Instrumentation Techniques and Improved Control of Stepper Motor Driven Machinery

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ABSTRACT.

Continuous path, multi-axis stepper motor driven machinery frequently suffers from performance limitations. This performance limitation manifests itself as microscopic rough motion. It has been hypothesised that rough motion present in stepper motor systems generally arises from the excitation of the machine dynamics through interpolation. The interaction of acceleration with interpolation severely compounds the problem. The 'art of stepper motor system design' suggests that motors typically having a torque capability of double the load requirement should be sufficient to ensure reliable operation. However when full account of interpolation and ramping are considered, yet larger motors may be required. Steiger has shown theoretically that the resulting rough motion of the machine, causes the limitation in performance through an increase in load being reflected back to the motor. The aim of the current research is to confirm the principal causes of this performance limitation through experimentation, and further, to develop control techniques to minimise the effect.

Building upon the work of Steiger, a test platform representing a single axis of a conventional CNC machine was built. Sensors in the form of incremental encoders where attached to the rig at all the motion transfer points. The sensors include a high-resolution linear encoder able to measure the movement of the end effector to a resolution of 0. 2 microns, whilst the shaft encoder mounted on the stepper motor can measure angular movement to a resolution of $1/50^{th}$ of a full motor step. Experiments under a range of variable conditions have been closely examined in order to characterise the resulting continuous path motion of a stepper motor driven machine. A special purpose data acquisition system was enhanced to allow the synchronous capture of pulse timing data over multiple channels. Validation of the performance hypothesis mentioned above was obtained through the instrumentation of a production CNC machine provided by Pacer systems Ltd. , a manufacturer of CNC machinery collaborating in this research.

The information obtained from the test rig, and later from the CNC machine substantiated the theoretical findings of Steiger. However other features of the mechanical system and electronic drives were shown to have an equally important effect on the overall performance. These factors have been examined in detail and ranked as to their significance in typical application.

Experimental and simulation work confirmed the validity of the approach suggested by Palmin concerning maximum torque utilisation for single axis systems. The experimental results supported the solution implemented, a solution to smooth the motion on the secondary axis. Palmin's principles are extended to the continuous path multi-axis case through a processor based velocity smoothing algorithm minimising system excitation and corresponding path errors. This solution was realised in the form of a novel inverse velocity filter algorithm implemented on a DSP. The research then continued towards developing a technique of measuring the true stepper motor's performance. This method of deducing the static and dynamic rotor position can form the basis of calculating the torque/speed graph for the stepper motor driven system under test. This method was used to measure the effectiveness of the filtering solution.

Thus the significant performance limitations of continuous path stepper motor driven machinery have been demonstrated. These experiments concerning the motion of CNC machinery have resulted in new knowledge being developed within the field of engineering. The work has successfully shown the validity of both Steiger and Palmin. Steps to overcome the performance limitation have led to the development of a novel inverse velocity control algorithm. This solution was realised in hardware and reduces the excitation of the machine dynamics significantly. The research has verified the path following accuracy of the solution through step error measurement demonstrating increased performance together with increased accuracy. To summarise the research has shown that the engineer can achieve a higher level of performance in terms of path accuracy and top speed if a technique such as the one presented is utilised in the control system.

The work described in this thesis is the author's own unless otherwise stated, and is as far as he is aware, original.

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GLOSSARY.

AC:	Alternating current.
AMC:	Automated Machining Cells.
Approximation :	Finding a curve for a given sequence of data points which pass close to the data points, as opposed to Interpolation.
APT:	Automatically Programmed Tool. A software compiler for simplifying numerical control programming.
ASIC:	Application Specific Integrated Circuit.
BLU:	Basic length Unit. A basic length unit is the smallest defined movement of a machine tool in one direction.
CAD:	Computer Aided Design. Using a computer to support and increase the productivity of the design process.
CAM:	Computer Aided Manufacture. Using a computer to create an NC/CNC command file. This tool determines and adds motion parameters for the machining process as well as instructions for the appropriate tool type and auxiliary device control.
CISC:	Complex Instruction Set Computer. Microprocessor with a large number of instructions.
CNC:	Computer Numerical Control. The process of controlling a machine via numerical instructions that are derived from a computer application. Usually it is dedicated to a single machine.
Continuous Path:	A machining process which consists of path segments that when formed together form a continuous path. Relies on synchronised axial movements as opposed to Point to Point which does not.
CPU:	Central Processing Unit.
DC:	Direct Current.
DCM:	Direct Comparison Method. Interpolation method for geometric path.

DDA:	Digital Differential Analyser. Interpolator for geometric path.
DNC:	Direct Numerical Control. There are two meanings for this acronym. Originally it referred to the sharing of a mainframe computer for transmitting motion information to the machine being controlled. The more common application is now Distributed Numerical Control in which the host computer transmits entire part motion sequences to the machine tools CNC memory.
DNC:	Distributed Numerical Control (see DNC above).
DSP:	Digital Signal Processor. Fast microprocessor for mathematical applications. Uses optimised mathematical algorithms for calculation. Can perform multiplication, division in one or two clock cycles.
FIFO RAM:	First In First Out Random Access Memory. Specialised memory buffer.
FMS:	Flexible Manufacturing System.
HPGL:	Hewlett-Packard Graphics Language. - A widely used plotter command language.
Interpolation:	Process of finding a curve which passes through a given sequence of data points and/or satisfies some other imposed conditions.
MDCM:	Modified Direct Comparison Method. Interpolation method for geometric path.
MMI:	Man Machine Interface. Important interface between human operator and a tool (usually a computer).
NC:	Numerical Control. The use of numerical data for the control operations such as complex machining. The numerical data is generated through a computer and is input to a machine either by storage medium (tape, floppy disk etc.) or a direct link via a network (Ethernet etc.).
OSE:	Optical Shaft Encoder. Rotary measurement device for closed loop feedback information (displacement). Can be either give out either digital information (pulses) or analogue (sinusoidal).
PC:	Personal Computer.
PCB:	Printed Circuit Board.

PFM:	Pulse Frequency Modulation. Change of frequency (different time periods)
PLC:	Programmable Logic Controller .
Point-to-Point:	Machining operations that are performed at discrete points. No demand on the path between two operation points.
Pull-In Rate:	Sometimes called Starting Rate. Maximum demanded stepping rate to which a motor can respond without loss of steps when initially at rest.
Pull-Out Torque:	A stepping motor must produce sufficient torque to overcome the load torque and accelerate the load inertia. Since the torque of a stepping motor is not constant over the whole frequency (velocity range), this information is supplied in the form of a graph, known as the pull-out torque/speed characteristic. This graph shows the maximum torque exceeds the pull-out torque at a certain speed the motor pulls out of synchronism with the magnetic field and the motor stalls.
Pulse Period Counting:	A method of capturing information from such devices as digital encoders (OSE). Distance is determined by counting the number of pulses arriving with respect to time. Also referred to as discrete differencing.
Pulse Rate Counting:	A method of capturing information from such devices as digital encoders (OSE). Distance is determined by measuring the length of each pulse with respect to time. Also referred to as inverse time or reciprocal timing.
PWM:	Pulse Width Modulation. Change of pulse width by changing the mark to space ratio. Sometimes misused to refer to a method of PFM where the mark period is constant, and the space period is changed to alter frequency.
RAM:	Random Access Memory.
RISC:	Reduced Instruction Set Computer. High performance microprocessor which gains its performance from optimised design and reducing the range of implemented instructions.
Slew Curve:	This is another definition used in reference to the speed/torque characteristic of the stepper motor. The slew curve is another name to describe the Pull-Out Torque mentioned above.

Slippage :	When a stepper motor suffers from localised stalling. The stepper motor stalls but is pulled back into step, albeit with a small displacement error, by the mechanical and/or electrical components.
Stalling :	When the load torque of the machine exceeds the available stepper motor torque then the load is no longer held in place by the torque of the stepper motor. In multi-axis machinery this means that a given axis stops moving and suddenly becomes stationary.
Starting Rate :	Maximum demanded stepping rate to which a motor can respond without loss of steps when initially at rest.
Start/Stop Curve:	Another definition used to describe both the Pull-In Rate and Stopping Rate of a stepping motor. This assumes both are equal, however research has shown that the friction of multi-axis machinery can aid the stopping rate
Stopping Rate:	The maximum stepping rate which can be suddenly switched off without the motor overshooting the target position.
VDU:	Visual Display Unit.

Nomenclature

$X(t)$	Distance with respect to time value t.
$V(t)$	Velocity with respect to time value t.
$A(t)$	Acceleration with respect to time value t.
$J(t)$	Jerk with respect to time value t.
$V_x(t)$	Velocity for X axis for current value of time t.
$V_y(t)$	Velocity for Y axis for current value of time t.
Vo	Resultant constant tangential Velocity for the end effector.
$\vartheta(t)$	Tangential angle with respect to time t.
R	Radius of circle.
K_1,K_2,K_3	Constant for polynomial equation.
S_1,S_2,S_3	Sample point for polynomial equation.
F_n	Natural Frequency.
T	Resultant Torque.
TL	Load Torque.
Tpk	Peak Static Torque.
J	Stepper Motor / load inertia.
T'	Stiffness of the torque/position characteristic.
θ_e	Static Position Error
P	Number of Rotor teeth.
t	Time.
a_1,a_2,a_n	Constant for FIR filter equation.
n	Number of data points in FIR Filter.
Y_t	Output from FIR filter with respect to time t.
X_t	Input from FIR filter with respect to time t.
Vo_d	Current Filtered Output Velocity.
a_x	Constant depending on filter algorithm.
x_d	Input Velocity at distance (d).
d	distance.

Chapter One

Introduction

Chapter One Introduction

1.0 Overview

Stepper motors have been available for many years, but commercial exploitation only began in the sixties, when improved silicon wafer fabrication techniques made available devices capable of switching large DC currents in the motor windings. This feature of switched winding currents gives the stepping motor its unique digital machine properties, which are a considerable asset when interfacing to other digital systems. The rapid growth of digital electronics through the seventies and eighties assured the stepper motors future. Today, where everything is heading towards the digital revolution, there is world-wide interest in its manufacture and application.

The use of stepper motors within CNC machinery has traditionally been considered to be a poor substitute for AC or DC servomotors because of their poor high-speed capabilities. However, advances in motor design and drive systems make stepper motors very attractive actuators in digital control systems. In a simplistic sense stepper motors can be considered to be digital in nature since the rotor moves in discrete steps when the coils are energised with power. The stepper motor offers lower cost than other electrical motors, and with advances in technology, comparable performance. Stepper motors provide excellent torque at low speeds, up to five times the continuous torque of a brush motor of the same frame size, or double the torque of the equivalent brushless motor [8]. These advantages mean that stepper motors are commonly used in a variety of control systems from disk drives & printers to CNC lathes and robot arms.

The advances made with stepper motors are only a small part of the overall advances made within the automated control industry. The driving force behind the advances in automated control came from the military and aerospace industry where there is the constant need for highly accurate machined components. Postlethwaite et al. [1] discuss the need for these advances in terms of machine accuracy, and precision. In the manufacturing industry the need for highly accurate machined components, has resulted in the need for controllers being able to produce smooth continuous path motion.

The research documented within this report is concerned with the use of stepper motors within smooth continuous path multi-axis machinery. The author builds upon the theoretical work carried out by Steiger [2]. Sherkat & Steiger[3] have postulated that problems may arise from the excitation of machine dynamics as a subsequence of acceleration interacting with interpolation. Steiger [2] has theoretically shown that the velocity profile on a slave axis may be very rough. The consequence of this rough motion is lower achievable system performance. Engineers fail to fully realise the extent of the problem and compensate against the performance limitation by designing systems with larger motors and drive amplifiers.

Building upon the work of Steiger [2], this project investigates the performance of continuous path multi-axis stepper motor driven machinery, by measuring the true extent of rough motion within the machinery, identifying key problem areas and by showing how the performance can be improved. An experimental test rig was first developed to carry out velocity profile measurements, this is shown in Figure 1. From the experimentation and literature survey the limitation in performance was analysed, which in turn led to a solution. This thesis documents these ideas, experiments, and theories and concludes with an evaluated solution to the problem of generating smooth continuous path motion.

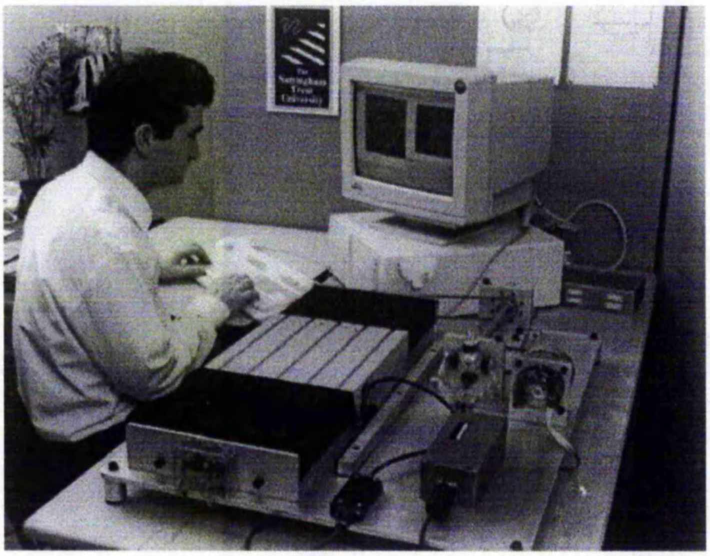


Figure 1: Photograph of Experimental Platform.

1. 1 A Brief History of Computer Numerical Control.

Numerical Control can be traced back in history to the industrial revolution in the nineteenth century when in 1808 Joseph M. Jacquard introduced his first automatic, punch card based loom. He used punched sheet metal cards, arranged on weaving machines in various ways, for the automatic control of weaving patterns. The presence or absence of a hole determined whether or not a needle would be activated. Little progress was made until M. Fournaux, 55 years later patented the automatic piano player under the world-renowned name of pianola, Kief [4].

This was the general pattern of advancement until 1949 when John C. Parsons and the Massachusetts Institute of Technology (MIT) won a study contract from the United States Air Force. After the end of the Second World War the aircraft industry had become aware that machined structural parts would be required in order to meet the strength-to-weight ratio for future supersonic aircraft, Kief [4]. Manual controlled machines could not meet the accuracy requirements, nor could they readily accommodate the many engineering design changes required in the manufacturing of aircraft. These highly geometric, complex shapes were normally machined by the use of French curves or splines; for the same shapes to be machined by the use of numerical control they would need precise mathematical descriptions and voluminous mathematical computations that would take years if computed manually. This is the reasoning behind utilising a computer. The initial study contract would therefore develop a system for machine tools that could control the positions of the leadscrews from the output of a computer. This output would then act as input into a system that would automatically operate the machine tool. The idea was realised to involve the following: The computer would compute the path of a cutting tool and store the computed cutter positions onto punched cards. These would then be fed into the machine control system. This machine control system would continuously output the appropriate data to servomotors, which were attached to leadscrews, in order to drive the cutter over the complex geometry to be machined.

