Intelligent Precision Position Control of Elastic Drive Systems

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***Abstract*—The use of elastic transmission elements, such as timing belts or plastic lead screws, for precision position control applications is proposed in this paper. The characteristics of elastic systems are analyzed and transmission elasticity and friction are identified as the main obstacles in obtaining high precision positioning. The development of a fuzzy-logic based control system is described in detail. The experimental results obtained demonstrate the feasibility of such an approach in obtaining positioning accuracy better than 25 microns.**

***Index Terms*—Belt-drive, electric-drives, elastic-link, friction-compensation, fuzzy-logic, precision-position-control, servo-control.**

I. Introduction

PRECISION positioning systems are used in a wide variety of applications such as CNC machines, surface mounting devices, vision systems and in the semiconductor and biomedical industries. Almost all of these systems use a rotary actuator, such as a brushless DC motor and convert the rotary motion to linear motion using mechanical power transmission elements like belts, chains, ball-screws or lead-screws [1]. The choice of the drive system is governed by the positioning accuracy and repeatability required by the application, the length and maximum speed of travel and the load carrying capacity required [2]. At present, high precision systems requiring positioning accuracy and repeatability of less than 100 microns have to use rigid or stiff elements such as lead-screws and ball-screws, since comparable performance cannot be obtained with elastic transmission elements such as belts, chains or plastic lead-screws. Further improvement in accuracy is obtained by using direct load position feedback using linear encoders [3].

If positioning systems using elastic elements can be developed so as to provide performance close to that obtained by rigid elements, significant reduction in system cost can be achieved. Total system cost can be reduced by up to 50% using elastic elements [2]. The major obstacle to the use of elastic elements is the presence of significant nonlinearities in the system due to the elasticity, friction and backlash. This calls for a more elaborate control system which can compensate for the elasticity and friction to improve system performance.

In this paper, we propose the use of load position feedback, elastic transmission elements and high performance intelligent control systems for high precision position control (HPPC) systems. Positioning accuracy better than 25 microns is achieved using a belt drive over a 1 m long stage. To the best of our knowledge, such a level of performance has not been reported to date.

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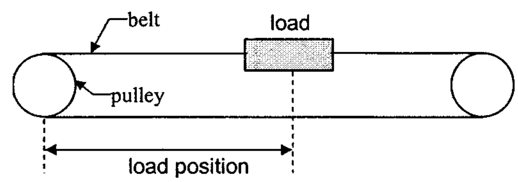

Fig. 1. Outline of a belt drive system.

The next section describes the characteristics of elastic transmission elements and compares them to rigid elements. Section III describes the requirements of the control system and the controller used in this work, while Section IV details the characteristics of a test system used as a demonstration prototype. Experimental results are presented in Section V followed by a description of the work in progress in Section VI.

II. Characteristics of Elastic Elements

Fig. 1 shows an outline of a typical single-axis positioning system. The actuator (motor) is connected to one of the pulleys either using a gear-box or directly, depending on the motor speed. The torque applied by the motor on the pulley is transferred to the stage (load) by the transmission element, which can either be a belt or a chain.

A state space model of the system shown above can be formulated using Newtonian mechanics, assuming for simplicity that the belt can be represented by a spring and damper with constant parameters. Such a model has four states: stage position and velocity and the angular position and velocity of one of the two pulleys. In comparison, a state space model for high accuracy precision positioning systems using rigid couplings has only two states corresponding to the rigid body dynamics: stage position and velocity [4].

However, the assumption that a belt can be modeled as a spring and damper with constant parameters does not hold in practice. This is because the characteristics of elastic elements change considerably as a function of the operating conditions, such as stage speed and belt tension, and over a period of time due to aging. On the other hand, properties of rigid elements do not change over time or operating conditions. Rigid elements are manufactured with a high degree of precision rendering a predominantly linear system, while an elastic drive system is highly nonlinear. The elastic transmission element is stretched before it moves from standstill. This effect introduces an additional dead-band whose magnitude depends on the direction of travel and the absolute position of the stage from the driving pulley. Other significant characteristics of elastic elements are the presence of low frequency oscillatory modes resulting in transverse and axial belt vibrations. These modes make it difficult to obtain high control bandwidth, and the variation of these modes with belt tension, load speed etc. causes additional significant nonlinearities [5].

