constant velocity. The study is progressive in complexity: a kinematic study based on pure rolling motion is followed by a steady-state dynamic study taking into account the sliding friction force and the two rolling friction moments. The dynamic aspect is also investigated considering a typical sliding friction coefficient decreasing with increasing local sliding velocity. The main results are the optimal fixing angle of the roller shaft for maximum lateral displacement, as required by the users of such devices. The kinematic optimal angle is corrected following the dynamic analysis, and the influence of sliding and rolling friction coefficients is clearly emphasized. The possible oscillations of the roller are also discussed.

Keywords: - roller conveyor, rolling-sliding motion.

1. INTRODUCTION

Efficient and flexible material transport is a fundamental requirement in modern manufacturing, warehousing, and intralogistics systems. Conventional conveyor technologies, such as belt and roller conveyors, primarily enable unidirectional motion along predefined paths. While dependable, these systems impose significant constraints on layout adaptability, dynamic routing, and scalability, particularly in environments characterized by high product variability and frequent reconfiguration. As a result, there is growing interest in conveyor solutions that support planar, multidirectional transport within compact footprints.

Horizontal conveyor modules equipped with omnidirectional wheels offer a promising alternative to traditional linear conveyors. Omnidirectional

wheels enable motion in arbitrary directions on a plane without requiring mechanical reorientation of the transported object. This capability arises from the use of passive rollers mounted around the wheel circumference, allowing force components to be generated both longitudinally and laterally. The fundamental concept of omnidirectional motion using angled rollers was introduced by Ilon in the 1970s and has since been widely adopted in nonholonomic mobile robot platforms [1].

Extensive research has been conducted on the kinematics, dynamics, and control of omnidirectional wheeled mobile robots. These studies established mathematical models that relate individual wheel velocities to platform motion, providing the foundation for precise planar motion control [2], [3]. More recently, researchers have begun adapting these principles to stationary

conveyor systems, where omnidirectional wheels are embedded in modular cells forming a continuous horizontal transport surface.

One notable development is the concept of cellular omnidirectional conveyors, in which multiple driven omni wheels are arranged within discrete modules to manipulate packages across a flat surface. Such systems enable real-time redirection, merging, and sorting of items without mechanical diverters or lift-and-transfer mechanisms. Kinematic modeling and control allocation methods have been proposed to determine wheel speed distributions that generate desired object trajectories while respecting actuator constraints [4]. Experimental results demonstrate that omnidirectional conveyor modules can significantly improve routing flexibility and throughput compared to conventional conveyor layouts.

Trajectory planning and performance evaluation of omnidirectional conveyors have also been explored, with particular emphasis on velocity limits, slip behavior, and load-dependent performance. Analytical and simulation-based studies show that appropriate wheel arrangement and control strategies are critical to achieving stable and repeatable object motion across the conveyor surface [5], [6].

The mounting angle of the wheels sets a hard constraint on how axial and lateral forces are distributed, and getting this wrong at the design stage creates problems that no control law can fully compensate for [7]. Knowing where a package is and how it is oriented on the conveyor surface turns out to be non-trivial in practice [8], and tracking it with precision [9]. Control design for these systems is complicated by the fact that the dynamics are nonlinear. Methods like sliding-mode approaches with fuzzy rule bases [10], reinforcement learning [11], and vision-guided controllers [12] have been applied.

Symmetric conveyor layouts have been analyzed in some depth, dealing with problems such as workspace limits, selecting which wheel patterns are kinematically admissible, what happens when an actuator fails, and whether certain motion primitives are even feasible under the constraints [13]. Omnidirectional platforms have been used to study how chassis geometry and drivetrain reduction interact when the platform needs to pivot under load [14], [15], including an analysis of energy consumption [16].

Despite their advances, several challenges remain. These include: optimizing module geometry and wheel orientation, mitigating frictional losses and lateral slip, ensuring robustness under varying

payloads, and developing scalable control architectures for large conveyor networks. Addressing these challenges is essential for the broader adoption of omnidirectional conveyor technology in industrial environments.

The present paper investigates the design, kinematic and dynamic modeling, including vibrations, of horizontal conveyor modules based on skew-mounted wheels. By bridging established omnidirectional wheel theory with conveyor-specific requirements, the proposed work aims to contribute to the development of flexible, high-performance material handling systems for next-generation automation. The main results, such as ratios of lateral displacement vs. axial motion and oscillation characteristics, have direct application for manufacturers and clients of these devices. The importance of rolling friction moments in this approach is distinctly highlighted.

