Optimal Control Design for Robust Fuzzy Friction Compensation in a Robot Joint

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Abstract- **This paper presents a methodology for the compensation of nonlinear friction in a robot joint structure based on a fuzzy local modeling technique. In order to enhance the tracking performance of the robot joint, a dynamic model is derived from the local physical properties of friction. The model is the basis of a pre-compensator taking into account the dynamics of the overall corrected system by means of a minor loop. The proposed structure does not claim to faithfully reproduce complex phenomena driven by friction. However, the linearity of the local models simplifies the design and implementation of the observer and its estimation capabilities are improved by the nonlinear integral gain. The controller can then be synthesized robustly using Linear Matrix Inequalities to cancel the effects of inexact friction compensation. Experimental tests conducted on a robot joint with a high level of friction demonstrate the effectiveness of the proposed fuzzy observer-based control strategy, for tracking system trajectories when operating in zero velocity regions and during motion reversals.**

Index Terms— **friction compensation, fuzzy modeling, fuzzy observers, LMI, optimal H_∞ control.**

I. Introduction

In control applications involving small displacement, low velocities and motion reversal, friction modeling and compensation is of paramount importance. In particular, many physical phenomena such as stiction and presliding displacement can have a considerable influence on the system performance and stability; this can result mainly in stick-slip motions. In mechanical systems, nonlinearities are considered as a serious issue and have been the center of attention for many years. The large amount of research dealing with the problem has led to the development of various compensating strategies of nonlinear friction [1],[2],[3]. Some of the proposed approaches are based on a reasonably accurate modeling of the nonlinearity while others have considered the friction as part of the disturbances acting on the system [4]. In this case, a disturbance rejection technique [5] or a nonlinear controller can be applied to improve the system performance [6], [7],[8]. In the first approach, friction is seen as a physical phenomenon characterized by micro-sliding displacements, varying break-away force, and frictional lag. This has motivated the use of a dynamic model instead of the classical static friction-velocity map. Dynamic models have essentially been developed to give a better description of friction phenomena in mechanical systems characterized by the following physical observations:

- presliding displacement: motion during stiction with contact deformation at zero velocity where friction is only a function of displacement.
- frictional memory: effect observed in the form of hysteresis loops relating friction to input velocities.

Starting with the Dahl model [9], many dynamic models have been proposed: LuGre model [10], Leuven model and many others [11][12][13]. In fact, these proposed dynamic models claim fidelity for the reproduction of friction behavior, however, the precision required in the context of friction compensation is associated with considerable identification effort due to the model complexity. Furthermore, the control algorithms based on these models are even more complicated at the design level and during implementation.

The idea is to represent local friction behavior by a dynamic linear model, then design a local friction observer for each model, the overall observer is constructed using the principle of parallel-distributed compensation resulting in a local-based friction compensator. Based on the general Stribeck [14] curve with Dahl effects [9] and inspired from the dynamic nature of the bristle interpretation of friction phenomena [15], an equivalent dynamic model of nonlinear friction is designed to cancel friction in the robot joint at low velocities. This model is used with a tracking controller primarily considered in the controlled robot joint.

The rest of this paper is organized as follow: Section II introduces a dynamic structure of friction in its general form in a simple model of a single robot joint. In Section III, the local modeling approach is then developed taking into account the identified friction behavior at different velocities. As a modeling-control approach is based on dynamic fuzzy models, the friction parameters are identified locally for the model definition, and used afterwards for the design of a fuzzy observer of friction forces for compensation purposes. The overall control scheme is the sum of the nonlinear

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compensating term provided by the proposed observer that compensates the major part of friction and a robust H_∞ controller design based on LMI approach for disturbances and uncertain compensated term rejection in an outer loop. Finally, Section IV presents some experimental results to validate the control system effectiveness in tracking different velocity ranges.

II. Friction Dynamics and Robot Joint Modeling

Since the following development concerns a friction compensation task, we consider a single robot joint's dynamics, which can be described by:

$$J\ddot{q} = \tau - F + \delta_0(t, q, \dot{q} ...) \tag{1}$$

Where J is the inertia of the joint, τ is the control signal, F represents the system's friction forces, δ_0 is an unknown bounded function considered for the robust control design in Section III which includes all disturbances and other nonlinear dynamics of the robot joint after cancellation and $q, \dot{q}$ and $\ddot{q}$ are the position, velocity and acceleration, respectively.