Additionally, elastic drive systems differ from rigid systems with regard to the load carrying capacity and the maximum speed of operation. Lead-screw and ball-screw based systems provide very high thrust capacity (up to tens of thousands N) while elastic systems are limited to about 1000 N [2], [6]. On the other hand, elastic systems can be used for high travel speeds almost up to 3 m/s, while rigid systems are limited to about 0.5 m/s, mainly due to the limit on the maximum recirculating speed of the ball-nut system. Lead-screws also tend to whip when they are rotated too fast, resulting in damage to the screw [2]. This speed limit decreases as the length of travel increases, making elastic systems more attractive where longer travel length is required.

Thus, if the performance of elastic systems can be improved, they can be used in a number of applications where the positioning accuracy required is of the order of 25–100 microns and where the thrust required is moderate. Such systems can overcome the speed and travel length limits imposed by rigid systems.

III. Control System

A. Sensors and Measurements

A single-axis positioning system similar to that shown in Fig. 1, but using rigid transmission elements, uses an encoder at the motor shaft for use with a feedback control system. Moderate accuracy (around 100 microns) can be obtained using such an approach. Since the feedback device and the actuator are located in the same place, this is called colocated control. This assumes that the load position corresponds directly with the motor shaft position, i.e., the system is fully observable from the motor shaft. However, stage bearing friction and backlash occur after the motor shaft and can significantly impair the performance. For positioning accuracy and repeatability better than 100 microns, a linear encoder is usually added to obtain load position information, thus giving full state feedback [3]. Using two encoders, accuracy and repeatability below 1 micron can be obtained. Fig. 2 shows such a system in block diagram form.

For elastic systems, due to the presence of significant dynamics downstream of the actuator—the belt, bearing friction and backlash, a linear encoder has to be used to obtain load position feedback. Without load position feedback, the system is unobservable so that the load position as indicated by measurement from the motor shaft encoder can be significantly different from the actual load position.

This difference can be 100 microns or more depending on belt elasticity, tension and direction of travel. Hence, a linear encoder for load position feedback is necessary for high precision

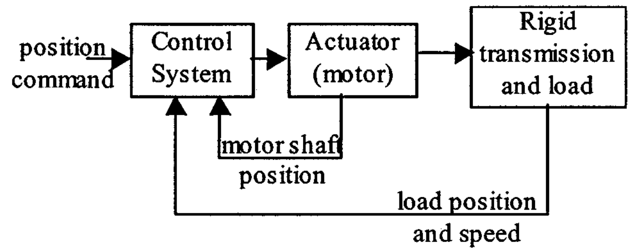

Fig. 2. Control system using two encoders.

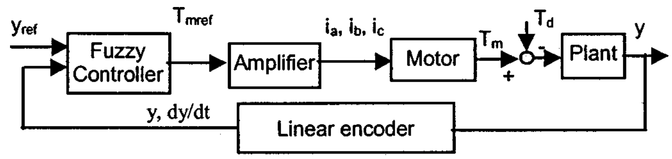

Fig. 3. Block diagram of control system.

elastic systems. Using two encoders—a linear encoder for load position and a motor shaft encoder—increases the cost and complexity of the system. It is hence, advantageous to use only load position feedback, yielding a noncolocated control system. The work described here uses such an approach. In addition to the nonlinearities present in the system, the noncolocation of the actuator and feedback device increases the challenge posed to the control system.