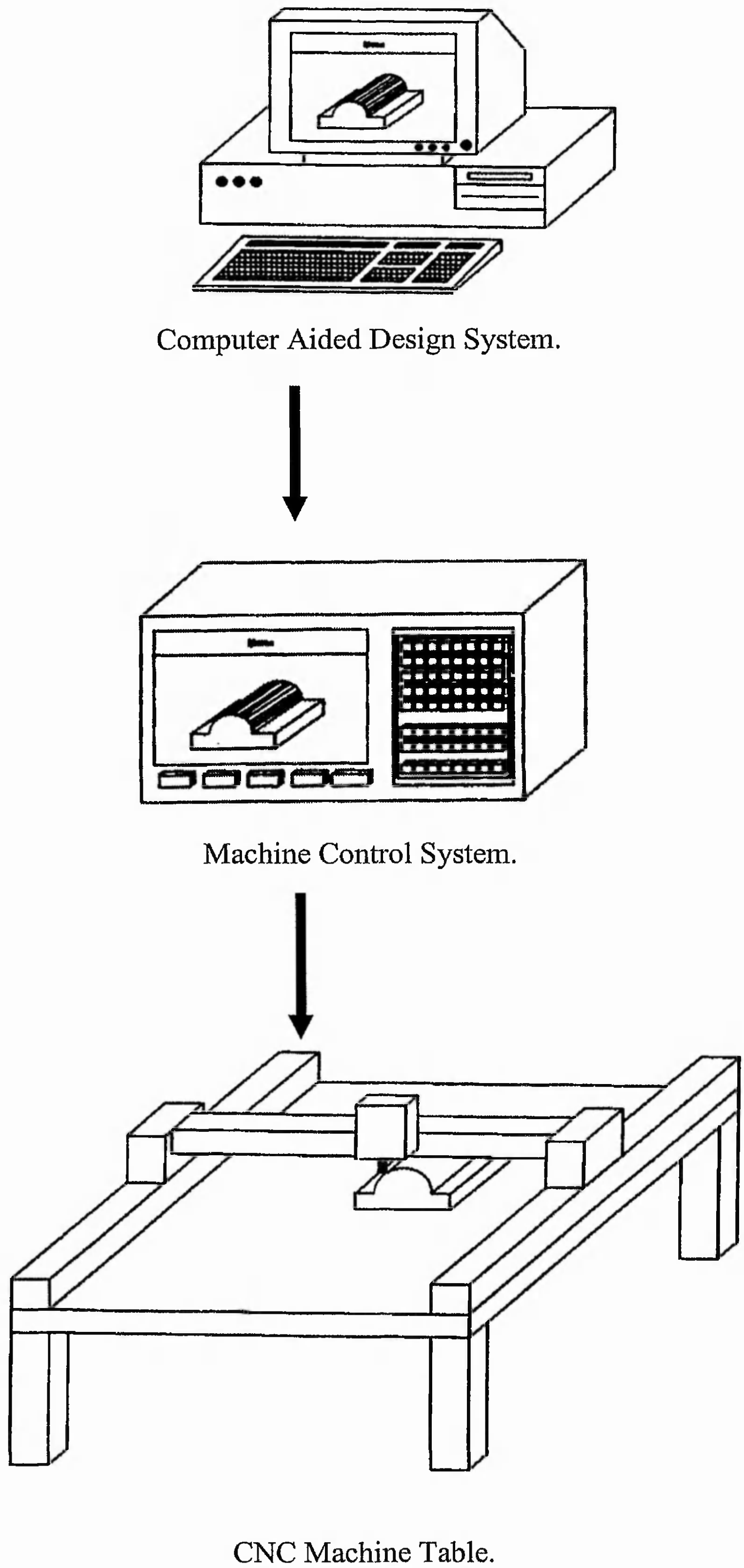


Figure 2 : Concept of a Computer Numerical Controlled Machine.

In 1952 the first numerically controlled machine, a Cincinnati Hydrotel vertical spindle milling machine, was successfully demonstrated at the MIT. The machine control unit was built with electron tubes, controlling three axes with Linear Interpolation and received its data via binary coded punch tapes. The manufacturing industry has changed significantly since then, especially with the invention of semiconductors and microprocessors, but the overriding principle of controlling machines by the process of numbers has not. This principle is known as Numerical Control. The process of transferring the data from computer to machine has progressed from being punched card to housing the computer on the machine and fitting a direct feed from the computer to the control system that would control the motors. This principle is known as Computer Numerical Control (CNC).

The wide varieties of CNC machines offer a range of capabilities. The most important differences lie in the degree of precision attainable and the method used for realisation of the predetermined cutter path. The complexity in geometry of the shapes has also increased in proportion to the capabilities of the machine. The increasingly wide range of CNC applications has led to the requirement for more complex control systems to fulfil these demands. A typical demonstration of the demands placed upon the motion control system can be seen in Figure 3, which shows a complex engraving pattern.

Figure 3 : Complex Engraving Pattern.

1. 2 Overview of Motion Control Systems.

Definitions of motion control vary widely in industry today. Depending on the application, motion control can refer to simple on-off control or a sequencing of events, controlling the speed of a motor, moving objects from one point to another, or precisely constraining the speed, acceleration, and position of a system throughout a move. Essentially they may be divided into three distinct categories: sequencing, speed control, and incremental motion (which again can be sub-divided into point to point and continuous path motion, also described below).

Sequencing refers to the control of several operations so that they all occur in a particular order. Perhaps the simplest example of sequential motion is the progression of events that take place through the mechanical linkages of a player piano (a mechanical piano player). When a player piano plays a tune, holes in a paper roll cause piano wires to be struck in a specific sequence. Similarly, opening and closing valves can be sequenced mechanically with camshafts.

Speed control refers to applications involving machines running at varying speeds or torque's. The source of power for such applications is generally an internal combustion engine, an electric, hydraulic, or pneumatic motor. Speed can be controlled either mechanically or, in the case of electric motors, electronically. Mechanical speed-control components include clutches and brakes, variable-speed drives, traction drives, transmissions, and fluid-coupled drives. There are also mechanical clutches and brakes which are electrically or hydraulically actuated. Examples illustrating where such technologies are used include web presses, filament-winding machines, and feedstock applications.

Incremental motion control, in contrast with velocity control, generally refers to applications where something must move from one point to another at a constant speed. An important requirement in such applications is that there are two factors that must be controlled: speed and distance. Incremental motion control can be split into two specific types, point to point motion or continuous path motion. Point to point motion is normally used for robotics or drilling machines where the machine will start at one point, perform an operation and then proceed to the next point. Straight line machining can be achieved with the appropriate cutter and feedrate. The other type of incremental motion control is continuous path control. Continuous path control, or contouring systems, use synchronised feed drives and therefore can provide accurate positioning along the designated path not only at the end point. The path for such machines is typically expressed as a mathematical function rather than discrete movements. An interpolator is used to calculate path positions up to the target point. This makes it possible to reach any point in the X/Y/Z space plane by rectilinear motion, whilst maintaining the X/Y/Z path ratio. Contouring is used to control such machine tools as lathes, milling machines and machining centres, which remove material over the surface of the workpiece, such as milling of a cam.

In general, the simplest continuous path control system might be found on milling machines. They typically contain a positioning system for moving the fixture holding the workpiece. The end effector in this case will be the milling tool mounted on the Z axis. The positioning system involves a number of axes, both translational and rotational. Each axis contains an electrical motor, an adjustable-speed drive, a positioning mechanism, and possibly a position transducer that reads out the position of each table axis. The positioning mechanism for such a system is usually a ball screw.

Many light manufacturing systems for continuous path motion use a stepping motor which is considered particularly well suited for this type of application. Stepping motors work by sending pulses to a drive circuit which in turn energises coils within the motor. The stepping-motor shaft moves through a specific angle in response to each input pulse to the drive circuit. This allows it to rotate in fixed, repeatable, known increments, unlike conventional servo motors, which require a position transducer.

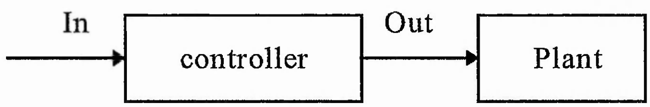
Most motors, regardless of type, usually require controlled acceleration and deceleration for best response, especially when the motor moves a load having significant inertia. To obtain this acceleration and deceleration in stepping motors, controllers typically use a technique called motor-clock ramping where the frequency of the control pulses are varied to accelerate the motor from zero to maximum speed. This variable motor clock rate is often generated through software. An example of this could be a computer that controls the time period between control pulses using precalculated constants stored in memory tables.

In the simple case, an incremental motion system moves a required distance. But for some applications, simply commanding the motor to move a prescribed distance does not provide enough accuracy. Machine slippage, back-lash, and other factors may prevent the motor from moving the load to the point desired. This error, expressed in terms of distance is called the Static Displacement Error. Several factors also effect the velocity of the movement, namely pulley mis-alignment, mechanical resonance and the excitation of any machine dynamics. These factors result in velocity fluctuations, which in turn lead to displacement errors, since velocity is the first derivative of distance and time. These errors are called Dynamic Displacement Errors.

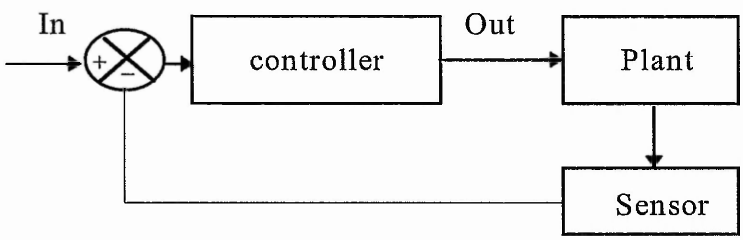
Closed-loop control systems can be designed, as demonstrated by D'azzo[5], & Kuo[6], to reduce these drawbacks. They provide precise position control with feedback, see Figure 4. Feedback minimises the difference between the commanded position or velocity of the system and the actual location or velocity. In other words, feedback minimises position or velocity error. Koren [7] provides a good insight on developing an adaptive control system that can be used for machining. The adaptive system he describes is basically a feedback system that treats the CNC machine as an internal unit, and in which the machining variables automatically adapt themselves to the actual conditions of the machining process. Koren claims that this will enable the machine to run at an optimum level of performance, rather than a level dependant upon the skill of the machine operator.

Closed-loop control systems do have disadvantages, namely, cost, complexity and problems with stability. A more important aspect to note is that feedback like open loop control can only correct errors within the torque capability of the motor. These disadvantages can often out-weigh any advantages gained from such a system. This in turn leads to manufacturers selecting a compromise between performance and cost.

The definition of motion control used in this report is that of incremental motion control, which use electronic open loop control systems and stepper motor drives to provide continuous path motion. Such systems also fall within a limited power range, typically up to about 1KW.



Open Loop Control



Closed Loop Control.

Figure 4 : Open & Closed Loop Control Systems.

1.3 Introduction to Stepper Motor Control systems.

Stepper motors are widely used in industry, especially the space industry, largely due to several advantages they have over the other types of electrical motor used in control systems. Steppers are inexpensive and economical to run. All stepper motors have the following properties; bi-directional control, built-in braking, no drift or cumulative error and direct digital control (the main reason the space industry uses stepper motors). The main disadvantages are transients produced by the constant stepping action and the low speed range. A simplified cross-sectional view of a stepper motor can be seen in Figure 5. The stepper motor consists of two main components; a stator (the outside body of the motor) typically with windings that are energised to set up a magnetic field and a rotor which is normally a toothed spindle composed of a ferrous alloy. There are normally a number of phase windings, typically two, contained within the stepper motor. When one phase is energised the rotor rotates such that a tooth is aligned into a position to provide the least magnetic reluctance. When the next corresponding phase is energised the rotor rotates to the next position, thus by sequentially energising the various phases the rotor moves one rotation.

Stepper motors can be viewed, in a very broad sense, as being digital in nature, moving a finite angular distance each time a pulse is applied to the drive circuitry. They do not necessarily require feedback to provide precise positioning or speed control. Stepper motors provide excellent torque at low speeds, up to five times the continuous torque of a brush motor of the same frame size, or double the torque of the equivalent brushless motor, Parker Hannafin [8]. These advantages mean that stepper motors are commonly used in a variety of control systems from disk drives & printers to CNC lathes and robot arms.

As expressed before, stepper motors are driven by a digital pulse train. The velocity is dependent upon the frequency of this pulse train; the higher the frequency the greater the velocity. The stepper motor, like other motors has inertia and therefore cannot simply be driven at high speeds from stand still. Instead they have to be accelerated up to this speed, and then decelerated to an acceptable level. This can be seen overleaf in Figure 6.

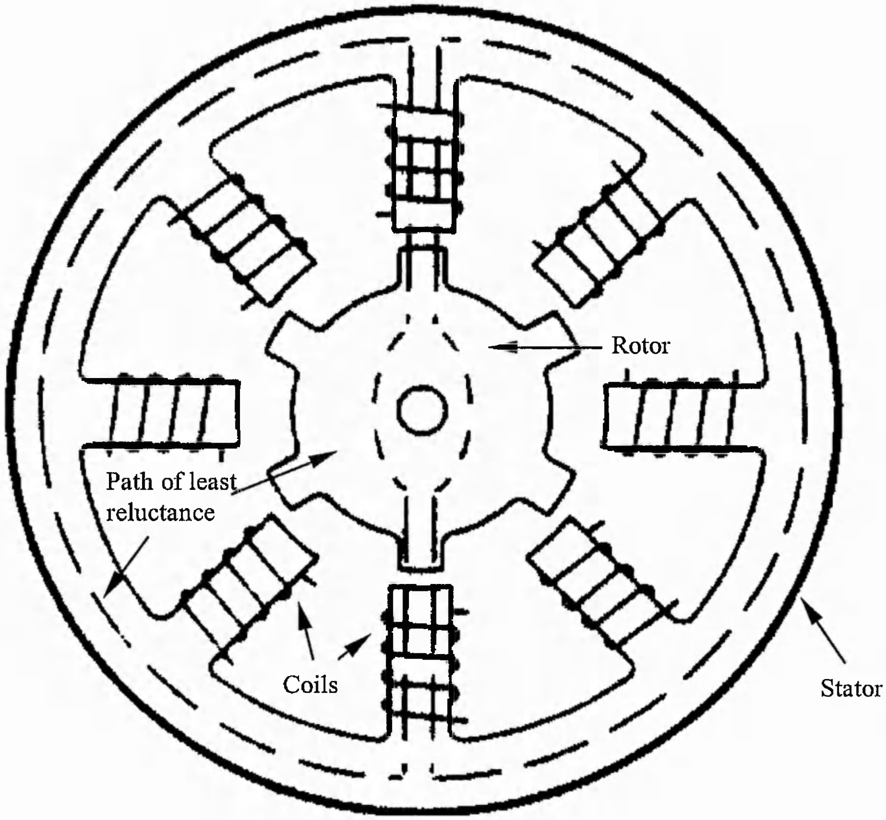


Figure 5 : Cross Sectional View of a Stepper Motor.

There are three main types of stepper motors: permanent magnet, variable reluctance and hybrid. Permanent magnet motors as their name suggests have a rotor that consists of a permanent magnet.