The kinematic model with independent sliding and rolling motions is developed in the second paragraph. The next paragraph extends the study to dynamic steady-state motion, including the influence of rolling moments. The fourth paragraph covers the oscillations due to the elastic limits of the roller's axial motion, and the conclusions are presented in the last paragraph.

2. MECHANICAL MODEL AND KINEMATIC ANALYSIS

The system is modeled under the following assumptions:

- Wheel rollers are ideal and frictionless along their rotation axis.
- Slip occurs only along roller axial directions in first-order kinematic analysis.
- The conveyor base surface is planar and rigid.
- The object of parallelepiped/cuboid shape remains in continuous contact with several active wheels.

Two reference frames are considered (Figure 1):

- a fixed frame $O_0x_0y_0z_0$ which includes the horizontal plane $O_0x_0y_0$ of the conveyor's surface, with O_0x_0 pointing towards the direction of the platform's motion and the O_0z_0 axis vertical, and
- a moving frame $Oxyz$ attached to the platform in translation, with the origin O at the mass center of the roller, the Oy axis along the relative rotation axis of the roller, and the Oz axis vertical.

The conveyor module containing a platform, roller, and shaft is in translational motion along the O_0x_0 axis with a constant velocity v_0 (Figure 1).

Take, for example, the point of contact between

the roller and the base surface, a point situated along the Oz axis in the $O_0x_0y_0$ plane, beneath the mass center O of the roller.

The angle θ between the fixed axis O_0x_0 and the moving axis Ox is fixed. The angular velocity ω of the roller of radius R remains to be determined.

Apparently, the problem has a simple kinematic solution, but the experimental results are not in sufficiently good agreement.

The roller is in transport motion, due to the translating platform with $\bar{v}_t = v_0 \bar{i}_0$ and has also a relative velocity due to the wheel's proper rotation: $\bar{v}_r = -\omega R \bar{i}$. Naturally, if a prismatic object is placed on a linked assembly of such platforms, it will move over the roller along the Ox direction with a relative velocity $|v_{lr}| = |v_{2r}|$ and an absolute velocity v_{2a}

(Figure 2b). As shown in Figure 2, a supplementary hypothesis is required to solve for the absolute velocity. The basic hypothesis of the kinematic model is to accept a pure rolling condition for the roller around its shaft superposed on a sliding motion along the shaft.

In this case, the relative velocity v_{lr} and the component of the transport velocity on the Ox axis v_{tx} must cancel each other, that is (Figure 1a):

$$v_{lr} = v_t \cos \theta \Rightarrow \omega = \frac{v_0 \cos \theta}{R}, \quad (1)$$

which implies for the carried object, a velocity relative to the platform (Figure 2b):

$$v_{2r} = \omega R = v_0 \cos \theta \quad (2)$$

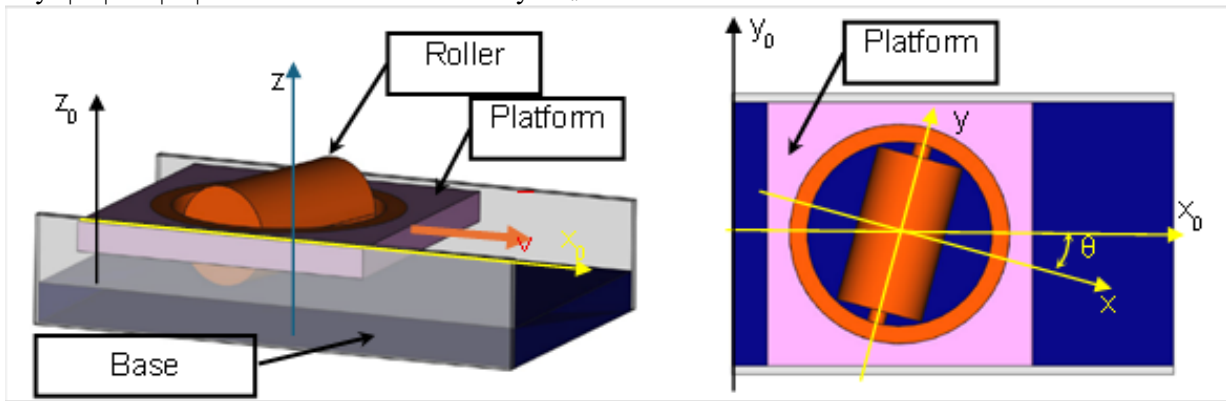


Figure 1. The conveyor module in 3D view (a) and top view (b)