B. Control Algorithm

Considering the characteristics of the control problem, a suitable control algorithm needs to be carefully chosen. Due to the difficulties present in obtaining an accurate model of an elastic drive system and the presence of significant nonlinearities, linear control systems like PID control fail to provide satisfactory performance. This is demonstrated in Section V. Full-state feedback control methods such as linear quadratic regulators and H-infinity or mu-synthesis require the use of additional encoders to measure the angular position and velocity of either the driving or the driven pulley. Even if full-state information is available, the linear techniques will be inadequate to compensate for the system nonlinearities. Thus, a nonlinear controller is required to obtain good performance.

Many nonlinear control system techniques are available. Most of these, however, depend on the use of an accurate system model. On the other hand, intelligent control systems using fuzzy-logic are based on system input–output data and expert knowledge about the plant. This makes them suitable for use in elastic drive systems. In addition, fuzzy-logic controllers can be easily made to adapt to the gradual changes in system characteristics over time, and can be tuned intuitively either with a man-in-the-loop or using self-tuning methods. Hence, a fuzzy logic based controller is selected for this application.

Fig. 3 shows a block diagram of the control system. The inputs to the fuzzy controller are the position reference, y_{ref}; the load position, y; and the rate of change of load position, dy/dt. The output of the controller is the motor torque reference, T_{mref}, which is fed to a servo amplifier. The servo amplifier, in turn, controls the motor currents and hence, the torque of

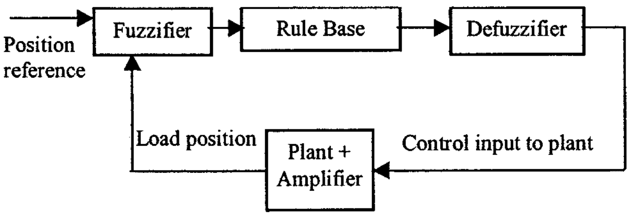

Fig. 4. Components of the fuzzy controller.

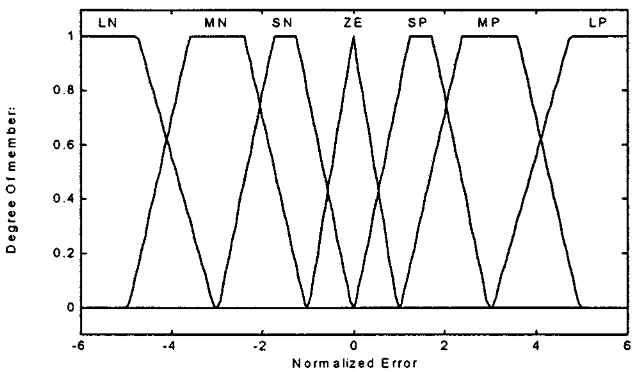

Fig. 5. Fuzzy membership functions for position error.

TABLE I
RULE BASE FOR THE FUZZY CONTROLLER

		Change of Error						
		LN	MN	SN	ZE	SP	MP	LP
ERROR	LN	LN	LN	LN	LN	MN	SN	ZE
	MN	LN	MN	MN	MN	SN	ZE	SP
	SN	LN	MN	SN	SN	ZE	SP	SP
	ZE	LN	MN	SN	ZE	SP	MP	LP
	SP	SN	SN	ZE	SP	SP	MP	LP
	MP	SN	ZE	SP	MP	MP	MP	LP
	LP	ZE	SP	MP	LP	LP	LP	LP

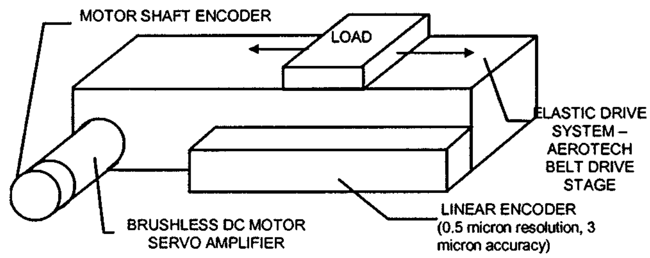

Fig. 6. Block diagram of test system.

the motor, T_m. The load torque is added as a disturbance input, T_d, and the resulting torque acts on the elastic drive system to produce the load position. The load position is measured by the linear encoder.

