Measuring Coulomb and Viscous Friction in Revolute Joint

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Abstract

In this research work, the emphasis is on the study of the effects of friction in the revolute joint for fine manipulation of robotic arm manipulators. The friction model is used to capture the friction in static mode, the breakaway point of displacement and the linear part of the friction when asperity junctions are broken. The experimental setup is designed to attain Coulomb and viscous friction parameters in the revolute joint with velocity sensor equipped with mechanical system. The velocity sensor detects velocity response as controlled ramp torque is applied to the physical model to evaluate friction in the joint. The physical model is designed and built in MSC ADAMS environment and simulated using the same parameters which were experienced by the real model. The block diagram is drawn in the Simulink (MATLAB) under same configuration and then simulated. The results, obtained from MSC ADAMS and Simulink, are compared with the experimental data.

Keywords: Friction Measurement, Revolute Joint, Mechanical Systems, Block Diagram Model, Control

Introduction

Friction occurs in mechanical systems where there is a contact between two objects. The contact may be either directly to the surfaces, that is, dry friction or between contacts lubrication is introduced called viscous friction. The friction force depends on the physical characteristics of contact surfaces, such as material properties, geometry of contact faces, relative velocity and displacement of contact surfaces, and lubrication. Many models of friction [1-3] have been proposed to deal with the various regimes of friction, each with their own limits. To compensate the frictional effects, classical friction models such as Coulomb and viscous friction are mostly used [1]. The control of manipulator end-effector is required in surgical and many industrial applications during slow motions and velocity reversals. The friction response during presliding regime plays great role in high precision and tracking systems [4].

The nonlinear frictional behavior in the joints causes stick-slip motions, limit cycle and static positioning errors [5]. In presliding regime, the adhesive forces are dominant and friction force is function of displacement [6]. The asperity junctions act as nonlinear springs and as displacement increases junctions will break up and thus move to gross sliding regime where friction is a function of velocity [6]. So a very accurate friction model is required that captures the overall frictional effect. The transition between static and dynamic regimes magnifies the nonlinear frictional behavior in the joints [7].

Coulomb and viscous friction parameters are of vital importance because of their presence in all friction models. Both produce forces against the relative motion between the bodies in contact. The viscous friction is proportional to speed while coulomb friction depends upon sign of velocity [3]. The friction in the sliding regime is a static relation between velocity and friction force, the simplest of which is combination of Coulomb and viscous fiction [6].

The models that include Stribeck effect, at low velocity determines friction comparatively better, but it is realized that at this stage friction force is a decreasing function of velocity [9]. Velocity control is a practical conscious issue in robotics because the joint velocity is input to robotic manipulator [8] and it is desirable that it must be known for precise control and fine manipulation.

The objective of this research paper is to measure Coulomb and viscous friction parameters experimentally in the revolute joint of a mechanical system. The velocity sensors are attached to the system to analyze velocity response. The Coulomb and viscous friction parameters are relevant to some velocity features [3]. The dynamic analysis of the mechanical system is carried out using the software "ADAMS". The block diagram of physical model is drawn in Simulink tool of Mat lab and velocity response is predicted in the revolute joint. The mathematical model of the system is solved in Maple and plotted to get velocity response in the revolute joint. From velocity responses the Coulomb and Viscous friction parameters are obtained. Finally a comparison of experimental results with ADAMS and SimMechanics are carried for comparison. A good agreement is achieved in the experimental and predicted results. This gives us confidence in our computational model.

Coulomb and viscous friction model

Coulomb and viscous friction parameters are the main ingredients present in all friction models. Both produce force against motion, the Coulomb friction depends upon sign of velocity while latter is proportional to speed. In the experimental setup a ramp torque τ is applied to the system, q is the angular position of the joint; J is its inertia and f the frictional force.

The Newton second law for an external torque τ, an

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angular acceleration $\ddot{q}$, considering friction *f* in the joint is represented below

$$J\ddot{q} = \tau - f \qquad (1)$$

A ramp torque $\tau(t) = mt$ is applied to the system where $t > 0$ is time and $m > 0$ is slope.