In the robot joint, friction dynamics can be expressed as:

$$\begin{aligned} \dot{z} &= \eta(z, q, \dot{q}) \\ F &= N(z, q, \dot{q}) \end{aligned} \tag{2}$$

Where z represents an internal non-measurable state of friction [10], η and N are nonlinear functions of z, q and $\dot{q}$ which may also include hybrid dynamics usually needed for a more faithful reproduction of the friction physical behavior. It should be emphasized that this form represents a single state dynamic model similar to many friction models like Dahl and LuGre models. Furthermore, it is natural to see this model as a general form of these models and most of the analysis related to the stability, passivity and mathematical properties are directly applicable to this model. Therefore, in this work, a friction compensator is proposed based on the fuzzy model structure and an optimal controller design to guarantee the stability of a pre-compensated system is then reviewed.

The complex model structure is decomposed into a series of linear state space time-invariant models. This will hold inside a set of velocities where the size of each set is decided according to how fast the dynamics of the identified input-output map is. For the friction model this means that more models are required for low velocities region, the region where friction is known to be highly nonlinear.

Using local approximation techniques, (2) can be expressed in the form of a linear state space model using a set of IF-THEN rules,

For Rule $i = 1 \ldots n$,

$$\text{IF } \dot{q} \text{ is } \Omega_i \quad \text{THEN} \quad \begin{aligned} \dot{z} &= a_i z + b_i \dot{q} \\ F &= c_i z + d_i \dot{q} \end{aligned} \tag{3}$$

Where Ω_i is the fuzzy set of velocities associated with the local dynamics; a_i, b_i, c_i and d_i are parameters of the proposed model