Since the load position is a time integral of the load speed, the plant has a pole at the origin. Hence, a PD type of fuzzy controller is required. Although, such a control system may result in some steady state errors, any use of integral control action may introduce limit cycles due to the presence of static friction. A standard, two-input fuzzy controller with Mamdani-type inference engine can be seen as a PD controller with variable proportional and differential gains. To use this structure here, the actual inputs to the controller are the load position error $(y - y_{ref})$ and the change of position error over one sampling period of the controller.

A fuzzy controller consists of three main blocks, namely, the fuzzifier, rule base and defuzzifier as shown in Fig. 4. The *fuzzifier* takes crisp values for the two inputs: *position error* and *change of position error*, and calculates the membership values for all the fuzzy sets defined for these two inputs. A singleton fuzzifier is used with trapezoidal and triangular membership functions to obtain the fuzzified values of position error and change of position error. Seven fuzzy sets are used, and the membership functions for position error are shown in Fig. 5, where the fuzzy set names stand for linguistic fuzzy variables *LargeNegative (LN), MediumNegative (MN)* etc. The memberships are grouped together near zero and are spaced apart at the limits, to obtain better resolution and control.

The fuzzified values of error and change of error are then used in a fuzzy composition operation to obtain the fuzzy set membership values for the controller output. This composition operation uses the *Rule Base*, which is a set of IF … THEN rules that maps the inputs to the output. The output of the rule base is a set of membership values for the controller output which are then converted to a crisp value using center of area defuzzification in the *defuzzifier*. The membership functions for the controller output (torque reference) are similar to those for the error and change of error.

A simple symmetrical rule base of 49 rules is used to map the error and change of error to the controller output. The membership functions are selected so that for a given crisp input, two fuzzy sets will have nonzero memberships. This results in four rules being fired simultaneously. This characteristic is effective for localized tuning of the rule base. Table I shows the rule base used.

IV. TEST SYSTEM

The test system has been built by assembling various, industry standard components. A reinforced polyurethane timing belt is used as the elastic transmission element on a one meter long single axis belt drive stage manufactured by Aerotech, Inc. A load of up to 10 kg can be carried on the stage which is moved by the belt and rests on ball bearings. The actuator used is a brushless DC motor controlled by a servo amplifier using a 20 kHz PWM inverter. Both the actuator and the motor are manufactured by Aerotech. An optical encoder, mounted on the motor shaft, is used only for analysis and design of the control system. The motor angular velocity is not used in the controller. The load position under control is measured by an optical linear encoder, with the read head mounted on the stage and the scale running along the length of the stage. The resolution of the encoder is 0.5 microns when used with a quadrature decoder and its accuracy is 3 microns over the 1 m length. The encoder can operate at a maximum speed of 1.5 m/s. Fig. 6 shows a block

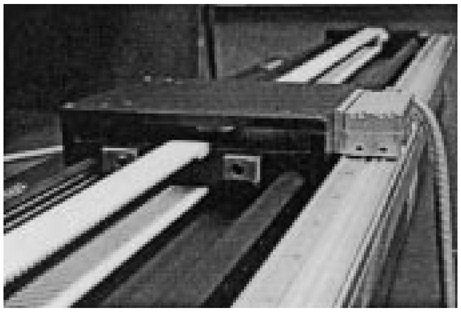

Fig. 7. Picture of test system.