Permanent magnet stepper motors offer low cost, large step angles and low torque. They are typically used in disc drives. Variable reluctance motors do not have a permanent magnet, so the rotor spins freely, without “detent” torque. This type of motor is frequently used in small frame sizes for applications such as micro-positioning. The hybrid motor is by far the most widely used stepper motor in industrial applications. The name is derived from the fact that it is a combination of the previous two motors.

The hybrid stepper motor is controlled by changing the phase current through two sets of magnetic poles. This means there is a total combination of four patterns that the current can flow in before repeating. The current required by a conventional stepper motor is typically 1-10 amperes and therefore some sort of power amplifier is needed between the motor and controller. This power amplifier is typically called the drive system since it provides the driving signal for the motor. The drive system accepts pulses from the controller and converts them into the appropriate phase current.

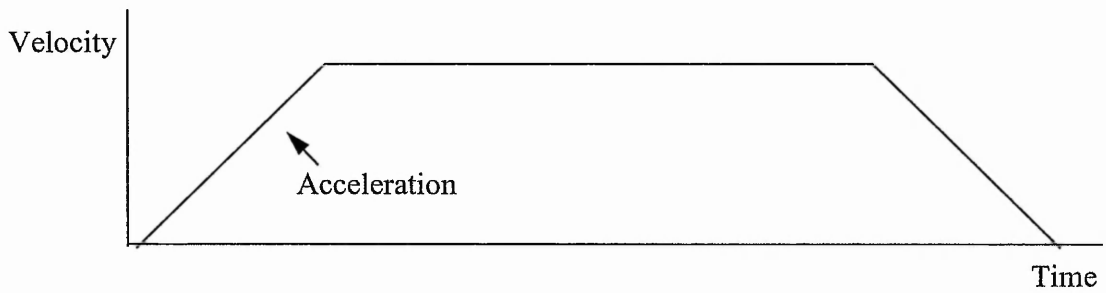


Figure 6 : Stepper Motor Velocity Profile.

Research into stepper motor machine control is centred on the various aspects of motion generation and its instrumentation; of particular interest is the generation of smooth continuous paths. In order to control the motion more effectively then the accurate measurement of this smooth motion is needed. The investigation of such motion requires dynamic measurement of angular position as well as linear motion, Ayandokun [9], Kadhim [10]. Precision measurement at operating speeds requires the use of high speed data acquisition techniques. Data acquisition systems with a high bandwidth and a high resolution can be complex and costly. The Real Time Machine Control group based within The Nottingham Trent University has investigated into the development of a high bandwidth, high resolution digital data acquisition board, which uses a reciprocal timing architecture, McManus [11]. The author fully developed and prototyped this system and utilised the board in support of his research.

1. 4 Continuous Path Motion Requirements.

Continuous path motion can be described as a series of path segments where the movement along each path segment is precisely controlled & known. These path segments when linked together form a continuous path, for example the continuous path could be the outline for an alphanumeric character. In this example if a machine tool followed the path whilst machining then the character will be reproduced in the designated material. However a continuous path does not have to be just the outline of a character, but could be a continuous path to mill out a pocket in a cam, etc.

The task of generating smooth continuous path motion, can be simply taken as moving a machine from point a to point b along a designated path, with constant speed and cutter path accuracy. However to fully understand the problems associated with generating smooth continuous path motion we must study all the stages that a given product goes through from design to manufacture. The four stages of Computer Aided Manufacture are shown in Figure 7.

The first process is often to design the object utilising a Computer Aided Design Package. The second process is to generate Path Information derived from the Computer Aided Design (CAD) drawing. The path information can consist of segments of linear, arc or spline movements that when placed together would result in a continuous path around the shape.

The next process is to take each path segment and compute the number of steps that would be required on each axis to generate the motion. This process often depends greatly upon the interpolator used to interpolate between the two axes. Each of these interpolation algorithms have their advantages and disadvantages in the areas of path or velocity deviation. The last process of CAM is to generate the necessary pulses for each motor and then feed them to the drive system to be amplified to a suitable level. The Elements of a typical Stepper motor control system are shown in Figure 8.

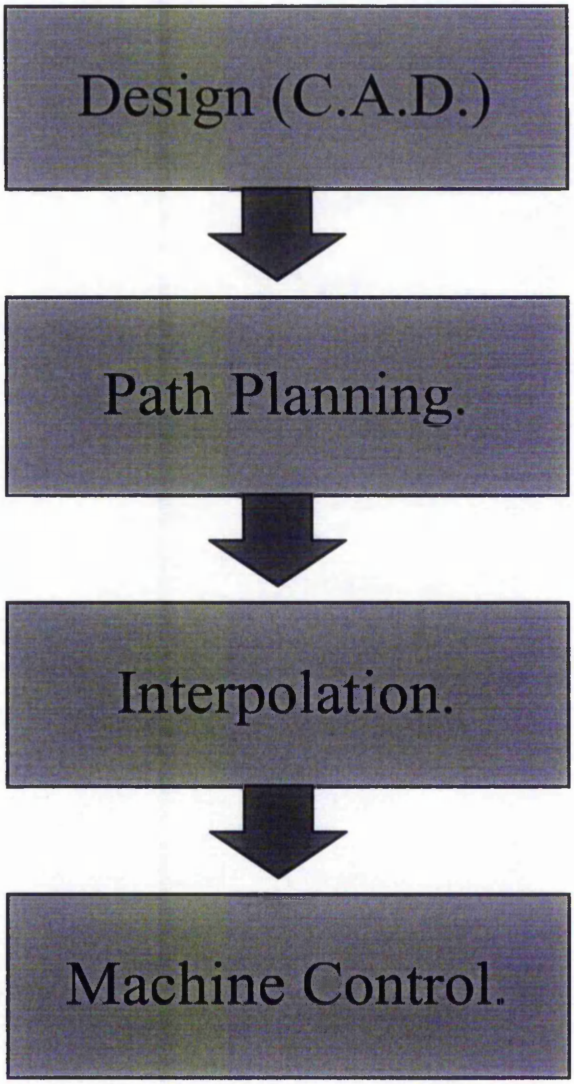


Figure 7 : The Four Processes of Computer Aided Manufacture.

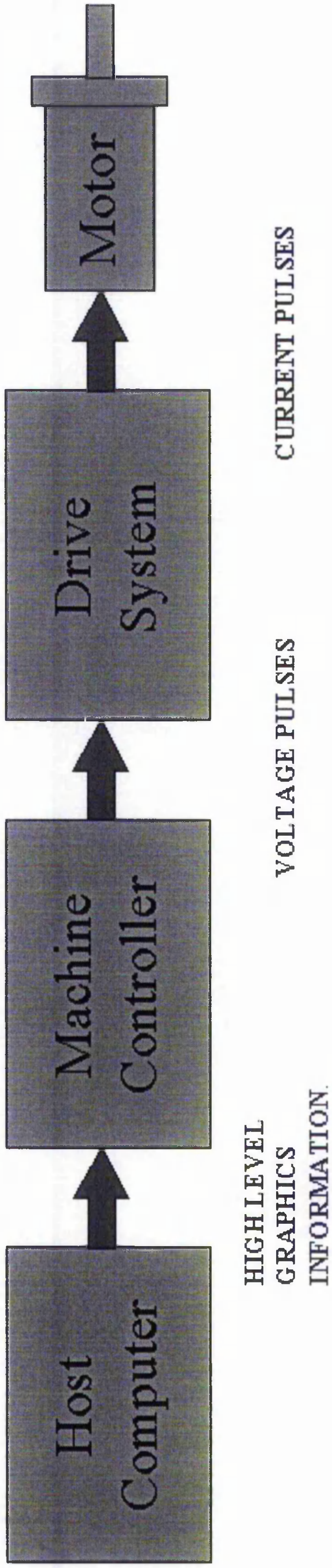


Figure 8 : Elements of a Typical Stepper Motor Control System.

Upon analysis of the four stages of CAM, shown in Figure 7, the last three are the principal causes of machine error. Machine errors are often the result of 'roughness' of motion or other inaccuracies, both of which cause a loss of performance. Errors can be introduced into the system from several points. A list of typical errors follows:-

- Rounding errors in the path planning algorithm, or interpolation algorithm.
- Inadequacies with the Interpolator algorithm, causing misalignment, or speed changes as the tool machines around arcs, or circles.
- Machine vibrations due to resonance effects, chatter, bearing clearances.
- Inadequacies with motion control system or drive system, due to component tolerances or control/drive algorithm.

These errors can be found though experimentation and a process of elimination, i.e utilising path planning and interpolation algorithms, Steiger [2]. Steiger investigated into these two areas, and the author continues this field of research by investigating the motion control area. Therefore the main requirements for smooth continuous path motion can be expressed as the following;

1, Minimal path error between the designed path and the actual path, taking into account, any tolerances.

2, A uniform surface speed whilst the machine tool is machining.

Failure on the part of the continuous path motion control system to meet these requirements in any way results in a significant loss of performance.

1. 5 Summary & Research Objectives.

The research concentrates on displacement errors arising from the excitation of machine dynamics within stepper motor driven CNC machines. Rough motion present in commercial CNC machines stems from the excitation of the machine dynamics as a result of the interaction of acceleration and interpolation. Previous investigations into stepper motor control have been concerned with the generation of smooth continuous path motion, namely the errors introduced by the path-planning and interpolation algorithms, Steiger [2]. Sherkat & Steiger [3] have postulated that problems may arise from the excitation of machine dynamics as a consequence of acceleration interacting with interpolation. Steiger in particular has theoretically shown that the velocity profile on a slave axis may be very rough. The consequence of this is likely to be the justification for the fact that stepper motor driven multi-axis continuous path machines rarely achieve their performance potential. The considerable benefit of cost effectiveness means stepper motors are used in a number of manufacturing areas. Overcoming this performance limitation will benefit a great number of manufacturing industries.

The aim of the research is to investigate & improve the performance of continuous path multi-axis stepper motor driven machinery. This involved:

i) Development of a test platform, with suitable instrumentation.

ii) Experimental identification of the scope for improvement in path smoothness through improved velocity profile shaping.

iii) The development of a new control technique arising from theoretical and experimental analysis of the motion data.

The first step in the project was to construct an experimental test platform incorporating important features of a typical stepper motor driven CNC machine. This platform is essentially a single axis of a multi-axis CNC machine which has been fitted with instrumentation at critical points where the motion is transferred. A custom digital data acquisition board facilitated the capture of the data. This was developed by the author, based upon earlier work within the research group. Several techniques & ideas arising from the literature survey formed the basis of the early planned experiments.

Based upon the experimentation carried out the significant performance limitations of continuous path stepper motor driven machinery have been investigated. New knowledge concerning the operation of stepper motors in multi-axis situations was derived from the experimentation. It has been practically demonstrated that rough motion arising from the excitation of machine dynamics, does lead to a significant limitation in performance. Steps to overcome this rough motion were identified and led to the development of a novel inverse velocity control algorithm. This solution was realised in hardware and reduces the excitation of the machine dynamics to a background level. The new knowledge arriving from the experimentation led to a new method of measuring the true stepper motor performance, taking into account the controller and drive used. This method of deducing the static and dynamic rotor position allows the user to obtain the true torque/speed graph for their system. The research has verified the path following accuracy of the solution through this step error measurement method demonstrating increased performance together with increased accuracy.

Chapter Two

Literature Survey.

Chapter Two Relationship to Other Work.

2.0 Overview

Chapter Two details the literature survey conducted by the author. The literature survey was broken down into three distinct fields. These areas are important to the understanding of continuous path multi-axis stepper motor driven machinery. One field is concerned with motion control techniques both in theory & practice. This area is covered to fully understand the problems associated with the generation of smooth motion for stepper motor driven machinery, and how the systems performance can be measured. The second field concerns interpolation techniques, with special attention paid to the interpolation techniques that take into account any dynamic behaviour of a machine. Since interpolation plays such a large part in the generation of continuous path motion, the choice of an interpolation algorithm greatly affects the cut quality and performance of any stepper motor based machinery. The third field concerns sensory techniques for measuring the dynamics of computer numerical control machinery. This area is important in terms of the experimental part of the work since the choice of measurement technique greatly effects the results of any experiment.

The literature survey describes previous work in these three fields and identifies their key points and their disadvantages. Relevant ideas from these fields were considered in the research of the novel inverse velocity filter.

2.1 Motion Control Techniques.

This report is only concerned with continuous path motion control systems, rather than point to point motion control systems, which are frequently used within robotics. The definition of motion control used throughout this section and the rest of the report is of the type where the motion controller precisely constrains the speed, acceleration, and position of a system throughout a move. Continuous path control, or contouring systems, use synchronised feed drives and therefore can provide accurate positioning at any point in space. An interpolator is needed to calculate the axial increments to traverse along the desired, known path. The control will also need a speed controller, which will govern the feedrates in order to maintain the exact feedrate ratio of f(x):f(y).

Many authors consider a particular aspect of motion control, these include positional control [7,15,16,18,21,22], improved interpolation techniques [20,23,24,32,33,34], velocity control [3,17] and improved drive performance [26,27], to name but a few. A wide selection of these techniques are reviewed in the following sub-sections.

2.1.1 PREVIOUS WORK WITHIN THE GROUP.

The research conducted by the author builds upon the theoretical work conducted by Steiger [2,3], and concerns the improvement of stepper motor driven multi-axis machinery, through improved velocity profile control & design.

Sherkat & Steiger [3] have suggested that problems may arise from the excitation of machine dynamics arising from the interaction of acceleration and interpolation. Steiger [2] in particular has theoretically shown that the velocity profile on a slave axis may be very rough. The consequence of this is likely to be the justification for the fact that stepper motor driven multi-axis continuous path machines rarely achieve their performance potential. The main contributors of rough motion in stepper motor driven machinery, are path planning, interpolation and machine control. The majority of the earlier work was into finding the best path planning algorithm and interpolation algorithm, in terms of smoothness of motion. Steiger came to the conclusion that the algorithm that would yield the best results, in terms of smoothness of motion, would be the Search Step Algorithm, Weck [84]. This algorithm is detailed later within this chapter. The conclusion of his later work was that the majority of the roughness of motion came from the motion control section. The authors work builds upon Steiger's theoretical work by investigating and thereby measuring the real limitation of performance caused by the excitation of machine dynamics within stepper motor driven machinery.

Interpolation of path segments can contribute to the excitation of machine dynamics. There are a variety of interpolation algorithms, each one offering various advantages and disadvantages; these are covered later within this chapter. It is essential to the requirement of continuous path motion that the movement of each axis is constrained in time. The desired path is normally interpolated into chord segments, see Figure 9. A chord segment can be comprised of a single step movement in length or many step movements. The maximum length of the segment is governed by the angle of curvature for the desired path. The interpolation algorithm within the controller governs the minimum segment length. In interpolation terms this length should be as long as possible, in order to finely resolve the angle, see Figures 9 & 10.

Single Step Interpolation Multiple Step Interpolation

Figure 9: Interpolated Path Segments

It can be seen that the matter of interpolation and velocity profile generation are inter-related. Concerning the matter of interpolation the reader can see from Figure 9 & 10, that the number of steps that can be interpolated greatly effects the available interpolation angle. If the interpolator can only interpolate on a single step basis then the machine can only interpolate in 45 degree increments, whilst with a step size of three, the resolution of the interpolation angle has improved to approx. 15 degrees.