The friction force *f* is a function of velocity [3] and it is combination of coulomb and viscous friction and written below

$$f(\dot{q}) = f_v\,\dot{q} + f_c\,\mathrm{sgn}\,\dot{q} \qquad (2)$$

$f_c > 0$ coulomb friction
$f_v > 0$ viscous friction
q relative velocity

The velocity response of motor under no-load and loaded conditions assuming that friction is modeled by (2) and ramp torque τ is applied to the system is

$$d\dot{q}/dt = 1/J\,[mt - f_v\,\dot{q} - f_c\,\mathrm{sgn}\,\dot{q}] \qquad (3)$$

Experimental

Friction modeling and compensating is an interesting and rising issue for precise control of a mechanical system e.g. micro assembly systems. It is desirable that nonlinear frictional behavior must be known for positioning and tracking systems. An electric motor with attached assembly has been considered as test bed to measure the coulomb and viscous friction. The motor used in the experiment is the model GM8224D091-R10 from PITTMAN MOTORS. In our application, the motor is operated in "torque mode," so the motor acts as torque source and accepts an analog voltage as a reference of torque signal. In this configuration, the motor is capable of delivering a maximum torque of 0.18 N m. The motor rotor is attached with gear mechanism at 31:1.

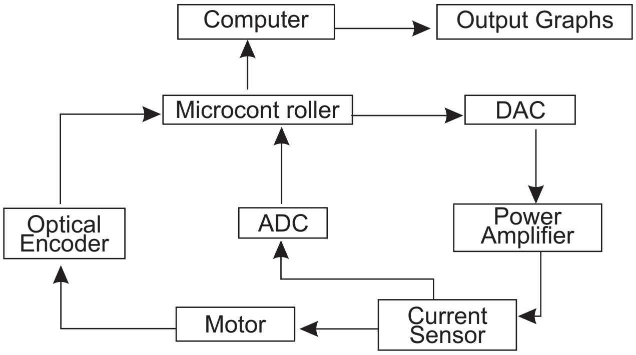

Fig. 1. Block diagram of experimental setup

Firstly the motor is rotated under no load condition and the coulomb plus viscous friction parameters are evaluated. The Microcontroller 89C51 is used to derive the motor through DAC0808. DAC0808 accepts the 8 bit digital signal from microcontroller to give an analog output to power amplifier. Operational amplifier LM324 combine with TIP120 is used to amplify the analog signal given by DAC with high output current. The output voltage of power amplifier is given to drive the motor. A current measuring circuit of Op Amp LM324 is installed between power amplifier and the motor to sense current in terms of voltage, which is given to microcontroller after converting the analog voltage to digital signal by ADC 0804.

Position information is obtained from an optical encoder located on the motor which has a resolution of 504 pulses/rev (accuracy 0.714286 [deg]). Microcontroller accepts the encoder pulses in interrupt mode for every 20 millisecond and transfer to computer through parallel port interfacing. The Turbo C program receives all the data send by microcontroller and save that data in .txt file. The Mat lab program uses that .txt file and plots the velocity graph. The velocity graph under no load conditions are plotted from where coulomb and viscous friction parameters are obtained as in [3].

Coulomb and viscous friction model in simulink

Viscous friction and coulomb friction are by far the most popular ingredients in friction models utilized for control of mechanical systems. Both produce forces against the relative motion between the bodies in contact, but the viscous friction is proportional to the speed while coulomb friction depends on the sign of velocity. These components of the friction are some of the major limitations in performing high–precision in positioning and motion of mechanisms. While these friction models are simple because they establish that friction force or torque depends on the instantaneous velocity.