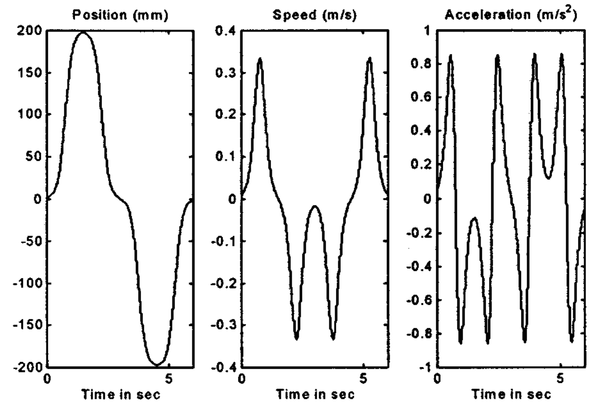

Fig. 8. Sigmoidal track for testing velocity reversal and repeatability.

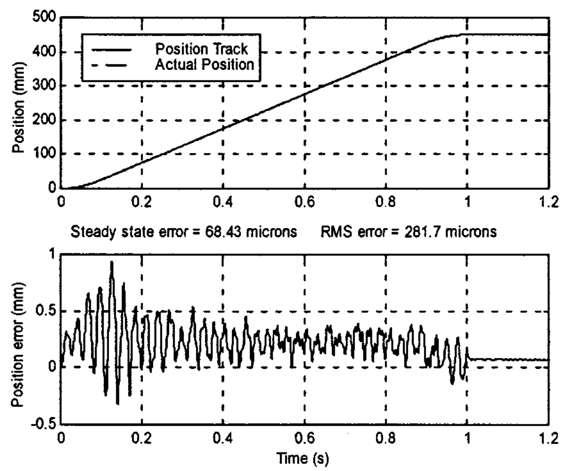

Fig. 9. Response to step-like track (450 mm @ 0.5 m/s, 5 m/s^2).

diagram of the system, while Fig. 7 is a picture of the actual system.

The control system is implemented on a PC using a 90 MHz, Pentium processor. Digital, analog I/O and quadrature encoder interface are obtained using ISA plug-in cards. The quadrature encoder interface is capable of decoding pulses of frequency up to 1 MHz, which translates into a speed of 1 m/s for the load carrying plate. The control software is written completely in C and is run on DOS to ensure real-time operation of the control loop, which is updated every 1 ms.

V. Experimental Results

The test system is run using the developed fuzzy controller. Based on the results, the controller membership functions and rule base are tuned manually. Automatic tuning of controller parameters is currently under development.

A. Control Objective

The load position feedback is not reliable for speeds above 1 m/s. If the stage speed exceeds 1 m/s, the stage position as recorded by the controller remains constant, resulting in an increase in tracking error. This results in a higher motor torque being applied which accelerates the stage further, resulting in a runaway condition with the stage hitting the end stops of the test setup. Due to this limitation, any control objective has to be defined in terms of the closed loop system response to a desired position track. The test system is designed for a maximum system acceleration of 5 m/s^2. Hence, position tracks are generated with the desired acceleration and speed limited to the physically possible values. Two kinds of position tracks are generated to test the performance. To obtain the step response of the closed loop system, a step-like reference track is generated with the acceleration limited to 5 m/s^2 and the speed limited to 1 m/s. The control objective is then defined as follows:

- An RMS tracking error of the order of 300 microns (or 0.3 mm) over any position track, and,
- A final positioning error of less than 25 microns

Another test is carried out using a sigmoidal position track to test the system for bi-directional motion, velocity reversal and repeatability. An example track for this test is shown in Fig. 8.

B. Step Response

The step-like tracking test is carried out for different travel lengths at different maximum speeds, and in both directions. The results show that the final error is of the order of 75 microns, while the tracking error is of the order of 300 microns. The final steady state is reached without any significant settling time, after the position track reaches its steady state. Fig. 9 shows a typical result obtained for a displacement of 450 mm at a speed of 0.5 m/s and with a maximum acceleration of 5 m/s^2. The top half of Fig. 9 shows the position track and the actual stage position on the same scale (as shown on the plot legend). Due to the small tracking error (compared to the scale of the plot), the error between the track and the actual position cannot be discerned on a scale of 450 mm. Hence the positioning error is plotted on a smaller scale in the bottom half of Fig. 9.