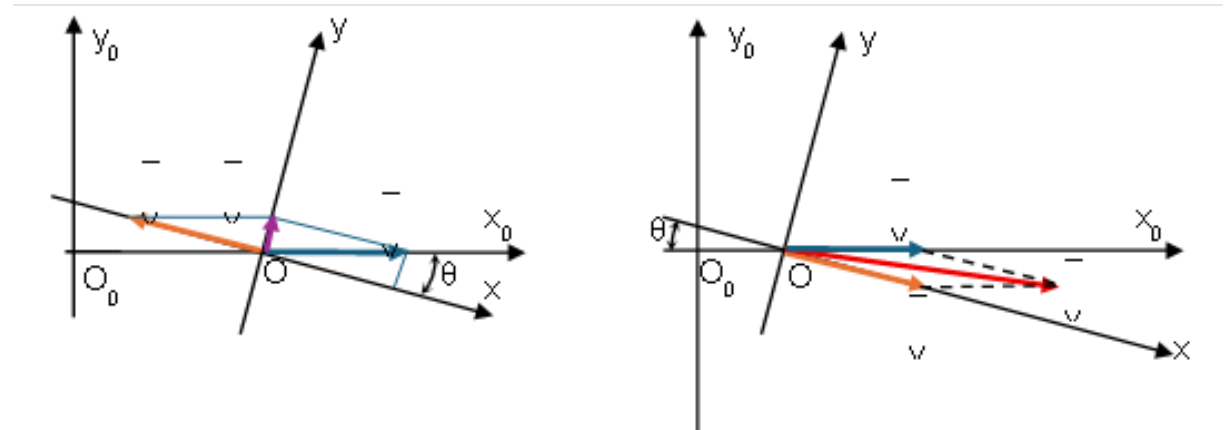


Figure 2. Absolute velocity of contact point 1, roller-platform (a), and contact point 2, roller-upper object (b)

The absolute velocity of the transported object in translation is (Figure 2b) :

$$\bar{v}_{2a} = \bar{v}_{2r} + \bar{v}_t, \quad (3)$$

which can be projected along the $O_0x_0y_0$ fixed reference frame:

$$\begin{cases} v_{2ax} = v_0 + v_{2r} \cos \theta = v_0 + \omega R \cos \theta \\ v_{2ay} = v_{2r} \sin \theta = \omega R \sin \theta \end{cases} \quad (4)$$

This last component of the absolute velocity leads to a lateral displacement of the object along the fixed Oy_0 axis:

$$y_{2a} = \int_0^t v_{2ay} dt = x_0 \sin \theta \cos \theta = \frac{x_0}{2} \sin(2\theta), \quad (5)$$

in which $x_0 = v_0 t$ represents the platform's translation distance covered in arbitrary time t . Simultaneously, the object's displacement along the longitudinal O_0x_0 axis will be:

$$x_{2a} = \int_0^t v_{2ax} dt = x_0 + x_{2r} \cos \theta = x_0 (1 + \cos^2 \theta). \quad (6)$$

From a practical point of view, it is important to determine the ratio of the lateral displacement y_{2a} to the longitudinal one x_{2a} , which corresponds to the ratios of absolute velocity of the object, projected along the fixed axes:

$$\delta = \frac{v_{2ay}}{v_{2ax}} = \frac{y_{2a}}{x_{2a}} = \frac{\sin(2\theta)}{2(1 + \cos^2 \theta)}. \quad (7)$$

The ratio δ is plotted vs. the roller inclination (skew) angle θ on Figure 3. Two particular cases encountered in practical implementations are:

$$\theta = 30^\circ \rightarrow \delta = \frac{\sqrt{3}}{7} = 0.247$$

and

$$\theta = 45^\circ \rightarrow \delta = \frac{1}{3} = 0.333.$$

An important conclusion of the kinematic analysis is that the maximum lateral displacement ratio of $\delta = 0.35$ occurs for $\theta = 54.9^\circ$ between the roller axis and the normal to the platform's motion direction.

According to an industrial beneficiary feedback, the displacement ratio was $\delta = 0.207$ for a wheel skew angle of 30° , which is not in good agreement with the results shown on Figure 3. This discrepancy motivates the continuation of the present study by a dynamic analysis in the following paragraph.