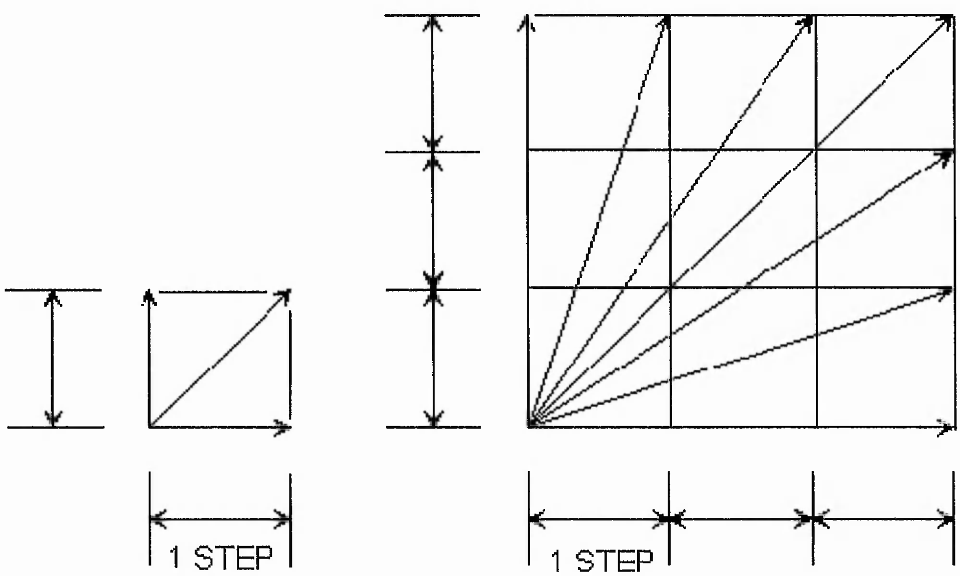


Figure 10: Interpolation step size.

The axes are synchronised at the intersection of these chord segments to maintain continuous path motion. At these intersection points the velocity is changed to maintain the correct feed-rate. These velocity changes are often coarse in nature, due to the discrete nature of the calculations. If the chord segment is only a single step in length then each axis will frequently start and stop, causing very rough motion.

Thus for interpolation the step size is linked to resolution, the larger the step size, the finer the resolved interpolation angle becomes. Unfortunately due to the requirement of synchronising the interpolation step size to the resultant velocity, the velocity profile is adversely effected. Step size and velocity quantisation are proportional, as the step size increases, the coarser the velocity step size becomes. The reader can see from Figure 11 that the resultant input velocity profile is stepped in nature.

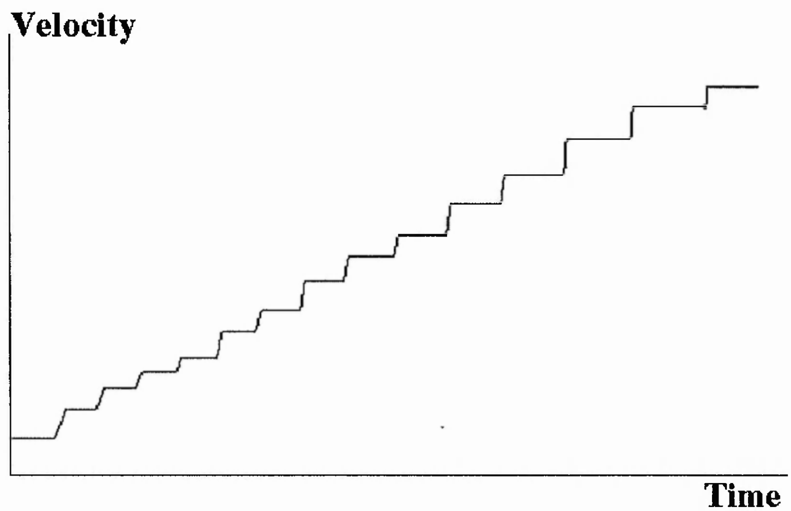


Figure 11: Velocity Profile of Motor Input Pulses.

Ideally the velocity profile would be as smooth as possible with little quantisation, i.e. velocity controlled on a step by step basis. In multi-axis applications the problem arises that if the velocity profile was controlled on a step by step basis then the interpolation angle would be coarse, which in turn leads to machine inaccuracies.

Various authors (Steiger [2], Lho [33], Palmin [50]) have analysed the resultant motion generated by a motion controller and have found that the resultant motion to be quite rough with sharp changes of acceleration. Sometimes these errors are due to the interpolation algorithm, such as the Search Step algorithm which suffers from increased velocity whilst interpolating around a circle. The resultant velocity can be raised by a factor as high as $\sqrt{2}$. This is caused by using the intended end-effector velocity as the scaling factor in the sine / cosine equations. Sometimes the sharp changes of acceleration are caused by exciting the mechanical dynamics of the system, or by inaccuracies in the drive technology. Steiger studied the four stages of CAM, presented in the previous chapter and came to the conclusion that since the performance is based upon a great number of factors, a good measure of the systems performance could be derived from measuring the Jerkiness of the machine. Jerk is the derivative of acceleration against time, in other words the third derivative of distance-time, Doebelin [19].

Velocity $$V(t) = \frac{dx(t)}{dt} \qquad (2\text{-}1)$$

Acceleration $$A(t) = \frac{dV(t)}{dt} = \frac{d^2x(t)}{dt^2} \qquad (2\text{-}2)$$

Jerk $$J(t) = \frac{dA(t)}{dt} = \frac{d^2V(t)}{dt^2} = \frac{d^3x(t)}{dt^3} \qquad (2\text{-}3)$$

Where x(t) is distance.

By calculating the level of system jerk then the engineer could obtain a measure of the machines performance. Later he concluded that the real limitation of system performance was caused by the control/drive system. The author agrees in part with Steiger's hypothesis, but in light of the Palmin's work [50], concerning parabolic velocity profiles finds that measuring the system jerk should not be the only gauge of performance assessment. The reason behind this is partly due to the nature of a parabolic velocity profile, where the acceleration changes constantly during acceleration phase. Ideally if the torque capabilities of the stepper motor could be determined, (taking into account the controller used etc.) rather than the guide supplied by the stepper motor manufacturers, then this could be used as a measure of the performance. The following sections discuss the reasoning and ideas around this issue.

2.1.2 SMOOTH MOTION.

Research into the generation of smooth continuous path motion generally follows two trends, one following the development of better path generation and interpolation algorithms and the other trend following the process of increasing the performance and accuracy of drive systems (Dong-il [16], Schweid [17], Bollinger [18]). The first area has received great interest from researchers and manufacturers alike, resulting in a variety of algorithms. Regardless to the area researched the main objective is the generation of Smooth motion. Smooth motion is taken as being motion that is not erratic in nature, and therefore does not exhibit jerkiness of motion. The author has the opinion that this is not exactly quantifiable as a measure of performance since the motion of the machine might be smooth yet the machine might still be limited in performance.

Often the theoretical ideas of interpolation often do not meet up to the expected performance when utilised in CNC machinery. The reason for this may not be due to the technique but rather the control system used to actuate the machine. The control system can cause the system dynamics to be excited. The author has found that the drive and control system utilised greatly influenced the generated resultant smooth path motion. This explains the phenomena of "rough surface texture" whereby a work-piece can be manufactured within specified tolerances but can still appear to have a rough surface when somebody feels the texture. The reason why the surface appears to be 'rough' is due to the microscopic inaccuracies of the surface. The reason a person might sense this roughness is due to the sensory nature of the human hand, as in the case of computer graphics where an inaccuracy in picture quality is spotted although the fault is too small for the human eye to see. The cause of these inaccuracies is normally taken as being due to vibrations. Exciting the dynamics of the machine typically causes these vibrations. These vibrations result in changes of acceleration and therefore jerky motion. In summary Smooth motion can only be practically achieved if each sub-system of the overall system is matched to the other parts of the overall system, knowledge generally based on experience.

2.1.3 PERFORMANCE CURVE INTERPRETATION.

The performance of a given stepper motor is normally assessed by analysing its speed/torque characteristic in the form of a curve on a velocity/torque graph. To utilise the graph the design engineer determines the total load on the system for a given acceleration and reads off the maximum velocity, taking into account a safe region of about 20%, Sigma motion control [49]. This safe region provides a necessary overhead to compensate for minor torque disturbances and / or slight changes in friction, as such the percentage of overhead is a matter of conjecture and can be as high as 50%. The total load is the combination of frictional and inertial loads. Frictional load predominates in determining the necessary power for constant speed operation. Inertial load defines the amount of additional torque required to start and reach full speed within a specified time (in the process compensating for frictional load), and also to stop (conversely, aided by frictional load). In simple speed control applications, frictional load may be the most important. In position-control applications that require numerous start/stop operations and high system resolution, inertial load becomes the dominant factor, Sigma motion control [49]. Therefore two curves are normally plotted on this graph, a stop/start curve and a slew curve.

Raising the friction from zero to a particular value and then applying a uniform pulse train for one revolution generates Start/stop curves. If the motor follows it successfully, a higher speed uniform pulse train is tried until one is reached which the motor does not follow correctly. The last successful speed is then plotted as a point on the curve. Load inertia is normally equated to rotor inertia for matched start/stop curves, Sigma motion control [49].

Accelerating the motor from zero speed to a specific speed shown on the curve generates slew curves. This is ideally done with no frictional load and with as much acceleration time as required. When at speed, friction is increased from zero to the value at which the motor stalls. These points are plotted to form the slew curve. It represents the outer limit of performance capability, Sigma motion control [49].

For practical purposes, true system performance limits fall between these two curves. Instantaneous torque output approaching that of holding torque (inherent in permanent-magnet & hybrid stepping motors) serves to boost the start/stop curve far closer to the slew curve than would initially be expected. A Torque/speed graph for the experimental platform is shown in Figure 12.

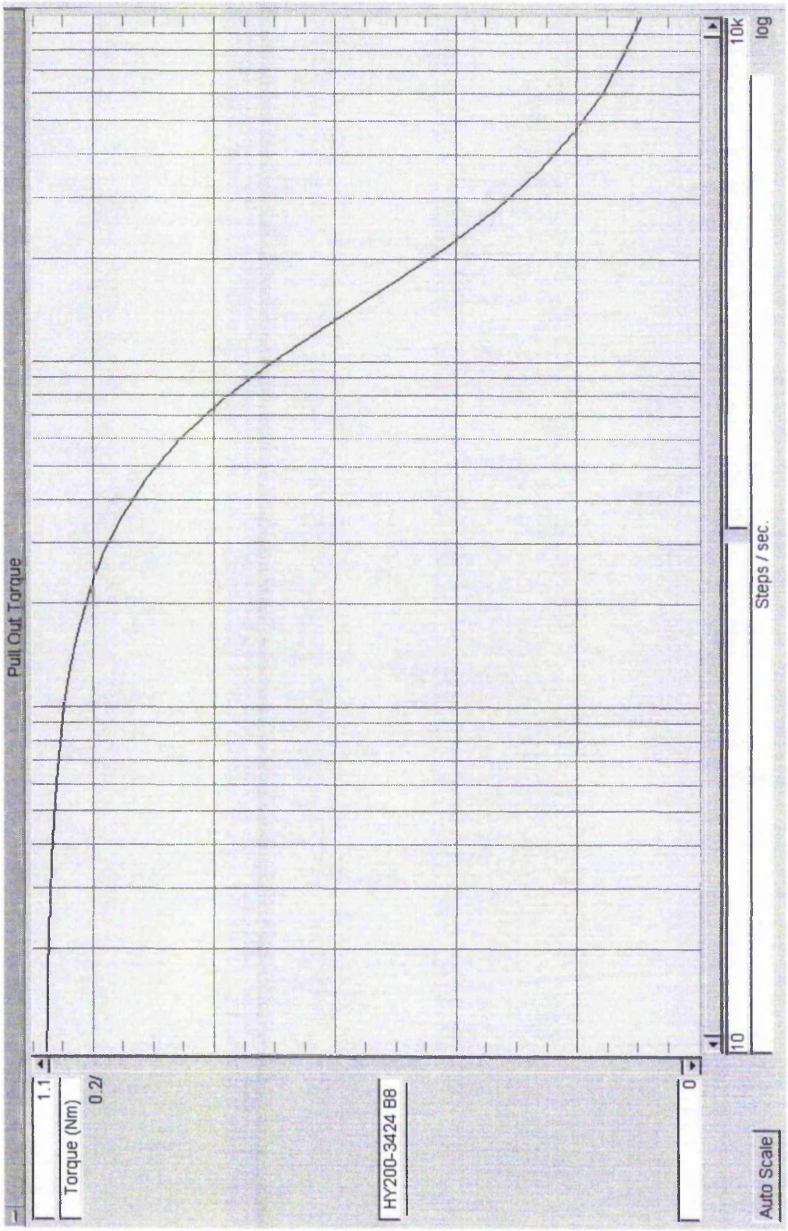


Figure 12 : Torque / Speed Characteristic for stepper motor.

2.1.4 MOTION CONTROL DESIGN CONSIDERATIONS

Given the load parameters and motor torque/speed requirements, the selection of the correct motor/driver combination is usually straightforward for most applications. However the real world and the theoretical world are often two worlds apart, when the full account of interpolation and ramping are considered, yet larger motors may be required. Mechanical stiffness is often used to gauge real performance of stepper motor driven machinery. Good motor stiffness is one of the primary advantages of the permanent magnet and hybrid stepping motors over other motor choices. Stiffness is stated as being a measure of a change in torque vs. a change in angular rotor shaft deflection, Acarnley [27]. The shaft position of a stepping motor is determined by the strength of the magnetic field holding the rotor in place and the external load torque, which deflects the rotor from the commanded position. This deflection can be reduced by decreasing the effective load torque through gear reduction (thereby increasing the steps per work increment), or by selecting a motor with more torque, Parker Hannafin [8].

In stepper motor control systems there is no great need for feedback since each pulse to the motor will move the table a finite distance. They are stable in open loop control. Therefore to decrease the complexity and cost open loop control systems are often used. Some interpolation algorithms suffer from the disadvantage that whilst machining around a circle the start and end points fail to meet. To overcome this problem on-line compensation techniques or feedback would be needed, which again adds both cost and complexity to the overall system.

One of the established ways of improving the accuracy of stepper motor machinery is to ensure that any machine vibrations are damped out by the mechanics of the machine itself. This is achieved by lowering the resonant frequency by increasing machine mass. Most CNC drive trains can be adequately modelled as a low order system (a lead screw indirectly coupled to a stepper motor drive can typically be modelled as a third or fourth order system). These mathematical models can help designers by showing the resonant frequency of machine. Shin et al. [15] proposes a procedure for modal analysis of machine tool structures, this procedure is called the dynamic data system method. He claims this provides better estimation of data contaminated by noise and better results since no averaging is required. Normally the CNC system will be designed to have sufficient mass to damp out a large percentage of machine vibrations that exist. The added mass lowers the natural frequency of the machine, which may overcome the vibration problem. The designed system will also have inherent viscous & friction damping to reduce the vibrations further. In some cases torque dampers or frictional driven flywheels may be fitted to the drive to provide extra damping.