The steady state error obtained varies considerably over different trials, and also depending on the initial position of the stage. It is also observed that the error is considerably higher when the stage is moved in the negative direction (away from the driver pulley). This indicates the effect of the belt dynamics and the stage bearing friction which varies widely depending on the position of the load along the stage. To overcome these effects, a fuzzy friction compensator is developed based on the technique proposed in [7].

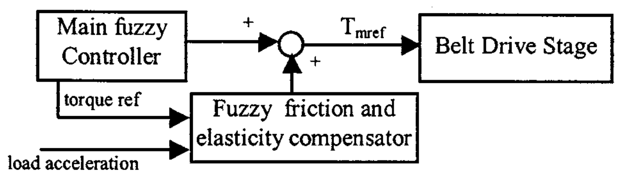

Fig. 10. Controller with fuzzy friction and elasticity compensator.

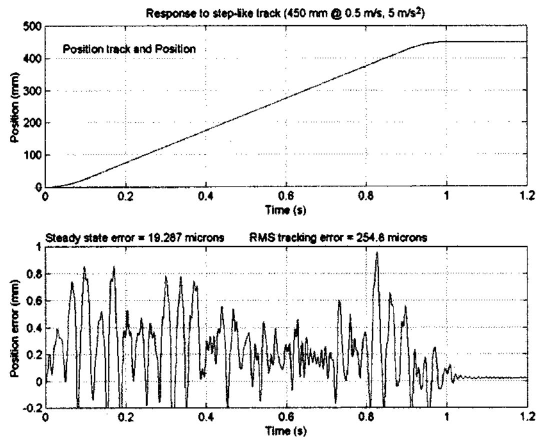

Fig. 11. Step response with friction and elasticity compensation.

C. Friction and Elasticity Compensation

Many methods have been proposed for friction compensation in high precision control systems [8]. *Ostertag et al.* [7] show that a fuzzy friction compensator is equally or more effective than other more complex disturbance observer-based methods. Here, the belt elasticity adds another nonlinearity making the fuzzy compensator more suitable. The compensator compares the motor torque reference with the acceleration of the stage. If the resulting acceleration is smaller than expected but in the proper direction, the motor torque reference generated by the main fuzzy controller is increased by a suitable amount. On the other hand, if the direction of acceleration is opposite to that of the torque reference, then the torque reference is not increased. Thus, the inputs to the compensator are the filtered stage acceleration and the output from the main fuzzy controller. The output of the compensator is added to that of the main controller to obtain the motor torque reference T_{mref} as shown in Fig. 10.

The rule base for this compensator is also symmetrical. However, the added torque reference is completely dependent on the *actual* acceleration of the load, which makes it asymmetrical. Since the direction of acceleration is more important than its magnitude, it is fuzzified into 3 sets while the main controller torque reference is fuzzified into 7 sets. The output of the compensator has 5 fuzzy sets and a crisp output value is obtained using a center of area defuzzifier.

Fig. 11 shows a typical result obtained using the fuzzy controller with the friction and elasticity compensator. The top half of the figure has the position track and the actual position plotted on the same scale. The steady state (final positioning) error is reduced considerably and is consistently less than 25 microns for numerous tests performed with different initial positions and maximum speeds in both directions of motion. The rms tracking error is about 250 microns.

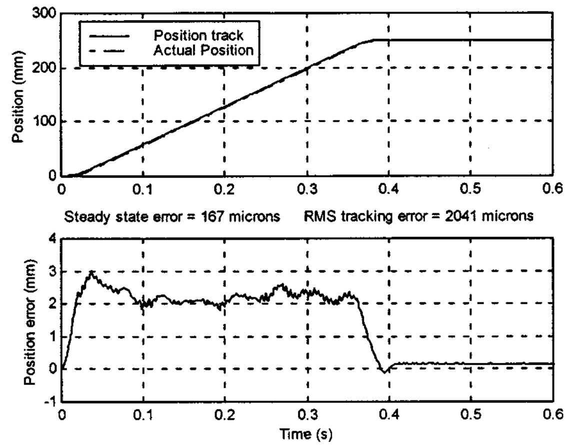

Fig. 12. Response to step-like track using PD control (250 mm @ 0.35 m/s).