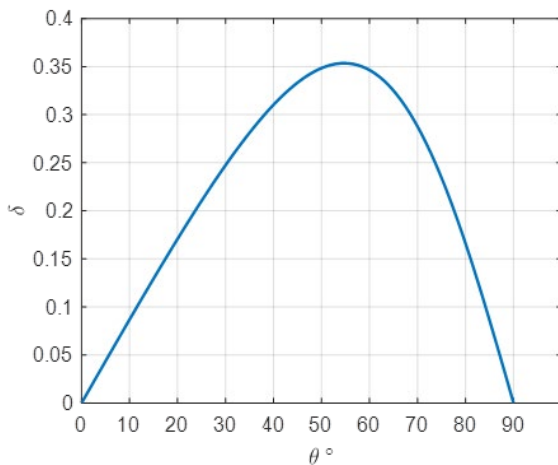


Figure 3. Ratio δ of lateral vs. axial displacement of a body placed on conveyor modules, as a function of roller angle θ .

3. STEADY STATE DYNAMIC MODEL

The roller and its platform, having a total weight $G_l = m_l g$, are placed on the base plate (normal

reaction force N_l , sliding friction F_{fl} , rolling friction moment M_{rl}).

A prismatic object of weight $G_2 = m_2 g$ produces a normal reaction force N_2 , a sliding friction F_{f2} , and a rolling friction moment M_{r2} at the contact with the roller. Reaction forces R_x and R_y are acting between the roller and the platform. The mass of the roller and the platform is denoted by m_l , and the mass of the transported object m_2 ($g=9.81 \text{ m/s}^2$).

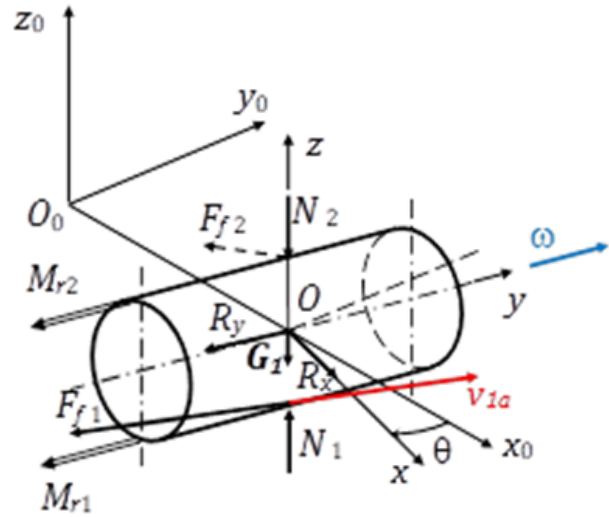


Figure 4. Forces and moments applied on the roller

In the following, index 1 refers to the roller and roller-base-plate contact and index 2 indicates the placed object and the contact roller-upper object.

The kinematic conditions concerning the components of the absolute velocity of the tangent point roller-base, on the moving frame Oxy , are:

$$\begin{cases} v_{1ax} = v_t \cos \theta - v_r = v_0 \cos \theta - \omega R \\ v_{1ay} = v_t \sin \theta = v_0 \sin \theta \end{cases} \quad (8)$$

This generalized condition leads to the following expression for the angle φ between the absolute velocity and the Ox mobile axis:

$$\tan \varphi = \frac{v_{1ay}}{v_{1ax}} = \frac{v_0 \sin \theta}{v_0 \cos \theta - \omega R}. \quad (9)$$

As the friction force F_{fl} (Figure 4) must be opposite to the local absolute velocity between the roller and the fixed base, the ratio of the projections of the friction force on the mobile axis is also

$$\tan \varphi = \frac{F_{f1y}}{F_{f1x}}.$$

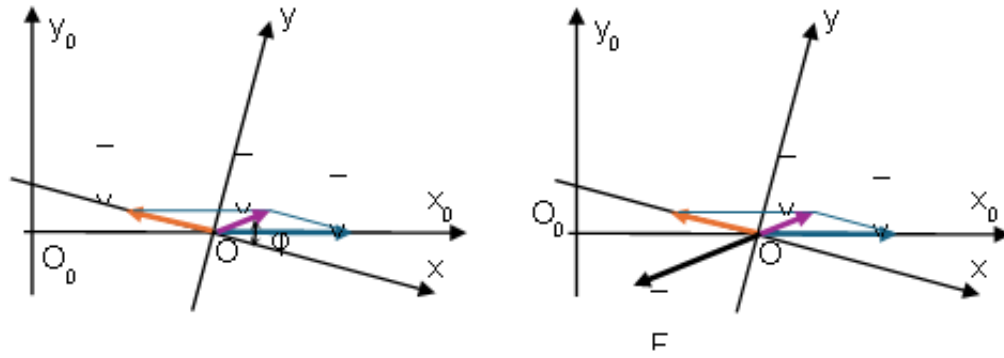


Figure 5. Absolute velocity of the contact point roller-platform in dynamic conditions

Consequently, the dynamic equilibrium conditions are written for the rolling friction moments M_{r1} and M_{r2} :

$$\begin{cases} RF_{f1x} - M_{r1} - M_{r2} = 0 \\ \sqrt{F_{f1x}^2 + F_{f1y}^2} = \mu N_1 \\ M_{r1} = s_1 N_1; M_{r2} = s_2 N_2 \end{cases}, \quad (10)$$

with rolling friction coefficients s_1 and s_2 for the corresponding contact surfaces.