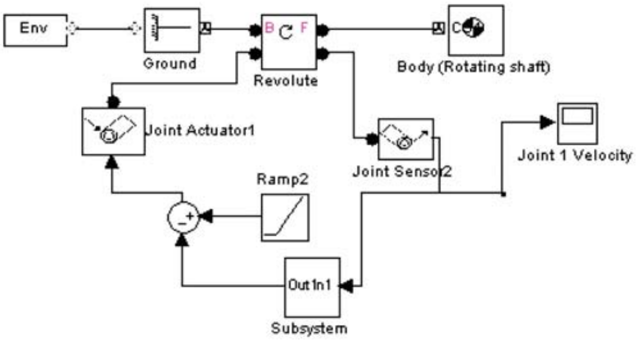

Fig. 2. Simulink block diagram of physical model

The environment block represents the surroundings of physical model (machine) to which it is connected. The gravity, dimensionality, linearization, visualization and constraint solver type are explained in this block. The ground represents the fixed location of joint in the world coordinate system. The revolute joint represents one degree of rotational system. The follower body rotates relative to the base body about a single rotating axis going through collocated Body coordinate system origins. The revolute joint is rotating about z-axis. There are two ports on the revolute joint for sensor and actuator. Base follower sequence and axis direction determine sign of forward motion by right hand rule. The ramp input with slope 1 and start time 2 sec is applied to the system. The sign block is used that give different values depending on sign of velocity. It gives +1 for positive velocity, 0 for zero velocity and -1 for negative velocity. Actuator actuates the joint primitives with generalized force/torque motion signals. Joint sensor measures linear/angular position, velocity, acceleration and computed reaction force/torque on a joint primitive. The angular velocity sense by sensor is visualized by scope. The Simulink block diagram model is shown in Fig. 2.

ADAMS/view

ADAMS/View is a powerful modeling and simulation environment used to build and simulate the physical model shown in figure. The model is tested in the same environments as actual product experienced, validated against test data, refined model with flexible parts, realistic geometry, and controls. Finally, the important parameters that affect the product design are analyzed and find the right combination of design values to arrive at the best product design, saving time and money.

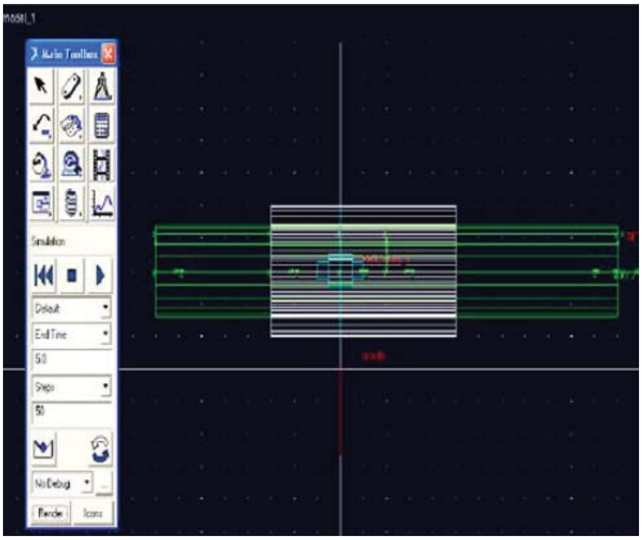

Fig. 3. Constraint Model Built in ADAMS/View

Experimental results

Velocity graphs with same applied torque under no load

The multiple velocity graphs are plotted at same ramp input and analyzed the results. It was found that it's really difficult to analyze the frictional behavior in the joint. The velocity graphs are plotted under no load conditions as shown in Fig. 4.

The graph in Fig. 4 below shows that under similar conditions, same applied torque, the gear ratio and same configuration parameters, the motor behaves slightly differently due to nonlinear frictional behavior. The graph in Fig. 5 also depicts the same fashion.