Sazbo [14] describes at length the process needed to limit vibrations when installing an ultra precise lathe laboratory. These include placing the machine on a solid, non-vibrational base that is independent from the building (a monolite concrete mass 1500x 1500 x 700 mm) and surrounding this mass with hard rubber sheeting which was consequently further embedded in pebbles. The pebble base he mentions is a very good damper of vibrations. This is simply the installation, to utilise the laboratory it has to be operated within an air-conditioned room, with the limit for dust content of the working area close to that of a medical surgery. For the general-purpose installation and use of a stepper motor driven machine this seems a trifle extreme, but nevertheless it serves to illustrate what is required to obtain ultra precise components.

The process of adding mass and friction dampers to limit vibration requires the use of stepper motors with higher torque ratios, and/or reducing the velocity/acceleration whilst the machine is contour path following. Unfortunately stepper motors have a non-linear speed torque characteristic, as presented in the previous section, with the motor delivering less torque at high speeds. If the machine is not utilised correctly then the motor can miss steps due to motor torque slippage, or in some cases the motor may stall. In the case of stepper motor driven machinery the cause of these vibrations could be the manner in which the machine is accelerated.

Many stepping motors are available in a choice of winding impedances. Torque output at higher operating speeds increases with lower impedance windings. Improved high-speed performance can also be achieved by decreasing the rise time of the winding currents, Parker Hannafin[8]. This same effect can also be achieved by employing a higher voltage driver. Certain motor/driver/load combinations may exhibit resonant modes within specific portions of the motor speed/torque curve. These often occur at about 50-200 and 1000-3000 steps/second, those within the former region can often be controlled by increasing frictional load, the latter by increasing viscous friction. Motor-torque output may decrease within resonant regions. Operating above low-speed resonance and accelerating through high-speed resonance is often recommended, Sigma motion control [49].

2.1.5 MICRO-STEPPING

Stepper motors have several qualities that make them particularly suited for control systems. They are stiff, exhibiting high holding torque when stopped, produce a reasonable torque for their physical size, and they are brushless, making them virtually maintenance free. Stepper motors do have several drawbacks however, one drawback is experienced when they are used for precise positioning. Standard hybrid stepper motors have a relatively large step size, usually one two hundredth of a revolution, or 1.8 degrees. The motor can be driven in half-step motor where the rotor is balance between two stator poles. Half-step drive operation reduces the 1.8-degree rotor/shaft movement to 0.9 degrees per increment. Half-step drive operation also minimises low frequency resonance and reduces settling time. However this resolution is still quite large when the motor is attached to a large pitch lead screw, and as such might lead to excessive coarse resolution of movement.

Within the field of stepper motor driven machinery one of the established techniques to improve the accuracy and resolution of the machine is to use Micro-Stepping [44]. If the resolution of a stepper motor is improved then the theory suggests that any 'roughness' of motion will also diminish by a similar amount. Micro stepping is achieved by electronic control within the drive circuits. The micro stepping drive sub-divides each full step electronically into a large number of smaller steps. For example a micro-stepping drive that subdivides each full step by 20 would effectively result in a stepper motor with 4000 steps per revolution.

Micro stepping does not only improve the resolution of the system, but also influences other parameters. The natural resonant frequency of most CNC machines in terms of frequency, is comparatively of the lower order of magnitude typically a few hundred Hertz or much lower. The consequence of this is that sometimes a stepper motor might be actuated at the machines natural resonant frequency, causing vibrations and possible tool damage. Luckily most of the time the period when the machine is actuated at this frequency is short or the designer chooses to by-pass the frequency altogether. In field research it was found that the designer using the micro stepping technique avoids most of this troublesome problem since the frequency of stepping action is greatly increased and therefore the motor is mostly actuated outside this resonant frequency range.

Though micro stepping provides increased positional resolution, it is not appropriate for all motion control applications. Firstly the frequency of the pulses also significantly increases, therefore bandwidth problems may be experienced when the user wishes to machine components at high speeds. Secondly the factor of diminished torque, in micro stepping the available torque is greatly diminished due to balancing the rotor between Stator poles. This in itself draws the use of stepper motors into two distinct fields. One field where the system is designed for accuracy utilising micro stepping techniques with the speed limited by the bandwidth of the controller, or the other field where systems are designed for high speed ranges with mid-range accuracy performance.

2.1.6 DYNAMICS OF MACHINING SYSTEMS

Research has been conducted into better path generation algorithms that consider the kinematics and dynamic relations of the machine in regard to the path design [18, 20-24]. The problem arises that these parameters are often addressed at such a high level in the system that they do not take into account errors produced by the processes lower down in the chain. Authors like Bollinger[18] and Debourse [20] approach the problem of smoothness of motion from the interpolation level and try and solve the problem from the kinematics point of view. They introduce different approaches and algorithms, such as spline and cubic polynomials as methods for interpolation. These methods may solve the problem of jerky motion caused by interpolators, but they fail to take into account the machines dynamic behaviour. They naturally assume that the motion controller is ideal and the stepper motor is linear in behaviour, when in fact they are not.

B. S. V. Prasad et al. [25] have briefly mentioned some of the problems found in most D. C. servo CNC systems. In particular they addressed the problem associated with overshoot in position caused by the motor inertia and machine dynamics. Prasad mentions that these problems can be compensated for by decelerating down to a creep feed at the end of each movement. Although this is unacceptable in contour machining since one of the prime directives for smoothness of motion stipulates that the feed rate whilst machining needs to be kept at a constant level.

They go on to mention that these problems can be compensated on a machine per machine basis by conducting machine trials over a period of time to determine the error. This in it self presents problems since the machine control algorithm would have to be continuously updated when the work piece changes. They present the preferred method of compensating for the errors by adjusting the machines next contour path motion. Since they use servomotors requiring feedback, then this information can be fed back to the controller. However this is difficult and is proportional to the frequency rate of the feedback error. They have successfully developed a software-based controller on a personal computer, favouring this approach to traditional hardware based controllers. The author considers this a strong point since software based controllers offer many advantages over hardware based controllers, namely flexibility and versatility. The other main benefit is that they can take into account Artificial Intelligence techniques such as neural networks.

Since some of the earlier papers have been published the technology of the computer industry has increased by three to four fold. Designers now have the option of embedding a whole host of system processors into their control solutions. RISC processors offer a fast streamlined way of implementing complex formulas whilst still maintaining machine performance, indeed some machine control companies have taken this approach to their design. Another approach is to use digital signal processors. Digital signal processors offer fast arithmetic processing combined with fast interrupt servicing. Researchers like Samudio et al [26], have analysed the use of DSP in drive systems, where the DSP can be used to control the conduction angle for each phase of a switched reluctance motor. The work conducted by Samudio is based upon the earlier work conducted by Acarnley[27].

Acarnley describes how the maximum torque can be obtained from a stepper motor if the exact rotor position is fed back to the drive controls and used to control the conduction angle or switching angle, for each phase of the stepper motor. Ideally the position pulses should be generated at the crossover points of the phase torque characteristic, see Figure 13.

If the stepper motor is operated at high speed then each phase is excited for only a short time interval and the build-up time of the phase current is a significant proportion of this excitation interval. Acarnley describes how this can be compensated for by triggering the switching angle slightly early, proportionally to the current speed. Samudio et al. [26] mention that they have taken this into account in their improved drive control and comment that under test it offers significant improvements. This solution is very useful since it provides a method by which the maximum torque from a given stepper motor can be obtained. The work conducted by Acarnley & Samudio is a major contribution to the field of drive technology. However the drawback, in terms of the author's work is that it is in essence a drive system. The DSP will still need to be interfaced to a suitable machine controller that would interpolate machine movements and calculate velocity profiles. Therefore this system would still suffer from any side effects caused by the machine controller. If the interpolation algorithm on board the machine controller caused excitation of the machine dynamics then the result will be a much lower level of machine performance.

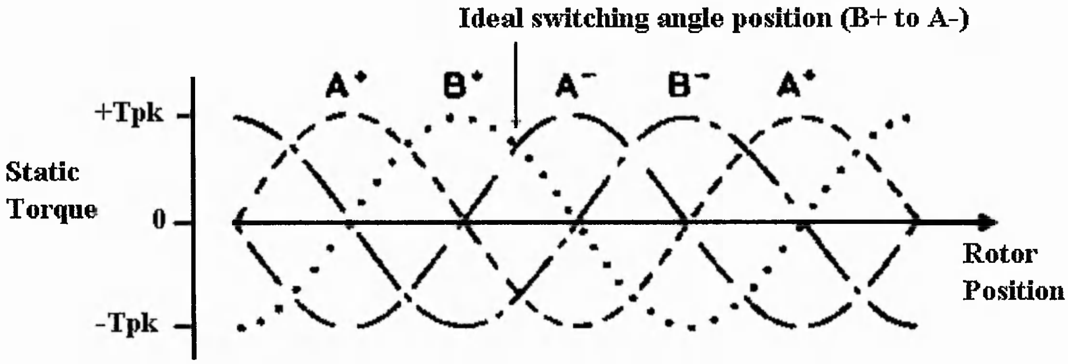


Figure 13 : Example of a Phase Torque Characteristic

One of the main factors concerning the improvement of performance of stepper motor driven machinery is that of torque, namely the utilisation of the maximum available torque that the stepper motor could deliver. Palmin et al. [50] describes how maximum utilisation of available torque is reached if a parabolic velocity profile is used in the motion controller. This is highly desirable since it would allow the CNC machine manufacturer to utilise smaller stepper motors on their production machines whilst still maintaining the same performance levels.

However Palmin et al. present a system where the pulse stream from the controller is controlled in an analogue fashion by a voltage controlled oscillator, compared to a microprocessor based controller that maintains the duration of each pulse precisely. The system in their case is not a continuous path CNC machine where the acceleration is constrained by the interpolator, but rather a single axis slide table.

The work conducted by Acarnley[27], Palmin[50], Samudio[26] has considerably advanced the field of stepper motor motion control. A pattern emerges that the performance of any stepper motor driven system is intrinsically linked to the amount of available torque that the system can utilise. This problem has to some limited success, been tackled at the drive level and at the control level. However these systems fail to meet the requirements for continuous path multi-axis machinery. If a control/drive system was designed to meet these requirements then the performance of such systems would be greatly improved, incorporating all the benefits presented by the aforementioned authors.

2.2 Survey of CNC Interpolation Techniques.

Interpolation is the process of determining an intermediate point, which lies between two known points. These points can be points on a graph or points through which a machine tool has to cut. To quote the correct technical terminology, Lapedes [29] :

*"**Interpolation** :A process used to estimate an intermediate value of one dependant variable which is a function of a second independent variable when values of the dependant variable corresponding to several discrete values of the independent variable are known. "*

Continuous path motion control requires that the geometric information of a design to be translated into axis-orientated co-ordinate orders of motion, in such a way that the addition of the velocity vectors of the axes under control always corresponds to the required contours. The main section of this requirement is the calculation of the positional co-ordinate values, this is carried out by what is known as the Interpolator. The positioning and orientation of a machine tool in space normally offers possibilities of movement with six degrees of freedom. In addition to the three translator axes X,Y and Z, the three rotary axes A,B and C may also be controlled. The calculations for the motion along these axes are carried out by a multi-axis interpolator. As a rule the rotational axis of the machine tool (e. g. the milling cutter) is not under control; hence on machining centres five controlled axes will suffice. On lathes two linear axes need to be controlled and on most milling and drilling machines provision is made for three linear axes, with the possibility of one rotary axis for a rotary table.

The process of interpolation for CNC machinery is carried out within the control system, under a sub section called the interpolator. Various interpolation algorithms are available each with their merits and drawbacks [30]-[34]. The type of interpolation algorithm defines the contour path generated by the interpolator. The most frequent type of interpolation used is Linear interpolation based on the work conducted by Bresenham [34,58,70].

As the name suggests the cutter path is interpolated into a series of straight lines. Another common interpolation algorithm is called Circular interpolation. Circular interpolation is a basic algorithm for interpolating around arcs and circles. If the surface to be machined is a curve, which has a mathematical description with the order higher then two, then the curve is often approximated or split into several lower order curves.

Several other interpolation algorithms have been developed over the years, among the most numerous are; Search- step technique, Digital differential analyser technique, direct calculation of function and Spline. The Digital Differential Analyser (DDA) technique is based upon the mathematical integration of the axes related velocity components, the integration of these velocity components provides the movements in each axis. The Search Step technique is based upon the implicit function representation of a curve within a plane. Each point which is a member of a particular curve satisfies the function equation $F(x,y) = 0$. Each of the aforementioned interpolation techniques was developed to tackle specific problems concerning CNC machining, and often have drawbacks to them. One such drawback is the increased velocity whilst interpolating around a circle. The resultant velocity can be as high as a factor of square root two, this is often the case for the search step algorithm. The direct differential analyser technique does not suffer from this problem, and is often considered by authors such as Bollinger et al [18], and Lho et al [33], to be ideal for open loop continuous path stepper motor control. Unfortunately the DDA technique does suffer from one drawback that search step does not suffer from, that of the start point and the finish point failing to match for a circular path.

One of the key points to distinguish between the different types of interpolators is whether they are incremental step or 'sampled data', 'reference word' interpolators.

Incremental step interpolation algorithms were introduced in the 1960s for displaying and draughting applications on 2D digital drawing devices (graphics screens and plotters). We can distinguish two types of technique for curve interpolation : analytical techniques, which use a totally mathematical approach, and DDA techniques, which use hardware integrators. Analytical methods were introduced by Bresenham[34], whose algorithms for lines and circles are well known and widely applied. Other researchers have also proposed algorithms for line and circle generation to improve speed performance or accuracy performance[59-61]. Jordan et al[60]. and Suenaga et al. [61] introduces the 'closest distance' principle. Analytical curve drawing algorithms have also been extended to ellipses[62-64,69]. The common characteristic of the above-mentioned work is that it uses only the implicit function f(x,y) = 0 as the input of the curve description.

DDA techniques are widely used in NC interpolators for their speed performance, which is due to their hardware implementation. As Danielson [65] explains in his pioneer paper, DDA can be used for both parametric and non-parametric curve generation. However, he also shows, as does Milner [66],some degeneration errors that occur with losses in interpolation accuracy. Improved DDA techniques were proposed by Tao for circles [67], and ellipses [68], while Nakartsuyama et al [72] proposed a DDA method for interpolating 3D implicit functions defined as the intersection of two surfaces f(x,y,z) =0 and g(x,y,z)=0.

In the NC field, hardware implementations of the basic linear and circular interpolators are still mainly used, while complex contours are made up of pieces of these two elementary curves. With the continuing increase in computer power at reasonable cost, there is a clear tendency towards the software implementation of CNC systems. This tendency has been reported from the early 1970's.