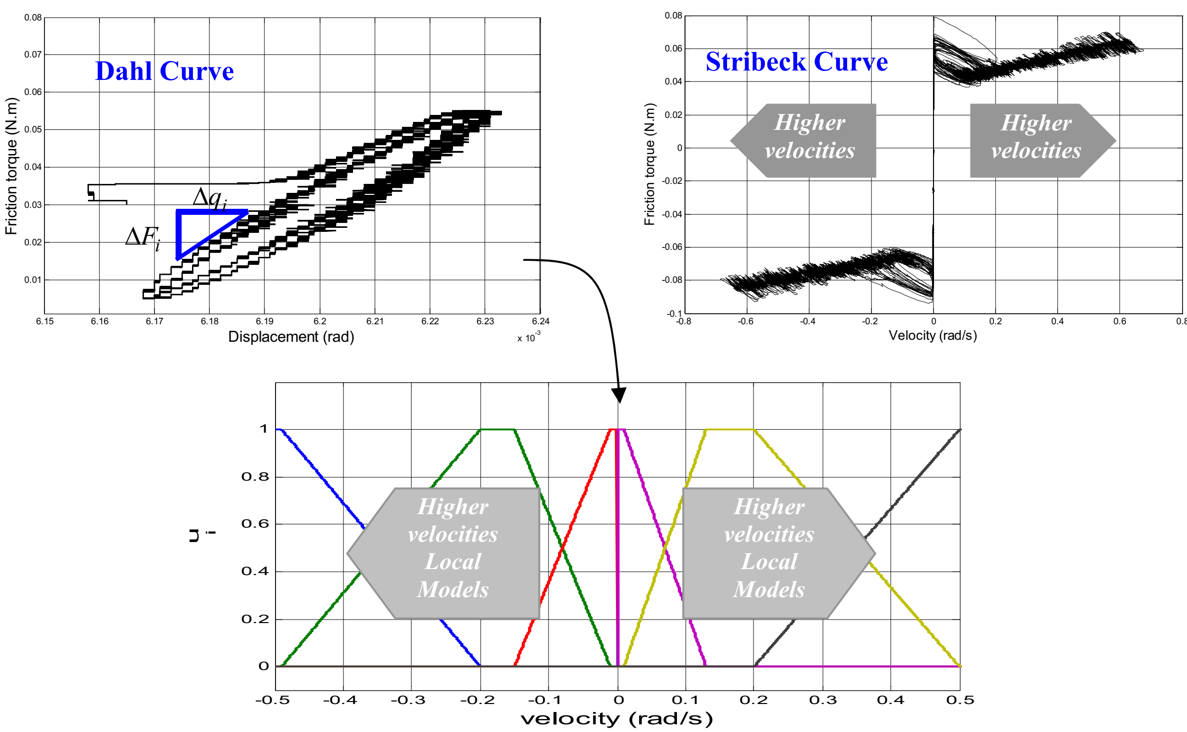

Fig 1. Basic idea in modeling friction phenomena: at zero velocity, friction is basically related to position, and becomes exclusively dependent on the velocity in the sliding regime, the membership functions allow a mixed regime and soft switching between dynamics and ensure good representation of friction forces in the robot joint. Top right: The Stribeck curve characterizing friction-velocity relationship, it might be clearly asymmetric in reality; on top left: the Dahl curve illustrating micro-displacements regime ($\sigma_0 = \Delta F_i / \Delta q_i$) in the robot joint: experimental curve.

that are able to describe friction characteristics locally and consequently they will be kept constant inside the equivalent set Ω_i, which will be defined in next section.Now, let $\mu_i(\dot{q})$ be the normalized membership function of the inferred fuzzy set Ω_i where $\Omega_{all} = \prod_{i=1}^{n} \Omega_i$ denotes the overall operating range of velocities of the considered system.

By applying a standard fuzzy inference method based on the singleton fuzzifier, product fuzzy inference and center average defuzzifier, the mechanism of estimation is an interpolation of all the identified local models along the operating range i.e. equation (2) can be reproduced accurately by mean of fuzzy dynamic models. However, it will depend strongly on the number of dynamic models used, the membership functions and the identification method used [16]. However, the discontinuity occurring at zero velocity can be a big challenge and switching functions are usually used as solution to this problem [17] [18].

$$\begin{aligned} \dot{z} &= \sum_{i=1}^{n} \mu_i(\dot{q}) a_i z + \sum_{i=1}^{n} \mu_i(\dot{q}) b_i \dot{q} \\ F &= \sum_{i=1}^{n} \mu_i(\dot{q}) c_i z + \sum_{i=1}^{n} \mu_i(\dot{q}) d_i \dot{q} \end{aligned} \tag{4}$$

Where μ_i denotes the membership functions.

Fuzzy models are known to be universal function approximators [19], and this property gives (4) the ability to reproduce (2) faithfully by using some available tools for parameter identification and tuning such as adaptive neural fuzzy inference systems (ANFIS) or genetic algorithms. However, the effort made to refine the model can be seriously compromised by the varying nature of friction. For this reason, local models are meant to reproduce the main feature of friction inside a certain set and the simplicity of the chosen dynamics allows relatively easy design of the compensator.

Since the friction model structure has been established, we can define the parameters in (4) depending on the operating input velocity; namely: the stiction level, the presliding displacement and the Stribeck effect. This will be detailed in the next section. Some effects such as varying break-away force and frictional lag will not be taken into account in the design, since we can avoid the complexity without affecting the performance of the designed compensator.

III. LOCAL-BASED COMPENSATION APPROACH

The proposed friction compensation scheme is composed of two main control actions: a nonlinear friction estimator generating a signal to be rejected for the elimination of friction-induced errors and an optimal H_∞ controller based on LMI design under the inexact friction compensation assumption.

Defining four parameters plus the size of each set Ω_i can be quite challenging, especially if the model is expected to be reasonably accurate. Therefore, the method developed in this paper requires that the model represents the main features of friction inside each local set. The observer based on this model structure can then be refined using a set of gains to improve its convergence to realistic values. Since we can have a prior knowledge about the main feature of friction and the Velocity-Friction torque map, it is possible to identify the model parameters in two successive steps, for the presliding displacement regime running at very low velocities and for higher velocities equivalent to the sliding regime.