D. Comparison with PID Control

It is desirable to compare the fuzzy controller with a conventional PID controller. Fig. 12 shows a typical response obtained using PD control for a step-like track with a displacement of 250 mm, maximum speed of 0.35 m/s and maximum acceleration of 5 m/s^2. It can be seen that the tracking error is 2041 microns, which is about 10 times that obtained with the fuzzy controller. The tracking error increases as the track speed increases. The steady state error is 167 microns which is more than double that obtained with the fuzzy controller and more than six times that obtained with the fuzzy controller when the compensator is used.

To reduce the steady state error, integral control was added. However, due to static friction and belt stretching, the stage oscillates around the final goal position without ever settling down. This limit cycle has an amplitude of about 200 microns. Thus, a PID controller results in considerable degradation in performance.

E. Sigmoidal Tracking

Fig. 13 shows the response of the system to a symmetric sigmoidal track having a displacement of 200 mm in either direction. The maximum track speed is 0.33 m/s. The rms tracking error is similar to that obtained for step response while the steady state error is slightly higher. It should be noted that the plot shows data for 7 seconds as compared to earlier plots which show data for 1.2 seconds, which results in the feeling that the system response includes very high frequency oscillations.

F. Repeatability

For position control applications, control system repeatability is very important, in addition to final positioning error. Repeatability is defined in terms of the variation in final positioning error for various tests at different speeds and for different final

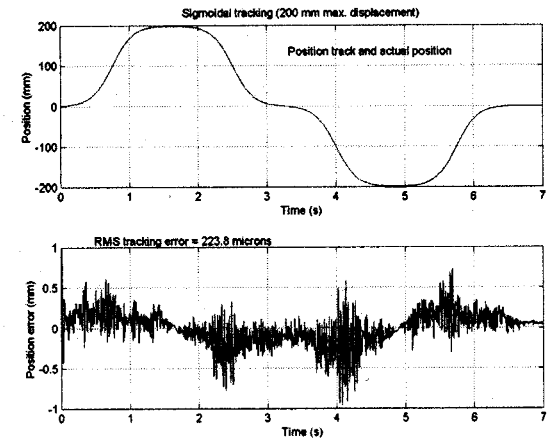

Fig. 13. Response to sigmoidal track with 200 mm displacement.

displacements. This method is used here to obtain the repeatability of the control system described above. For more than 50 step-like tests, the final positioning accuracy is between 5 and 25 microns giving a repeatability of 20 microns.

G. Analysis

The results obtained show that the controller including the compensator is capable of controlling the elastic system to achieve very high precision positioning. High frequency belt oscillation modes show up in the error in the form of an oscillatory component. The dominant frequency components in the error range from 20 Hz to 30 Hz. This frequency reduces as the load speed increases, while the amplitude of the oscillation increases with speed. Abrate [5] has studied the vibration of belt drives in detail and the results obtained here match the conclusions in his paper. Further work is currently in progress at the University of Washington aimed at improving the tracking and the final positioning accuracy.

VI. Conclusion

The feasibility of using elastic drive systems in precision position control applications has been demonstrated in this research. Using two fuzzy controllers—one for controlling the main system characteristics and another to compensate for the friction and elasticity—positioning accuracy better than 25 microns has been achieved for travel lengths up to 1 m. The belt drive system can be controlled to obtain consistent performance at various speeds up to 0.8 m/s, with the speed limit being imposed by the linear encoder and not by the control system. Using this control algorithm, elastic drive systems can be used for applications requiring moderate positioning accuracy and load carrying capacity at high speeds. Such systems can effectively replace rigid drives with significant cost savings for many applications.

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