The normal reactions correspond to the supported weights: $N_1 = G_1 + G_2$; $N_2 = G_2$, where G_2 is the weight of the transported object. With these conditions, eq. (10) becomes:

$$\begin{cases} \tan \varphi = \frac{F_{f1y}}{F_{f1x}} = \frac{v_0 \sin \theta}{v_0 \cos \theta - \omega R} \\ F_{f1x} = \frac{s_1 (G_1 + G_2) + s_2 G_2}{R} \end{cases}. \quad (11)$$

From these conditions, one can determine the angular velocity:

$$\frac{\omega R}{v_0} = \cos \theta - \frac{\sin \theta}{\sqrt{\left[\frac{\mu R (G_1 + G_2)}{s_1 (G_1 + G_2) + s_2 G_2} \right]^2 - 1}} \quad (12)$$

The solution is physically acceptable if the denominator $D = \sqrt{\left[\frac{\mu (m_1 + m_2)}{\tilde{s}_1 (m_1 + m_2) + \tilde{s}_2 m_2} \right]^2 - 1}$ is a real number with $\tilde{s}_1 = \frac{s_1}{R}$; $\tilde{s}_2 = \frac{s_2}{R}$, imposing thus a condition for the sliding friction coefficient:

$$\mu > \frac{\tilde{s}_1 (m_1 + m_2) + \tilde{s}_2 m_2}{m_1 + m_2}. \quad (13)$$

The solution provided by eq. (12) indicates a smaller angular velocity than the one obtained from

the kinematic analysis in eq. (1):

$$\omega = \frac{v_0}{R} \cos \theta \left(1 - \frac{\tan \theta}{D} \right) \quad (14)$$

Obviously, the supplementary condition for a positive angular velocity ($\omega > 0$) must be verified:

$$\tan \theta \leq D. \quad (15)$$

The translated prismatic object above the roller will have the components of the absolute velocity given by the same formula (4), in which the angular frequency given by formula (14) is injected. Consequently, the ratio of the lateral displacement y_2 to the longitudinal one x_2 , becomes :

$$\delta = \frac{\sin \theta \cos \theta \left(1 - \frac{\tan \theta}{D} \right)}{1 + \cos^2 \theta \left(1 - \frac{\tan \theta}{D} \right)}. \quad (16)$$

The influence of the rolling friction coefficients, assumed to be in the range $\tilde{s}_1, \tilde{s}_2 \in [0.01, 0.05]$, is presented in Figures 6-8. The roller's angular velocity is shown in Figure 6 as a function of the skew angle θ . Negative values were discarded, thus limiting the range of the skew angle θ . A sliding friction coefficient $\mu = 0.2$ was assumed in Figures 6 and 7. The lateral displacement ratios δ in these conditions are shown in Figure 7.

The maximum lateral displacement diminishes by 12% due to an increase in rolling friction s_1 or s_2 . Moreover, the rolling friction coefficient s_1 between the roller and the base plate reduces the optimal roller axis skew angle by 9° in the case of maximums $\tilde{s}_1 = 0.05$ and $\tilde{s}_2 = 0.05$ compared to the case of minimum rolling friction coefficients: $\tilde{s}_1 = 0.01$ and $\tilde{s}_2 = 0.01$.