A. Local Approach Applied to Dynamic Friction Modeling

In the zero velocity zone and during micro-sliding motions, the frictional force in (4) can be expressed by the Dahl effect formula where friction is a function of displacement and the dynamics due to velocity are not taken into account. This can written as :

$$F = \sigma_0 z \approx \sigma_0 q \tag{5}$$

Therefore $z \approx q$, which allows us to determine $c_i = \sigma_0 = \Delta F / \Delta q$ from the Dahl curve shown in Fig. 1. Similarly, the internal state of friction z is consequently equal to the displacement and the friction model dynamics can be completed as follows:

$$\dot{z} \approx \dot{q} \tag{6}$$

By comparing (4) and (6) gives the parameter $b_i = 1$ that holds for presliding regime. Note that (5) and (6) are a special case of (4).

For simplicity and knowing that the sum of membership functions $\mu_i(\dot{q})$ at any point of the operating domain is equal to 1, the parameters b_i and c_i can be kept constant for all operating points without loosing the capability of the proposed structure to describe the friction behavior. Basically, the main features of nonlinear friction are captured by the internal state z characterizing the stiff nature of friction which is combined, in the proposed structure, with a component having a damping effect on the system.

At relatively higher velocities, friction is more velocity dependent and for the steady state regime, two domains can be distinguished: at relatively low velocities, the nonlinear part is characterized by mixed dynamics and a negative damping term due to the Stribeck velocity; at higher velocities, the linear part is characterized only by the viscous friction as a positive damping term as described in Fig. 1.

The steady-state characteristics of the proposed structure of (4) may then be found. By letting $dq/dt = 0$ and taking into account the parameters identified previously using (5) and (6), we can write:

$$F_{ss} = \left(-\sum_{i=1}^{n} \mu_i(\dot{q}) \frac{\sigma_0}{a_i} + \sum_{i=1}^{n} \mu_i(\dot{q}) d_i \right) \dot{q} \tag{7}$$

The rest of the parameters can be deduced from (7) by comparison using the identified level of friction. The steady-state friction can be represented by a static map between friction and velocity; it takes the so-called Stribeck curve form, which is experimentally identified at constant velocities. Thus $d_i = d_0$, which represents the damping term associated with the viscous friction at relatively high velocities, and $a_i = \alpha_i$ will be varying with the velocity and takes the value calculated from F_i which represents the friction level at the velocity $\dot{q}_i$,

$$\alpha_i = -\frac{\sigma_0}{F_i}\dot{q}_i \tag{8}$$

α_i can be defined in a bounded domain described by the following inequality $-\frac{\sigma_0}{F_S} \geq \frac{\alpha_i}{|\dot{q}_i|} \geq -\frac{\sigma_0}{F_C}$ with respect to all operating point except for $\dot{q} = 0\ \ (rad/s), F_i = 0\ \ (N.m)$ where F_C and F_S represent in this case the levels of Coulomb friction and Static friction respectively.

The final form describing the internal state dynamics and the output of the proposed model structure can be written after substitution of all the identified parameters as:

$$\dot{z} = \sum_{i=1}^{n}\mu_i(\dot{q})\alpha_i z + \dot{q} \qquad F = \sigma_0 z + d_0\dot{q} \tag{9}$$

Equation (8) is important since it represents the bounds that encompass the nonlinear behavior of friction, which has a direct influence on the stability of the overall proposed control system. The validation of this model can be done via a simple comparison of its parameters to other existing models, so that the mathematical properties such us linearization, passivity and stability can be shown. Some comparative simulations can be found in [20]. Further development for the generalization of the model and the validation is currently being performed.

B. Observer-Based Friction Compensator Design

The proposed friction compensator is derived from a re-formulation of the friction dynamics in (9). A rejection of disturbances caused by inexact friction estimation is achieved by the compensating gains acting as a local integral action [21]. These gains are chosen within a pre-defined domain and their values will be fixed during the experiments in order to reach the best performance. The outer closed-loop system will satisfy the robust stability condition under the following assumptions: (i) inexact compensation resulting from uncertain estimation namely $\Delta\sigma_0$ and Δd_0 , (ii) varying parameters resulting from the fuzzy modeling α_i , (iii) controller design parameters in the pre-compensation loop such as l_i , κ_i and κ'_i , and finally (vi) existence of disturbances δ_0 . Fig. 2 shows the proposed friction compensation control scheme applied to the robot joint. Fig. 3 shows the local representation of the frequency response of the system with friction before and after introducing the pre-compensator which demonstrates a clear improvement on the local dynamics and allows the robust design of the control law considering uncertainties in the compensation.