Koren [55,74,75] presents a series of three papers that deal with the design concepts of CNC systems. His first paper details the two principal control structures denoted as Reference-Pulse and Sampled Data systems. The first technique is also referred to as incremental step interpolation (detailed above), where the computer produces a sequence of reference pulses for each axis of motion. Each pulse generating a motion of one basic unit (BLU) of axis travel. These pulses can actuate a stepping motor in an open loop control system, or be fed as a reference to a closed-loop control system. In the sampled-data technique the control program compares a reference binary word with the feedback signal to determine the position error. Both of these techniques require distinct interpolation routines to generate their corresponding references: pulses or binary words. The second paper in the series is concerned with the evaluation of circular interpolators for reference pulse CNC systems. Koren presents that since all reference pulse interpolators are based on an iterative technique controlled by an interrupt clock, then the maximum speed is limited by the number of times the microprocessor can interrupt in one second. In the third paper Koren [57] evaluates reference word circular interpolators for CNC systems. He presents that in contrast, that the maximum velocity in sampled systems is not limited by the computer. In these systems the circular interpolation is based upon approximating a circle with straight-line segments. For each segment the interpolator generates a reference word proportional to the local axial velocities which are transmitted to the corresponding software loop comparators of the control axes. In circular interpolation, the simultaneous motion of two axes generates a circular arc at a constant tangential velocity, or feedrate, V_0. The axial velocities satisfy the following equations. :

$$V_x(t) = V_0 \cdot Sin\,\vartheta(t)$$
$$V_y(t) = V_0 \cdot Cos\,\vartheta(t) \qquad (2\text{-}4)$$

Where $\vartheta(t) = \left(\frac{V_0}{R}\right)t$

and R is the radius of the circular arc.

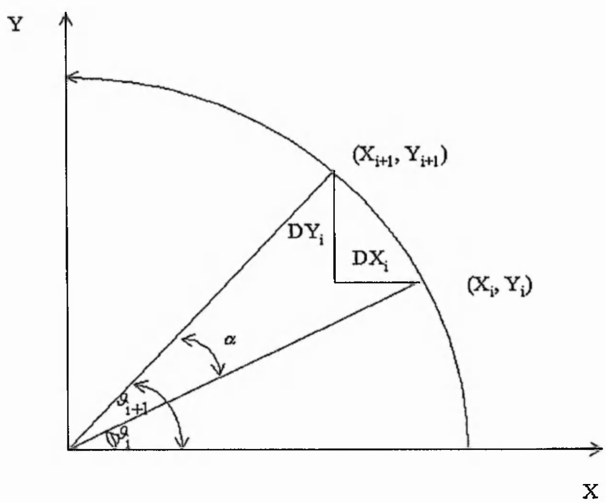


Figure 14 : Definitions for the reference-word Interpolation technique.

The velocity components V_x and V_y are computed by the circular interpolator and are supplied as reference inputs to the closed-loops. The circle generated in this case is actually comprised of straight-line segments. At the beginning of each segment the references are supplied by the interpolator and the end of the segment is located with the aid of a feedback signal. Increasing the number of these segments improves the accuracy of the generated circle, but increases the number of iterations, thus requiring more computation time. Koren [57] has appraised the Euler, Taylor & Tustin methods and found that the improved Tustin method will generate an arc with the fewest number of iterations.

Bedi et al. [51] describe some of the common problems associated with CNC machining. They discuss how a curved surface is approximated by a linear grid of closely spaced points, which are traversed in a linear or circular interpolation mode, in order to machine the desired shape. The sagittal error in the approximation increases as the grid size increases and decreases as the grid size decreases. Thus a small grid size is preferred to machine an accurate replica of the surface. This however results in a large number of points that must be visited by the machine tool. Machining surfaces with small linear motions does have at least one side effect. The frequent starts and stops necessitated by very small linear motion for every command cause a jerky motion to occur during machining. This phenomenon can cause a lot of wear and tear on the machine and must be avoided. Design engineers can often overlook this point when designing interpolation techniques.

CAD/CAM systems are widely used in today's manufacturing industries. Designers use CAD systems to design parts for visual and theoretical analysis. On the basis of the CAD model, the numerical control (NC) programmer uses the CAM system to generate an NC toolpath for a computerised numerical control (CNC) machine so it can produce the part. Besides using a CAD model, some CAM systems can process data acquired from other advanced data acquisition systems, such as laser scanners and computer tomography (CT) scanners, directly. The sets of data acquired from these systems are usually large and over-determined according to Yeung and Walton[76]. Over-determination occurs when the base object can be represented by fewer data points than the number of data points required. Many CAD/CAM systems use linear interpolation in the generation of NC tool paths to approximate non-circular curves or profiles that are represented by a large number of points. Yeung et al. [76] state that this method induces processing problems when the data set is large and over-determined. This method creates many short linear move commands with sudden changes in direction. Tool paths created in this way induce problems for the CNC machine during the machining process, and this can effect the finished quality of the part. To reduce these problems Yeung et al. introduce a method that uses arc segments to generate an NC tool path. They develop a biarc tool-path based on arc-spline theory, which they say reduces the aforementioned problem and therefore improves the finish quality of the machined part. They mention that their method uses continuous circular-arc segments to approximate the smooth curve and with a given tolerance, it can remove the excessive data points, thus reduce the size of the NC program.

Interpolation techniques such as the DDA method can cause the stepper motor to stop and start periodically whilst the controller is interpolating around the shape. The DDA method however still offers several advantages despite this drawback. The answer then is to measure the level of performance.

One of the aims of metalworking production is to increase the economy of machining. In milling surfaces one possibility is the use of a multi-axes milling machine. These machines are able to set the cutter tip and the cutter axis into each optimal position of the cutting range. In terms of cutting conditions it is possible to bring the cutter into the best position, with regard to cost it is possible to have shorter cutter paths generating the same roughness (see Figure 15). In this way a multi-axes milling machine produces an equivalent surface quality in a shorter time. Stute and Damsohn [52] discuss the special problems associated in the post-processing of cutter path data in multi-axis machines. They present that multi-axis milling machines should be used more and more for the production of sculptured surfaces. They have therefore investigated the problems associated with this by analysing a test milling machine with five axes. Special difficulties in making an APT post-processor for this machine are discussed. Deviations of tool tip and tool axis are caused by the simultaneous motion of linear and rotary axes. These deviations can be calculated and compensated in the post-processor. They mention that the methods to do this have been made more effective and more exact, on account of an analysis of the deviations and the other correction methods. The parabolic interpolation, used for this, improves the correction. They conclude that if these transformations are transferred into a CNC controller, then a further essential simplification of the post-processor would be possible.

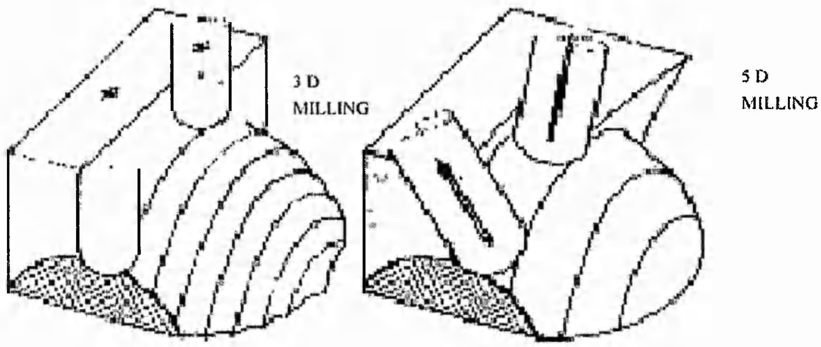


Figure 15 : Three and Five axes Milling.

In a more recent paper Takeuchi and Sato [53] present a new machining concept by making use of 6-axis control. 6-axis control implies that a cutting tool allows three rotational movements for positioning as well as three translational ones. They have developed a 6-axis machining control centre from a conventional 5-axis control structure with a main spindle capable of tool orientation.

A non-rotational cutting tool is mounted at the main spindle to carry out the 6-axis control machining, which they present offers a variety of machining possibilities. In their study, the control software for grooving with an asymmetric cross section is presented together with experimental cutting results. They mention that non-rotational cutting tools can machine a workpiece with the depth of cut of 1-2 microns, which results in fine surface roughness. This Figure is comparable to the roughness in upper end of conventional milling i. e. finishing. See Figure 16. This graph taken from Chryssolouris [77] shows the average surface quality for the different machining processes.

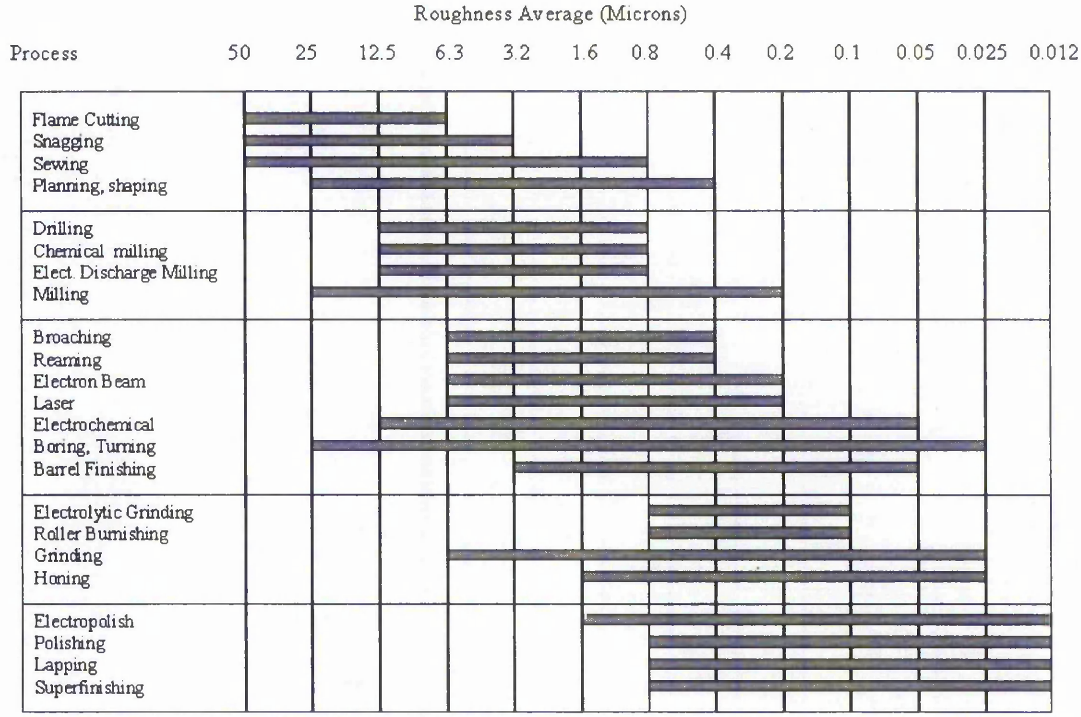


Figure 16 : Surface Quality for different Machining Processes.

Takeuchi et al. [53] go on to mention that low cutting speed and small amount of material removal also decrease the cutting heat generation and strain due to cutting, which leads to low influences on the surface integrity of the workpiece. In summary 6-axis control machining with a non-rotational cutting tool, where the cutting speed is equal to the feed rate, is suitable for the finishing process. The most desirable manufacturing method, taking into account the total machining efficiency, is to make the most of conventional machining (i. e. up to 5 axis control) with rotational tools during rough cutting and rough finishing process. Then, a 6-axis machining control centre with non-rotational cutting tools is employed in the final finishing process to obtain fine surface roughness. They conclude by stating that SPANCS/CAD enables the generation of arbitrary groove on the sculptured surface by a mapping operation; CL data for 6-axis control can be generated and that they have experimentally confirmed the validity of a 6-axis control grooving system.

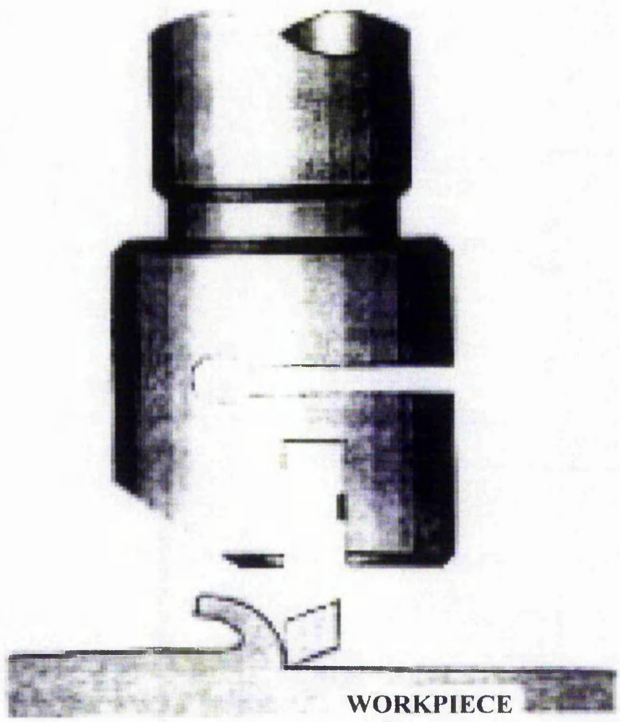


Figure 17 : Non-rotational cutting tool mounted at the main spindle.

The author studied the various interpolation methods in great detail discovering their various merits and drawbacks. The author concluded with Steiger that the choice of interpolation algorithm should be a critical factor within stepper motor control design since it influences a large number of parameters, not only the obvious ones concerning path error. From literature consulted the author found that both the direct differential analyser and the search step technique offered possibilities. One aspect arising from the study of the available interpolators is that of post-processing. If the improvements in machine performance can be achieved as a post process then these improvements can be gained irrespective of the interpolation algorithm.

2.3 Survey of Control System Sensory Techniques.

In the world of engineering there are numerous methods of measuring displacement, velocity and acceleration. The author found that control engineers would often use a variety of methods and sensors to obtain these factors. The author found in some cases people would use accelerometers to measure acceleration, encoders to measure velocity and other sensors to measure distance. This often clouds the problem because now we have to understand, attach and calibrate three sensors instead of one. The increase in sensors also increases the cost of data acquisition and feedback control. The author considered the range of sensors around suitable for the task and selected to use encoders to measure all three parameters.

An encoder is a transducer that generates a coded reading of a given measurement. Shaft encoders are digital transducers that are used for measuring angular displacements and angular velocities. Some of the advantages of encoders include high resolution, high accuracy, high noise immunity and relative ease of adaptation to digital control systems, Avolio [35]. The shaft encoder consists of a disc with slots or patterns machined into it, through which light can be shone. The light is shone through these patterns and is detected with a sensor on the other side. Thus as the pattern changes the sensor can detect the presence or absence of the light beam, see Figure 18. Encoders are classified into two categories : incremental encoders and absolute encoders, see Figure 19.

The incremental encoder has an outer ring with many pulses machined into its surface. When the encoder is rotated a series of pulses are generated. Counting these pulses with respect to time will give a measure of angular displacement and angular velocity. The encoder disc will have a reference pulse from which displacement can be measured. The absolute encoder has a disc that has many pulse tracks machined into its surface. These pulse tracks form a digital pattern consisting of several pulse trains when the encoder disc is rotated. These pulse trains will give out a reading that corresponds to the absolute angular position the disc is currently residing at. Absolute encoders are often used to measure fractions of a revolution, Avolio [35].