Considering a greater sliding friction coefficient $\mu = 0.4$, the influence of the rolling friction coefficients diminishes, as shown in Figure 8. The lateral displacement ratio is less influenced by the

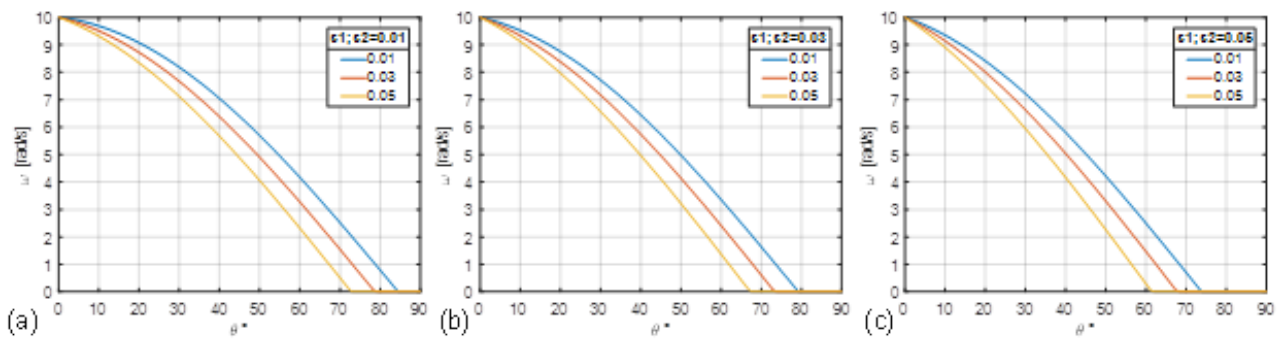


Figure 6. The angular velocity dependence on skew angle θ for $\tilde{s}_1 = 0.01$ (blue), 0.03 (red), 0.05 (orange) in the legends and for selected $\tilde{s}_2 = 0.01$ (a), 0.03 (b), 0.05 (c). Sliding friction coefficient $\mu = 0.2$.

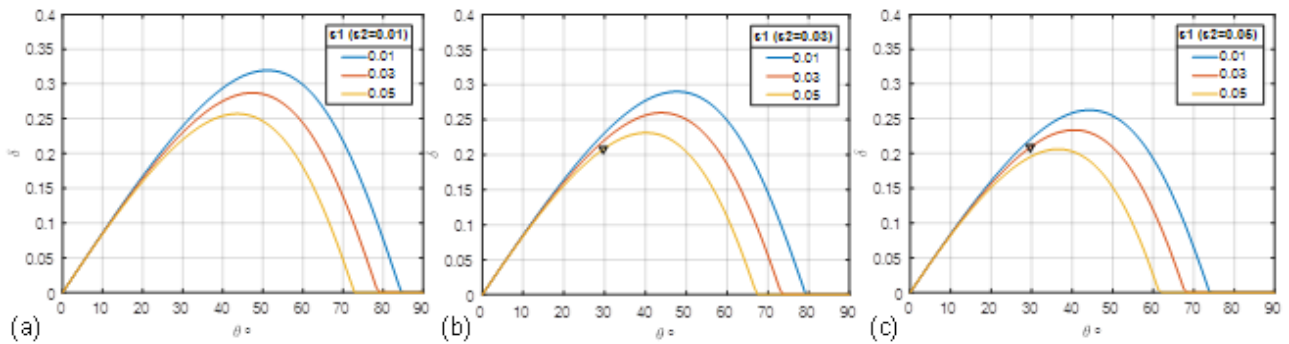


Figure 7. Lateral displacement ratios δ dependence on skew angle θ for $\tilde{s}_1 = 0.01$ (blue), 0.03 (red), 0.05 (orange) for selected $\tilde{s}_2 = 0.01$ (a), 0.03 (b), 0.05 (c). A triangle indicates $\theta = 30^\circ$, $\delta = 0.207$. Sliding friction coefficient $\mu = 0.2$.

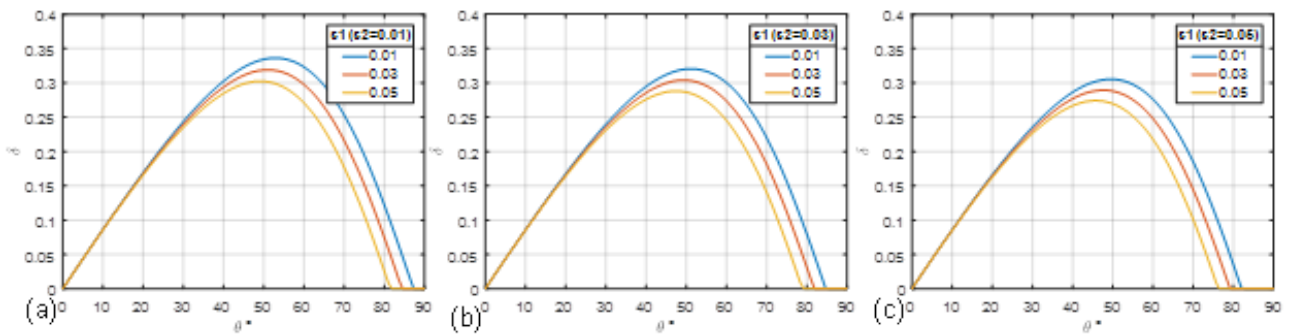


Figure 8. Lateral displacement ratios δ dependence on skew θ for $\tilde{s}_1 = 0.01$ (blue), 0.03 (red), 0.05 (orange) in the legends and for selected $\tilde{s}_2 = 0.01$ (a), 0.03 (b), 0.05 (c). Sliding friction coefficient $\mu = 0.4$.

rolling coefficients for increasing sliding friction coefficients.