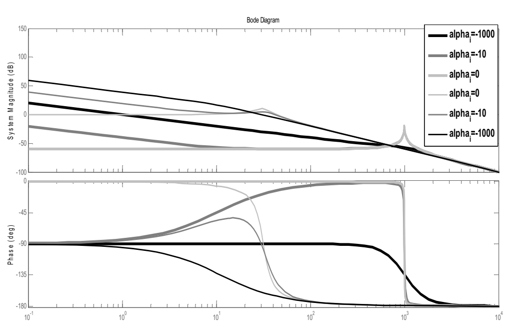

Fig.3. Frequency response of the considered system before (dotted line τ / q) and after pre-compensation (solid line τ^* / q), 3 local models are shown for different range of velocities (at presliding regime and for higher velocities)

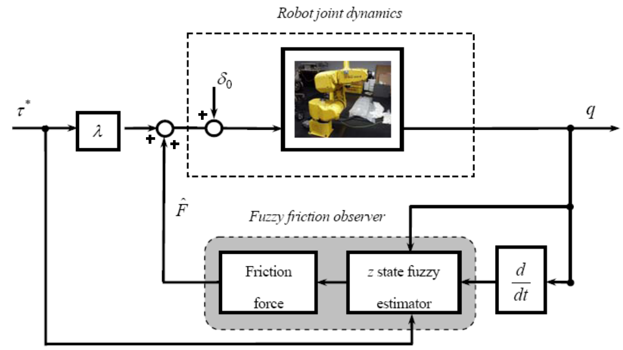

Fig.2. Friction compensation in robot joint based on the local design: minor loop control.

The applied control ensuring quadratic stability of the system given by (1) with friction modeled in (9) yields the following dynamics:

IF $\dot{q} = \dot{q}_i$ THEN

$$\dot{\hat{z}} = \alpha_i\hat{z} + \kappa_i q + \dot{q} - l_i.\tau^* \tag{10}$$

$$\hat{F} = \lambda\tau^* + \kappa'_i\, q + \sigma_0\hat{z} + d_0\dot{q} \tag{11}$$

In (11), $\lambda > 0$ is a fixed positive gain of the feedback controller that can be defined at the robust H_∞ design stage; κ_i and κ'_i are small positive gains added to the dynamics of the local model in order to satisfy the quadratic stability criteria and H_∞ control performance for the resulting polytopic uncertain form described by (1), (10) and (11); the pre-compensated dynamic model is characterized by bounded disturbances and uncertainty boxes that can be classified into two types: 1) parameters related to modeling uncertainties and mismatch in friction compensation such as $\Delta\sigma_0$ and Δd_0 , they can be varying locally or set to a value that represent the worst mismatch situation for all the operating domain 2)- design parameters such as : l_i , κ_i and κ'_i that will be defined locally and that can be decided later in the experiments after the calculation of H_∞ controller gains using Linear Matrix Inequalities approach. κ_i and κ'_i are set to a small value around zero velocity and then set to zero for the remaining operating velocities, they are necessary to find a solution to the set of LMIs into the overall domain. The optimal control problem is then formulated as follow: seeking a single quadratic Lyapunov function that enforces the design objectives for all plants in the pre-defined polytope; in other terms, find a stabilizing state feedback control τ^* that minimizes the closed-loop RMS gain of the plant from $\xi_\infty = q$ to δ_0. This problem can be transformed into an LMI problem, and the RMS gain is guaranteed not to exceed some prescribed performance value γ if there exists a positive matrix P_∞ that satisfies the following inequalities [22];

$$\begin{bmatrix} (A_i + B_{2i}K)P_\infty + P_\infty(A_i + B_{2i}K)^T & B_{i1} & P_\infty(C'_1 + D'_2 K)^T \\ B_{1i}^T & -I & D'^T_1 \\ (C'_1 + D'_2 K)P_\infty & D'_1 & -\gamma^2 I \end{bmatrix} < 0$$
$$P_\infty > 0$$