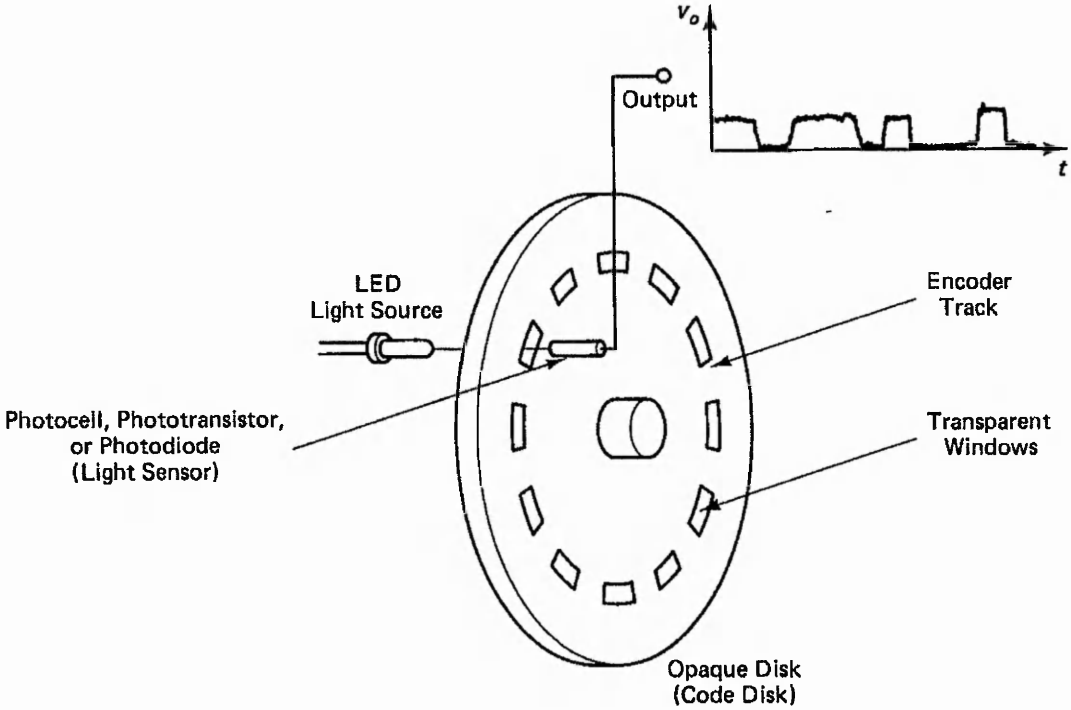


Figure 18: Rotational Encoder

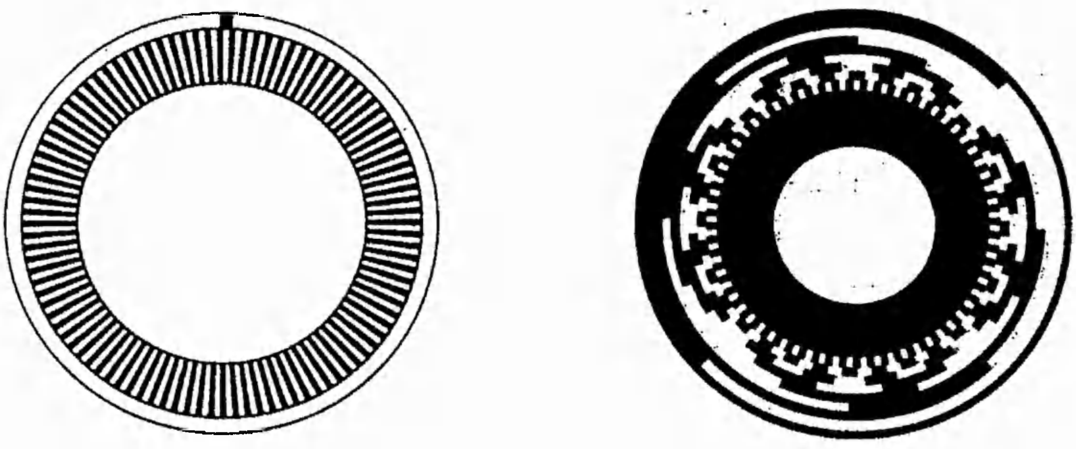


Figure 19 : Incremental & Absolute Rotational Encoders

There are several methods of detection that an encoder might use. The previous example used the common notion of shining light through a grating and there by detecting the absence or presence of light. Some other common methods of detection can be seen in the following Figures 20-22. These include the sliding contact encoder, magnetic pickup encoder and the proximity probe encoder.

The sliding contact encoder is constructed of an electrically insulating material. Circular tracks on the disk are formed by implanting a pattern of conducting areas. The advantages of sliding contact encoders are low cost and high sensitivity. The disadvantages are quite numerous include friction, wear and brush bounce due to vibration.

The magnetic pickup encoder is constructed with high strength magnetic areas imprinted on the encoder disc. The small pickup element is typically a micro transformer, this pick up device will have a high frequency primary voltage that will induce a voltage into the secondary coil. The level of this induced secondary voltage changes with respect to the presence and absence of the magnetic areas.

The proximity probe encoder uses a proximity sensor as the signal pick-up element. Any type of proximity sensor can be used, typically sensors include magnetic induction probes or eddy current probes. The output signal of the encoder changes with respect to the proximity of the disc. The disc could be constructed in the fashion of a tooth gear, combining functionality and sensor, if space is at a premium.

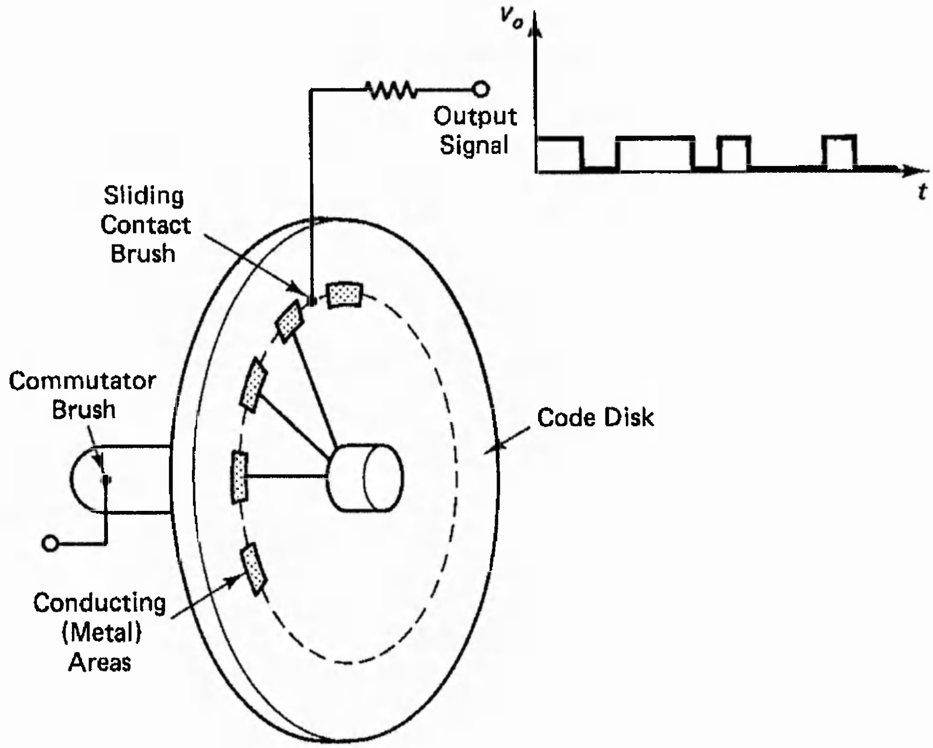


Figure 20 : Sliding Contact Rotational Encoder

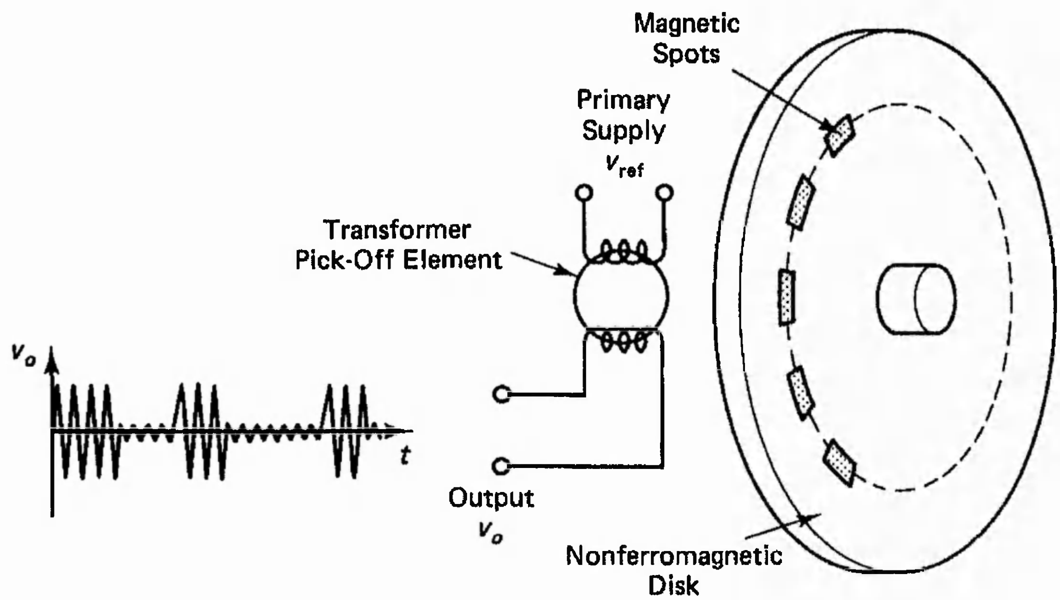


Figure 21 : Magnetic Pickup Rotational Encoder

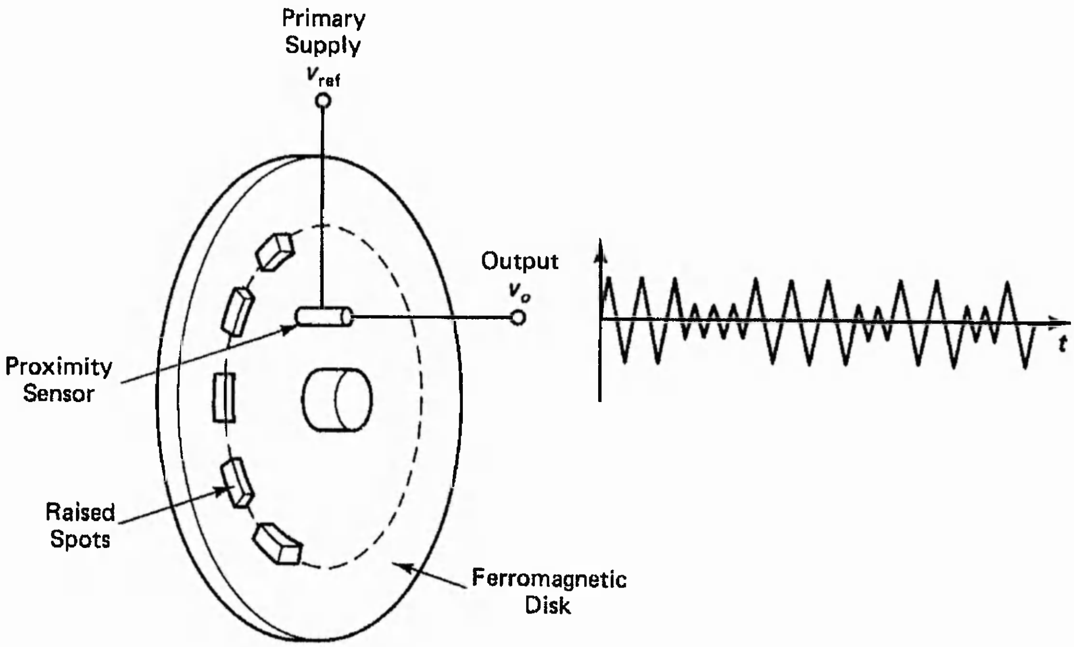


Figure 22. Proximity Probe Rotational Encoder.

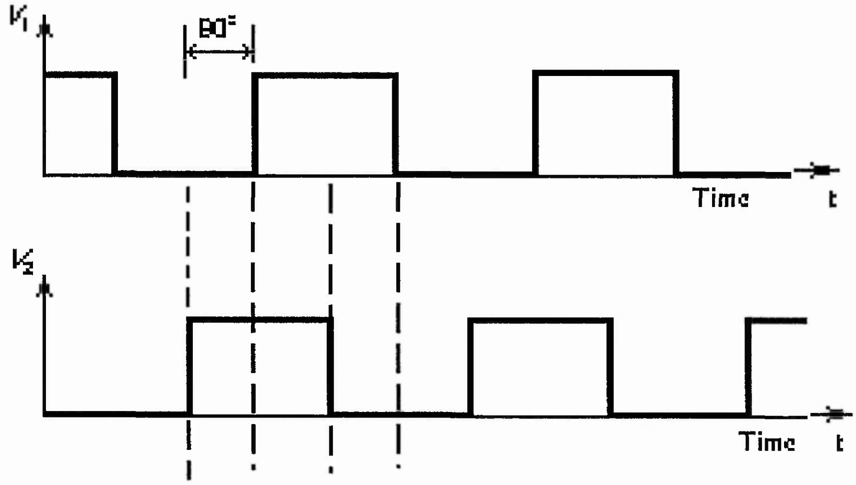


Figure 23 : Encoder Quadrature outputs.

The encoders mentioned previously typically give out at least two signals, these signals typically being two quadrature output signals, channel A and B. These signals that are electrically shifted by 90 degrees to provide a higher degree of accuracy, see Figure 23. The reason for the quadrature signals is to enable the user to determine the direction of motion. If channel A leads channel B the encoder will typically be rotating clockwise, and vice versa for anti-clockwise. Sometimes a third channel is added to the encoder to form a once per revolution pulse. In some instances the output signal can be sinusoidal in nature, reflecting the intensity of the light rather than the mere presence or absence of it.

For many years researchers have investigated the properties of rotary encoders , mainly trying to develop higher resolution encoders, and therefore gaining a higher degree of accuracy[36]. Several researchers have conducted research on how rotary encoders can be used to measure acceleration, torque and in some cases cutting force.

Hagiwara et al[37] describes a method of improving the resolution and accuracy of incremental rotary encoders using a phase encoding and code compensation system. They use encoders with a scale count of 3600 and claim to increase the effective resolution to 920,000. The encoders they utilise provide quadrature sinusoidal scaling signals. They feed these two sine waves into analogue to digital converters and use the output signal to reference a ROM. On analysis of the idea the author found that the idea held prospects for a precision machining system, except for the fact that many encoders simply give out digital pulse information not analogue. Several companies do produce encoders that give a sinusoidal waveform output signals such as Heidenhain Ltd. The Linear encoder supplied by Heidenhain for the experimental platform has a static resolution of one micron but gives out two quadrature sinusoidal waveforms upon which they interpolate, this gives an effective resolution of 0. 2 microns. The above method could be applied to the linear encoder to increase the precision, should that be necessary.