The dynamic approach to the investigated problem allows for the confirmation of the experimental solution to the displacement ratio $\delta = 0.207$, for an angle of $\theta = 30^\circ$. This result is expressed as possible pairs of relative rolling coefficients $\tilde{s}_1$ and $\tilde{s}_2$ for a given sliding friction coefficient. On Figure 7 two combinations ($\tilde{s}_1 = 0.05$,

$\tilde{s}_2 = 0.03$) and ($\tilde{s}_1 = 0.03$, $\tilde{s}_2 = 0.05$) are marked by black triangles, which lead to the experimental values for $\mu = 0.2$. For $\mu = 0.4$ (Figure 8) no confirmations were found. Based on many computations for $\tilde{s}_1$, $\tilde{s}_2$ in the same range and $\mu = 0.2$, the domains of existence of the experimental values can be found, shown as blue areas on Figure 9.

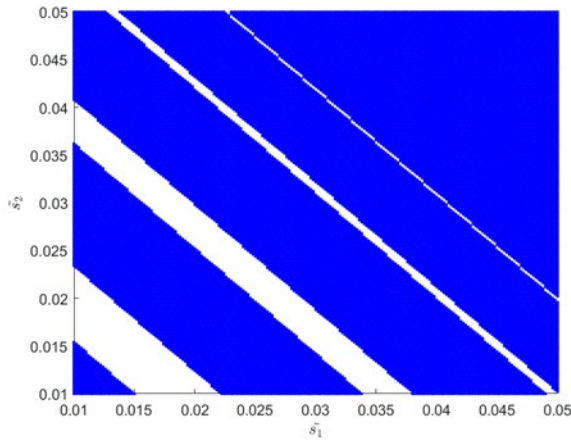


Figure 9. Regions for which the experimental values are obtained (marked in blue) in the $[\tilde{s}_1, \tilde{s}_2]$ search domain.

4. LONGITUDINAL OSCILLATIONS OF THE ROLLER

Due to the required simple mechanical design, the roller is simply mounted using a steel shaft rigidly fixed on the translating platform. The absence of ball bearings enables the cylindrical roller to move along the shaft, as the clearance permits. Being made of an elastic material, the relative axial motion of the roller of mass m can be assumed to be oscillatory. We model this system as a rigid body held axially by a spring of elastic constant k representing the elastic axial contact of the roller with the platform.

The differential equation of motion is :

$$m\ddot{y} = -ky - F_{f1y} \quad (17)$$

in which the friction force is: $\sqrt{F_{f1x}^2 + F_{f1y}^2} = \mu(G_1 + G_2)$ and the component F_{f1x} is given by eq. (11).

We assume here a sliding friction coefficient being linearly decreasing with the local sliding velocity,

$$\mu = \mu_0 - \mu_1 v_{ay}, \quad (18)$$

with v_{ay} the velocity of the roller along Oy :

$$v_{ay} = v_t \sin \theta + \dot{y} = v_0 \sin \theta + \dot{y}. \quad (19)$$

It follows:

$$F_{f1y} = \sqrt{[\mu_0 - \mu_1 (v_0 \sin \theta + \dot{y})]^2 (G_1 + G_2)^2 - d^2} \quad (20)$$

$$= \sqrt{(c_0 - c_1 \dot{y})^2 - d^2}$$

in which $c_0 = (\mu_0 - \mu_1 v_0 \sin \theta)(G_1 + G_2)$,

$$c_1 = \mu_1 (G_1 + G_2), \quad d = \frac{s_1 (G_1 + G_2) + s_2 G_2}{R}.$$

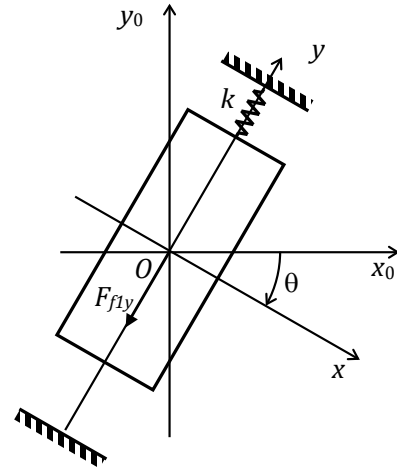


Figure 10. The roller can oscillate along the shaft (Oy axis).