Where all parameters for the robust design are given as follows,

$$A_i = \begin{bmatrix} \alpha_i & \kappa_i & 1 \\ 0 & 0 & 1 \\ \Delta\sigma_0 & \kappa'_i & \Delta d_0 \end{bmatrix},\ B_{1i} = \begin{bmatrix} 1 \\ 0 \\ 1 \end{bmatrix}, B_{2i} = \begin{bmatrix} l_i \\ 0 \\ \lambda \end{bmatrix};$$
$$C'_1 = \begin{bmatrix} 0 & 1 & 0 \end{bmatrix}, D'_1 = 0, D'_2 = \begin{bmatrix} 0 & 0 & 0 \end{bmatrix}$$

The estimation mechanism in (10) uses the dynamics of (9) added to an error compensating term modulated by a local gain l_i and local feedback terms. The local gains can be derived from linear design techniques to ensure stable behavior of the inner loop representing the pre-compensated system with friction separately. Then the stability of the overall controlled system is taken into account by solving the LMI and the existence of a Lyapunov quadratic matrix P_∞ leads to the following overall controller expression $\tau = \lambda\tau^* + \hat{F}$; where λ is a positive gain of the controller that will be set to 0.5.

$$\text{with} \quad \tau^* = KX = k_p q + k_d \dot{q} + k_z \hat{z}$$
$$\text{and} \quad \dot{\hat{z}} = \sum_{i=1}^{n} \mu_i(\dot{q})\alpha_i \hat{z} + \sum_{i=1}^{n} \mu_i(\dot{q})\kappa_i q + \dot{q} - \sum_{i=1}^{n} \mu_i(\dot{q}) l_i . \tau^* \qquad (12)$$

Where K represents the calculated state feedback vector of the optimal controller, k_p, k_d and k_z are the position, velocity and friction state gains respectively [23]. The third part of the control element τ^* in (12) is termed “virtual control” and can be seen as an additive compensation term of friction and a stabilizing part of the control at the same time.

The term “virtual control” is used to describe the fact that the state z is non-measurable and has been introduced to describe friction. The experimental results have shown that this can bring a slight improvement in terms of disturbance rejection, though further experiments and analysis are needed.

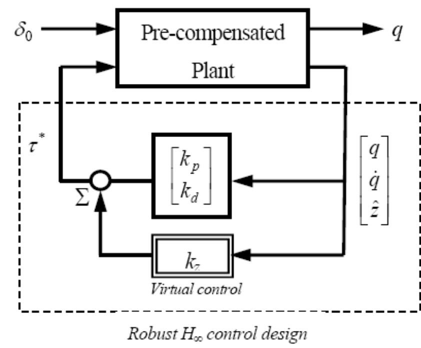

Fig.4. LMI-based robust controller: outer loop design.

Furthermore, it should be noted that this scheme relies upon the worst-case design using local models of friction with uncertain compensation and external disturbances. In our case quadratic stability of the pre-compensated system is checked for the varying parameters resulting either from the friction compensation mismatch or the design choice. This allows us to tune and choose the observer local gains l_i that ensure the best tracking performance without compromising the stability of the overall system.

For the velocity range of [-0.5,0.5]rad/s, seven local models are used to reproduce the behavior of the nonlinear shape of the Stribeck curve that characterizes dynamic friction inside the slow motions regime set including the reversal velocity region. By applying a standard fuzzy inference method, i.e. using a singleton fuzzifier, product fuzzy inference and center average defuzzifier, the mechanism of estimation will work as an interpolator of all the relevant linear estimators [24]. The control action combines a direct friction compensation ensured by the fuzzy observer and the action of an optimal tracking controller Fig. 4.