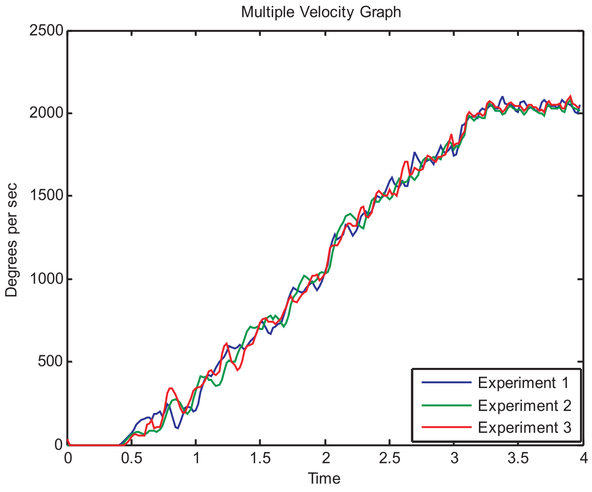

Fig. 4. Combined experimental under result under no load conditions

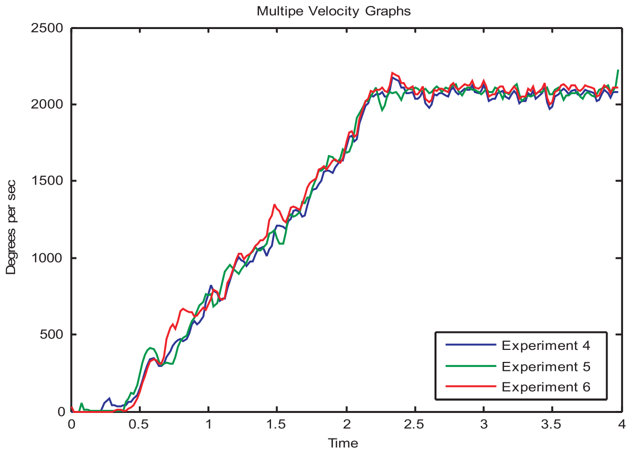

Fig. 5. Combined experimental result with no load conditions

There are many reasons that are assumed to play effective role in the different behavior. The applied torque that is ramped slightly varies because it is given through microcontrollers and current values differ at different experiments. The contact between gears disperse away as applied torque increases, it act differently under similar conditions. The stator resistance, rotor resistance and leakage of induced current are one of many reasons that cause motor to behave differently.

In this plot the velocity parameters are as calculated by the software

$a = 949.5407$
$b = 227.8898$
$m = 0.61$

Putting these values in the friction formula

$$fc = b/a\, m \tag{4}$$

$f_c = 0.1464$ N.m

$$fv = m/a \tag{5}$$

$f_v = 6.4\text{x}\, 10^{-4}$ N.m
$f_{\text{noload}} = 0.147$ N.m

Velocity graphs with same applied torque loaded

The velocity graphs of loaded motor with same applied torque and under same conditions are shown in Fig. 6.

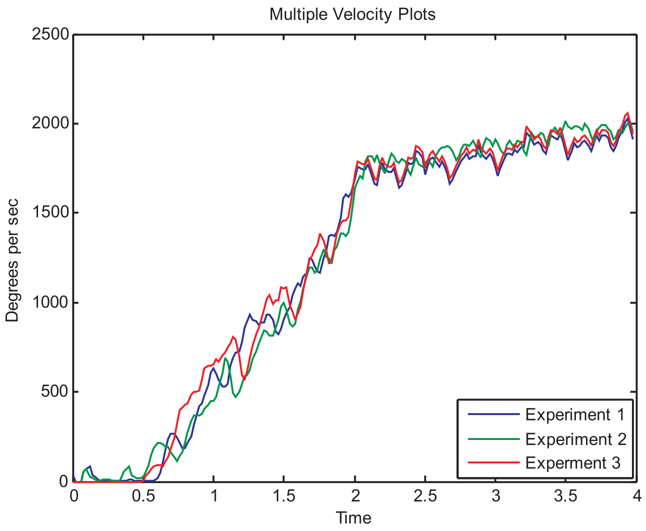

Fig. 6. Combined experimental result with load conditions

$a = 981.3603$
$b = 412.1713$
$m = 0.55$

Putting these values in the friction formula

$f_v = 5.6\text{x } 10^{-4}$ N.m
$f_c = 0.231$ N.m
$f_{load} = 0.231$ N.m

Friction between Load Shaft and Revolute Joint

From the behavior of the motor velocity plots Friction has been determined on both No Load and Loaded conditions. So the friction between revolute joint and load shaft will be as

$f_{joint} = f_{load} - f_{noload}$
$f_{joint} = 0.231 - 0.147$ N.m
$f_{joint} = 0.084$ N.m

It is observed that the velocity graph of loaded motor behaves slightly different under same conditions. The difference in this case is more as compared at no load condition. The reason is because of asperities contact of attached shafts. The asperity junctions act as non linear springs and as the applied torque increases the asperity junctions try to break up. If the applied torque changes its direction then velocity reversal occurs and as a result hysteresis loop is formed.

The first phase during which velocity approximately remains zero is a presliding regime or the micro-slip regime in which the adhesive forces (at asperity contacts) are dominant such that the friction force appears to be a function of displacement rather than the velocity. This is so because asperity junctions deform Elasto-plastically (depending on their individual loading) thus behaving like nonlinear springs. As the displacement increases more and more junctions will break resulting eventually in gross sliding regime. This "break-away" displacement may depend on diverse characteristics of the contact and surface texture: topography, hardness, surface layer metallurgy, it's the second phase. Lastly, as asperities breakup the velocity continuously starts increasing till the applied force approaches to its maximum, there the velocity remains constant.