If the rolling friction influence is neglected for large μ_0 (see the previous paragraph), then $d=0$ and:

$$F_{f1y} = c_0 - c_1 \dot{y}. \quad (21)$$

The differential equation of motion (17) becomes:

$$m\ddot{y} - c_1 \dot{y} + ky = -c_0. \quad (22)$$

The general solution of this equation is:

$$y(t) = \exp(\alpha t) [A \cos \beta t + B \sin \beta t] - \gamma, \quad (23)$$

$$\text{with } \alpha = \frac{c_1}{2m}; \quad \beta = \sqrt{\frac{k}{m} - \alpha^2}; \quad \gamma = \frac{c_0 m}{k}.$$

Assuming as initial conditions $y(0)=0$ and $\dot{y}(0)=0$, the equation of motion becomes:

$$y(t) = \gamma \exp(\alpha t) \left(\cos \beta t - \frac{\alpha}{\beta} \sin \beta t \right) - \gamma. \quad (24)$$

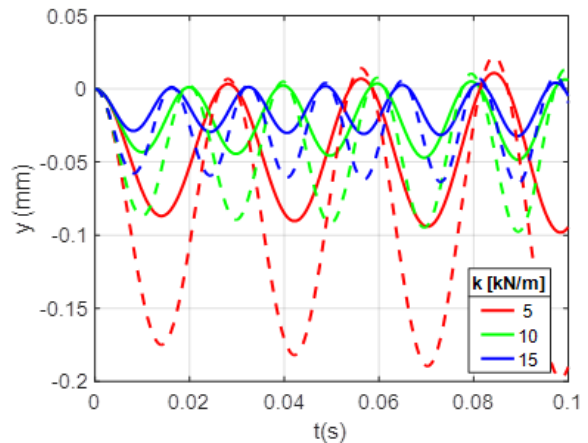


Figure 11. Axial motion of the roller for 0.1 s and $\mu_0=0.2$ (continuous lines) and $\mu_0=0.4$ (dashed lines), for a set of elastic constants $k=5, 10, 15$ kN

As a numerical example, we consider a small roller of $m_l=0.1$ kg, radius $R=0.01$ m, carrying a

mass $m_2=1\text{kg}$. The elastic axial support is represented by a range of constants $k \in \{5, 10, 15\}$ kN/m, since the exact value is not available at this stage. The platform is translating with a constant velocity $v_0 = 0.1$ m/s.

Two static friction coefficients are used in the simulation: $\mu_0 \in \{0.2, 0.4\}$ for the roller-platform, as in the previous paragraph, and $\mu_1 = 0.05$ s/m was estimated for realistic solutions. The oscillatory axial displacement $y(t)$ is given by eq. (24), for the elastic constants and selected friction coefficients are depicted in Figure 11, for a simulation duration of 0.1 seconds. The higher friction coefficient, $\mu_0=0.4$, leads to higher oscillation amplitudes due to the variable friction force with motion velocity. This aspect, pertaining to a stick-slip phenomenon, as well as the motion of the roller, especially the coupling between the oscillatory component and its steady-state motion, remains a perspective for continuation of this research.

5. CONCLUSIONS

In this study, the motion of a laterally transporting platform equipped with a skew hinged roller and pulled at constant velocity was investigated through progressively more complex kinematic and dynamic models. The analysis began with a pure rolling kinematic approach and was subsequently extended to steady-state dynamic formulations, incorporating sliding friction and rolling resistance moments, including the case of a velocity-dependent sliding friction coefficient. The results identified the optimal roller shaft fixing angle that maximizes the lateral displacement of the transported object, a parameter of primary practical interest for package-sorting applications. While the kinematic model provides an initial estimate of this optimum, the dynamic analysis reveals significant corrections due to frictional effects and highlights the influence of both sliding and rolling friction coefficients on the system performance. To the authors' knowledge, this represents the first comprehensive investigation of such a mechanism reported in the technical literature. Although direct comparisons with previous studies are not available, the predicted trends show reasonable agreement with observations reported by users of similar devices. Future work will focus on experimental validation of the proposed models through dedicated laboratory

testing and on further refinement of the dynamic description based on the obtained measurements.

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