IV. EXPERIMENTAL RESULTS AND EVALUATION

Experiments were performed on a joint of a FANUC robot in order to evaluate the proposed control strategy. The experimental setup consists of a PC 700 MHz running the operating system RT-LINUX, connected by an optical cable to a digital servo adapter that provides signal interfacing between the PC and a servo amplifier module. The control algorithm is implemented in C language. The gains of the observer were tuned in during the experiments after defining all the model and controller parameters. Since the current work deals with friction compensation, only one isolated joint will be used in the experiments, and the results can be extended to other joints. We should also note that the extension to other joints can be fruitful for relatively slow motions, since other nonlinear dynamics are velocity dependent and can be seen as minor disturbances, otherwise, they should be compensated beforehand. The control algorithm as implemented depends on the velocity which is by the way estimated using the signal of a position encoder and can have a direct influence on the quality of the control signal. A good estimation of the velocity by a differentiation-low pass filtering of the signal acquired from the encoder is then used for a better signal quality.

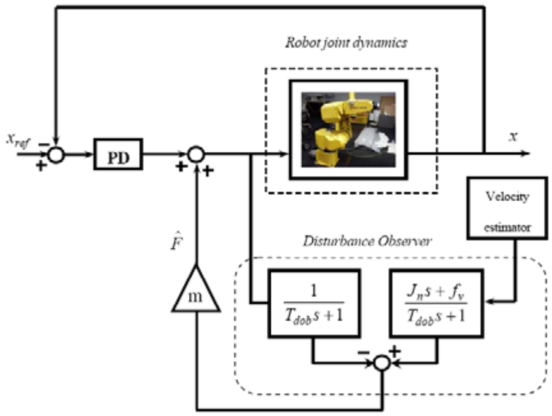

Fig.5. Comparative control methods. m=0: PD Control, m=1: DOB.

In order to evaluate the proposed control designed for friction compensation, experiments were performed on a robot joint system for a trajectory tracking task, with different velocity ranges. Comparisons to other control methods namely Proportional Derivative (PD) control and Disturbance Observer (DOB) based control are reported. Fig. 5 shows the results obtained with PD control for (m=0) and DOB control for $m \neq 0$. We used linear techniques to determine the parameters of the comparative control. Basically, these methods use only the linear parts of the considered robot joint dynamics, so that the pole placement used for PD design or the inverse model to form the DOB filters were calculated using nominal parameters of the joint inertia J_n , and the viscous friction f_v. Note that only the linear part of the system consisting of inertia and viscous friction as a damping factor is used for the control design in Fig. 4 [25]. The reference trajectory $q_{ref} = (1 - 0.1(t - 0.5))\sin(2\pi f (t - 0.5))$ is shown in Fig. 6.

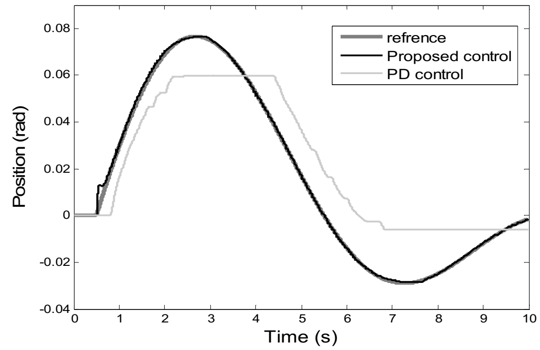

Fig.6. Position tracking performances before and after compensation.

Therefore, the robot joint will be operated in the low velocity region with $f = 0.1Hz$ and performing many velocity reversals during the experiment.

Since the robot joint comprises a considerable friction component, PD control has serious limitations and shows residual tracking errors that cannot be eliminated even with high PD gains.

Around zero velocity, it is clear that the tracking performance of the robot joint is severely affected by friction as shown in Fig. 7. The fuzzy observer with a gain scheduling property is proposed as an efficient way to compensate friction errors without using highly excessive control input for the local operating range.