Comparison of experimental, simulink and adams/view results

The comparison of experimental results with Simulink and Adams/View are shown in Fig. 7. The results are plotted in Mat lab by taking numerical data. As shown in figure 4 at point 3 seconds the trends of graph are almost similar. The discontinuities in experimental results are due to many reasons that are as

The sensor used to sense motion of shaft captures noise during running:

1. The motor starts for 20 milliseconds and stops for 1 microsecond, due to continuous start and stop of motor the irregular behavior appears.
2. The stator coils magnetized and demagnetized when motor is on and off.
3. The induced voltage stored in the coils when motor is off.

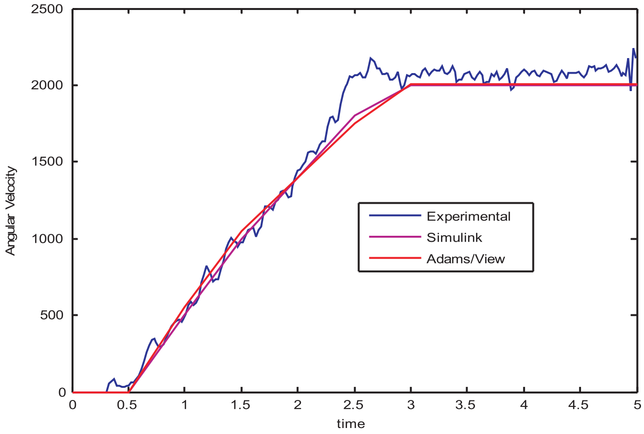

Fig. 7. Comparison of experimental, simulink and adams/view results

The graphs in figure 4 behaves differently after 2.5 seconds, the Simulink and Adams/View results are similar while experimental results vary. The variation is due to:

1. Experimentally the data points are determined and velocity graph is obtained.
2. In Simulink the block diagrams are used that make differential equations, those equations are then plotted to obtain velocity graphs.
3. In Adams/View the physical model is drawn defining constraint and forces. The physical model is run under the optimal region and velocity graph is obtained.

The difference in the results is due to different software limitations but the trends agree with each other. The improved results are shown in figure 5.2

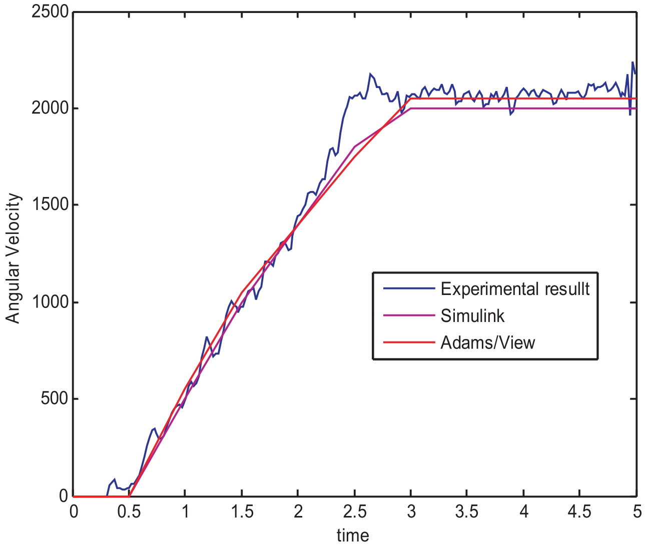

Fig. 8. Comparison of experimental, simulink and adams/view results

The error between Simulink and Adams/View is within ± 2 %, between experimental and Adams/View is ± 7%, between experimental and Simulink is within ±10%.

Results and discussion

The coulomb plus viscous friction model has been presented. It is a simple model that captures the static friction regime and the linear behavior of friction after crossing static regime when object starts sliding means in dynamic regime. The coulomb plus viscous friction parameters are present in all friction models that shows their essence. Both parameters have been found from the velocity response of the system. Then there is a comparison of experimental results, software simulation in Sim Mechanics and ADAMS that agrees. The coulomb plus viscous parameters did not describe the complete picture of the frictional effects as dynamic models require for further treatment during motion [7]. A much more should have to be completed for accurate results. If the motor rotates the cylinder in reversal direction then for it very high revolution encoder is require to determine the motion and lag time. The Elasto-plastic friction models, hysteresis loop formation at velocity reversal, Stribeck effect, stick-slip and asperities contact which act as spring that deformed and relaxed during velocity reversal are the areas on which work has to be done.

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