Fig. 7 shows a clear reduction in the friction induced error. This can be explained by the fact that a good estimation of friction by the fuzzy observer and the disturbance rejection leads to robust performances. There is a large residual error due to friction at zero and low velocity, this error can be minimized by the use of disturbance observer, but the performances reached by the DOB remains limited due to the highly nonlinear nature of friction in the low velocity regime for the

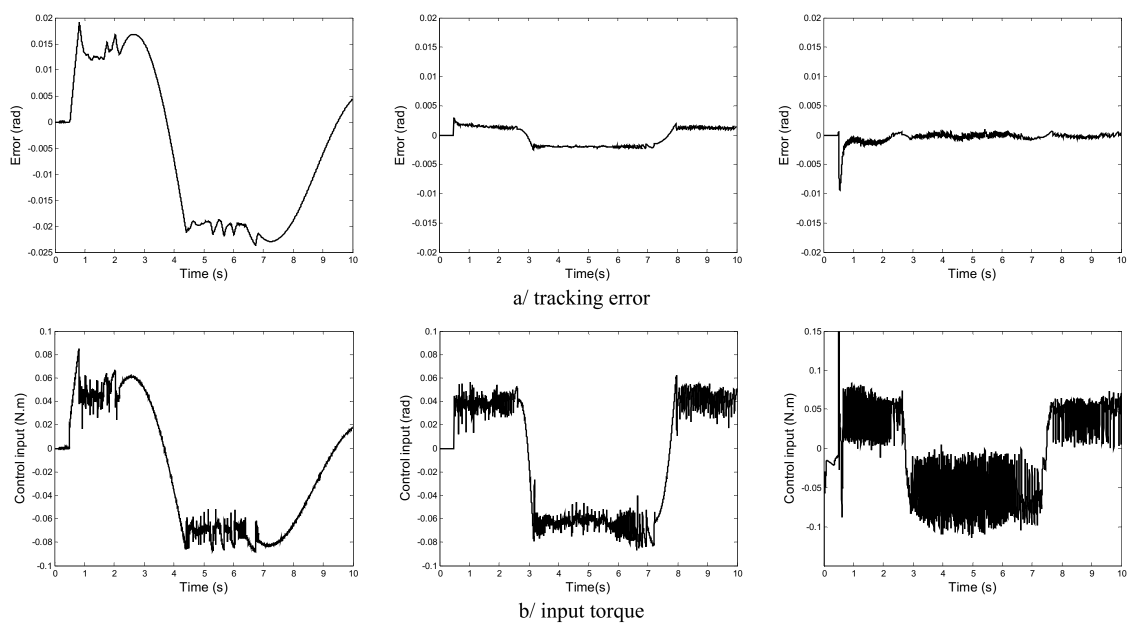

Fig.7. Experimental results showing tracking performances: left: under PD control, middle: Disturbance observer based, right: proposed method.

chosen reference trajectory. After robust friction compensation, the tracking error is bounded and minimized to a value less than 0.002 rad, and the robot joint responds more smoothly during velocity reversal.

The tracking performances can be measured by the calculated Recursive Mean Square Error that is reported for all cases in the following table;

RMS tracking errors (rad).

PD Control	*DOB control*	*LMI based (proposed)*
0.0125 rad	*0.0013 rad*	*0.0006 rad*

By using compensating gains in the low velocity region, the observer was able to give better results in friction estimation and reduction of tracking error. On the other hand the H_∞ controller has been designed to handle a bounded compensation mismatch since the friction phenomena itself is inherently variable and very difficult to model with accuracy.

V. Conclusion

A dynamic friction structure based on a local modeling approach has been proposed for the compensation of friction in motion control systems. Motivated by the dynamic nature of friction, the estimation mechanism uses local properties and adds a component to the control signal to cancel friction effects at low velocities. The proposed control scheme relies on local identified parameters and a relatively simpler design technique than other model-based friction compensation methods. On the other hand, the robust control design via LMI approach ensures robustness and performance under some severe assumptions like uncertain friction compensation and fuzzy varying gains for the observer. The number of tuning parameters is related also to the number of models and therefore can increase the complexity of the design. This can be the basis of further developments and investigations and a robust adaptive control can be proposed.

Appendix

Table of Controller parameters.

K_P (N.m/rad)	3.6
K_D (N.m.s/rad)	0.21
k_p (N.m/rad), k_d, (N.m.s/rad) k_z (N.m/rad)	1.8, 0.14, 1574.9
λ	0.5
κ_i , κ'_i	0.001, 0.02
$\dot{q}_i$ (rad/s)	-0.5,-0.1,-0.01,-0.001,0,0.001,0.01,0.1,0.5
l_i	0,-0.5,-1.5,-4.5,-1.5,0.5,0

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