Ramirez et al[38] have studied a method of estimating the speed and torque of rotating machinery. They examine the characteristics of speed estimation using an idealised encoder. They present a speed and torque estimation using an inexpensive magnetic probe. They express that this technique is well suited to milling and drilling equipment where the rotor experiences small transient speed changes during machining and speed fluctuations are harmonically related to the average shaft speed.

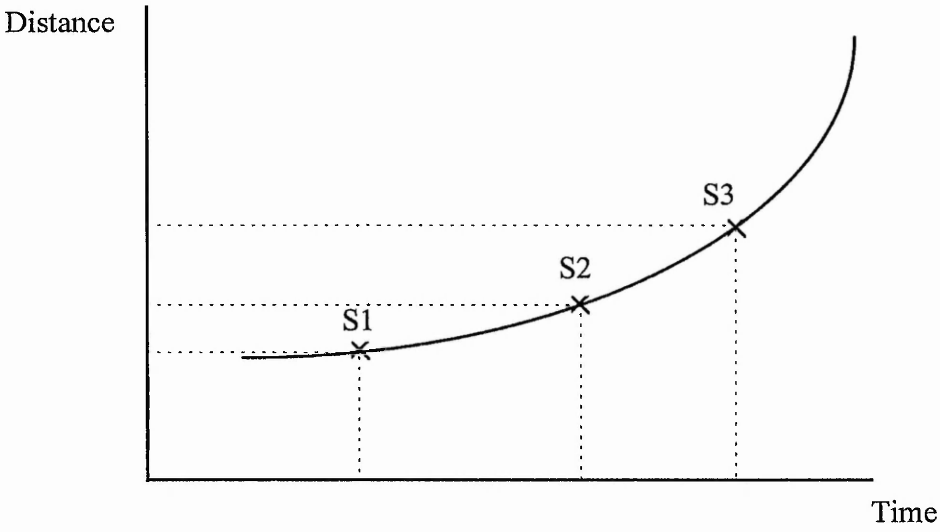
Christiansen et al[39] describe a digital method of measuring angular velocity for use in control applications. The main concern of the researchers is to extract velocity information whilst the movement of the rotor is very slow. They discuss various methods of measuring the encoder pulses and discuss the drawbacks of using encoders. The first is the large number of slots required to obtain frequent measurements at low speed over fixed time intervals and the second is jitter generated by imperfections in the slotted disk. They use optical incremental encoders for their experiments. Optical incremental rotary encoders have machining imperfections to some degree. The result of this is that when the optical beam is shone though the disc the generated digital pulse width can vary on a pulse by pulse basis. In the digital world this variation of the signal is termed Jitter. Another similar problem not mentioned but considered by the author is that of unequalled mark to space ratio of the disc gratings. The average angular distance of the mark pulses does not equal the corresponding average angular distance of the space pulses. The corresponding signal output then is 'banded'. The easiest method to alleviate this problem is to average the two distances together such that the mean is shown. This assumes an even mark to space ratio which basic encoder interpolation is based upon.

If we consider the other end of the scale, where the rotating encoder rotates at high speed we find other problems. Bélanger[40] conducted research concerning measuring angular velocity and acceleration by using an encoder. He mentions that when measuring velocity by finite difference of angular measurements at regular time samples (pulse counting) then the estimates become significantly degraded as the sampling rates increase. He suggests using Kalman filtering of the angle measurements, based on some model of signal generation. The research carried out by Bélanger seems to demonstrate an important point with regard to the selection of the sampling theorem. In Bélanger's case the measurement of the signal becomes more quantised as the sampling frequency increases, resulting in inaccuracies. If Bélanger switched over to the pulse period counting method (inverse time) at this interval, then the inverse is true. In the pulse period counting method the signal becomes less quantised as the sampling frequency is increased, provided the encoder disc has sufficient gratings. In certain instances pulse period measurements, where the length in time of each pulse is logged, although more difficult to implement presents the most advantageous solution to recording velocity and acceleration information from optical rotary encoders.

Godler et al[41] have developed a novel rotary acceleration sensor based on rotary incremental encoders, using two encoder discs. One of the encoder discs has elastic elements to transform the angular acceleration into radial displacement. They use an optical read head to detect the displacement without direct contact. They mention that a low-pass filter is needed to obtain a true waveform of the acceleration because the detected signal includes a high frequency component, which is caused by the Eigen frequency vibration of the disc. The novel system was developed out of the need to acquire acceleration information from a moving joint that does not move sufficiently to produce the amount of error signals in time for the controller.

Godler [41] claims that the use of differentiation methods is not accurate enough and as such led to the development of the novel sensor. Although the design promises good results, Godler fails to mention why the use of differentiation methods are not accurate enough for his work. He also fails to mention about the original resolution of the encoder used before inventing his new acceleration sensor, leaving little evidence to fully comment upon the technique suggested.

Forsythe [45&46] and Thomas et al. [47] conducted research into developing suitable digital algorithms for prediction, differentiation and integration. They provide a full proof and give an example application were their algorithms could be used to obtain accurate velocity and acceleration information from low resolution encoders, in this case from encoders attached to locomotive trains, see Figure 24. Thomas[47] suggests that when dealing with low speeds it would be feasible to base a measurement system upon time between pulses rather than the pulse rate counting method. This method would have a time base, which would be latched and stored when a change of state occurred.



Where : -

$$V(t) = k_1 S_1 + k_2 S_2 + k_3 S_3 \quad \text{(Velocity)} \tag{2-5}$$

$$A(t) = k_1' S_1 + k_2' S_2 + k_3' S_3 \quad \text{(Acceleration)}$$

Figure 24 : Calculation of Velocity and Acceleration Information.

Brown et al [54]. have conducted investigations into the algorithms used for constructing velocity approximations from discrete position based systems verses discrete time based systems. In their paper they review some existing velocity estimator algorithms and their mathematical theory. They evaluate the algorithms in terms of dynamic range and transient response, from this they propose a new set of algorithms for velocity estimation based on at a least squares approach.

The algorithms investigated in their study fall into two general classes of velocity estimators, defined by how the time and position information are acquired. When position and time information are obtained using an optical encoder the user can make a choice regarding whether time information or position will be required. With the advent of the microprocessor, many motion control systems are being implemented as discrete time systems. The performance of motion control systems can often be enhanced by including some type of velocity feedback, where for example, the velocity is estimated from discrete position versus time information provided by an optical encoder.

The author considers all the aforementioned methods of sensory techniques and concludes that most of the problems seem to arise due to the incorrect choice of the sampling method, sampling frequency or encoder resolution. Initially back in the sixties & seventies data storage was very expensive. This lead the engineer to select the pulse counting method as their preferred choice for collecting velocity and displacement data, since this method uses the minimum amount of storage space. However this method suffers heavily from quantisation errors as the sampling frequency is increased. In order to obtain accurate data the engineer was required to sample the signal at an ever-increasing rate. This progressed into finding techniques that could be applied to the data such that the resultant information was not too degraded [40], [47]. In the nineties storage space is relatively cheap and it is more feasible to built a high precision clock than a high-resolution encoder. This development has lead to a re-emergence of the pulse period counting technique, where the time is logged when the relevant circuitry receives a digital pulse. This technique has the advantage that as the sampling clock is increased then the level of quantisation diminishes. The choice of these parameters is very much application specific but what is evident from the literature is that if they are not chosen wisely then problems are sure to arise. The author concluded that the use of the pulse period counting (inverse time) sampling method, coupled with high-resolution encoders (2000+ gratings / rev) would yield the best results in both quantity and quality.

2. 4 Summary

Sherkat & Steiger[3] have demonstrated that problems may arise from the excitation of machine dynamics arising from the interaction of acceleration and interpolation. Steiger [2] in particular has shown theoretically that the velocity profile on a slave axis may be very rough. The consequence of this is likely to be the justification for the fact that stepper motor driven multi-axis continuous path machines rarely achieve their performance potential. The main contributors of rough motion in stepper motor driven machinery, are path planning, interpolation and machine control. Once the problems associated with path planning and interpolation have been tackled, then the major contributor to rough motion is the machine controller.

The work conducted by Acarnley[27], Palmin[50], Samudio[26] has considerably advanced the field of stepper motor motion control. A pattern emerges that the performance of any stepper motor driven system is intrinsically linked to the amount of available torque that the system can utilise. This problem has to some limited success, been tackled at the drive and control level. However these systems fail to meet the requirements for continuous path multi-axis machinery because they are based on single axis motion. If a control/drive system was designed to meet these requirements then the performance of such systems would be greatly improved, incorporating all the benefits presented by the aforementioned authors.

In Summary, from literature reviewed the author finds that a post-processor solution involving the pulse period counting technique, will overcome the significant limitations in performance of multi axis stepper motor driven machinery. This solution should monitor and re-shape the stepper motor pulses such that the maximum utilisation of the available torque can be achieved. The most effective way to achieve this is to firstly understand in greater detail how this limitation is exactly caused. This will be achieved in the form of experimentation.

Chapter Three

Understanding Continuous Path Motion through Experimentation.

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3.0 Experimental Design

It has been hypothesised that rough motion present in stepper motor systems generally arises from the excitation of the machine dynamics through interpolation. The interaction of acceleration with interpolation severely compounds the problem. To find a solution to this performance limitation, we need to first understand why the performance is limited when stepper motors are used in multi-axis systems. This knowledge will be gleaned from experimentation. The tests were designed in such a way as to study how various parameters of the system affected the motion of the machine. There already exist a few methods on how to minimise the effects of the rough motion, but there is no real evidence to validate the effectiveness of these methods for smoothing the motion of continuous path systems. The experiments conducted encompassed these design rules, and provide evidence of the validity of such rules.

The first section details the experimental rig and production CNC machine used in the experimentation, providing information on the mechanical construction of the apparatus and the accuracy of the instrumentation. The first section also details the digital data acquisition board, used to capture the data arising from the instrumentation.

From the second section of this chapter the reader can find out about the experimentation in greater detail and view some of the results in the form of velocity-time graphs. This section also details the experiments conducted on the production CNC machine provided by Pacer systems Ltd. These experiments were conducted to verify the results obtained from the experimental test platform. These results form a knowledge base on the significant causes of multi-axis performance limitations.

3.1 Test Systems

3.1.1 Detailed Description of Experimental Test Rig.

The research is applied to continuous path multi-axis systems in general, but more specifically to the range of machines produced by Pacer Systems Ltd, part sponsors of the work. The experimental test rig built solely by the author was modelled on the general drive system used in such machines; namely indirect coupling, via pulleys, of the stepper motor to the ballscrew. The experimental rig was designed to meet specific requirements, namely;

1, The system should represent principle features of a multi-axis stepper motor driven machine.

2, Should not attempt to minimise transmission errors.

3, Sensors to be incorporated at every point where drive power is transferred.

4, A suitable sensor to measure linear displacement of the slide table.

5, Size requirements 800 x 600 mm.

The requirements were met, which in turn resulted in a suitably designed experimental platform, a plan view of which can be seen in Figure 25. The experimental platform was not only designed to investigate rough motion, but rather to measure the extent of these inaccuracies, in order to prevent them via suitable velocity profile control. This is the reason why the transmission errors have not been minimised, if everything is nearly perfect then there would be no errors to monitor. This does not mean that the errors present are higher than normal but they simply represent the tolerance of machined parts produced by a skilled machine operator. Once the requirements for the experimental platform were specified, the author produced technical drawings for component manufacture. The components were fixed to the experimental platform and the instrumentation installed.

The first main requirement that was incorporated into the design, was to make the platform modular in design with respect to drive method such that the slide table can be driven directly or indirectly. This factor is extremely important since the platform was going be used to model a variety of CNC machining systems. For greater stiffness the motor would typically be directly coupled to the leadscrew of the slide table. Quite often this is not possible due to a variety of reasons and the leadscrew has to be indirectly coupled to the motor. This method of coupling leads to the system becoming less mechanically 'stiff', which in turn leads to inaccuracies due to backlash and the 'spring' or 'flexible' characteristic of the coupling.

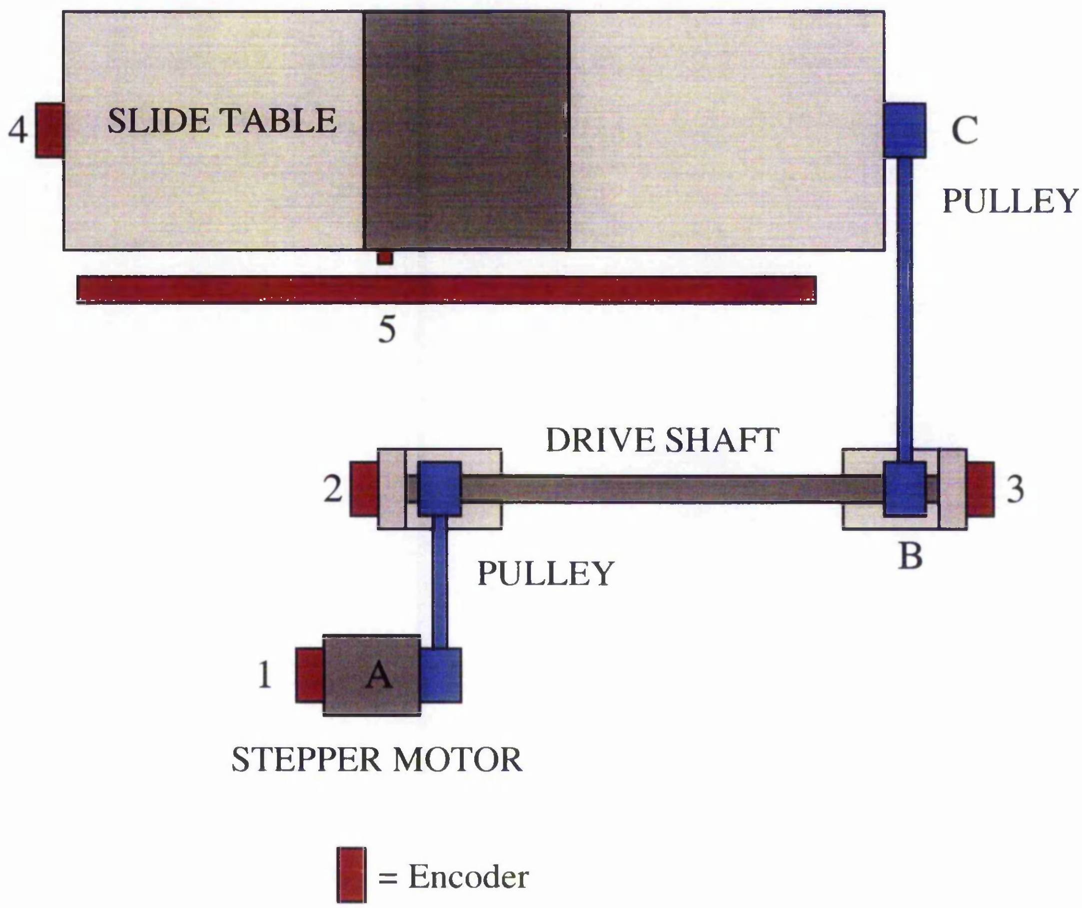


Figure 25 : Plan view of Experimental Platform.

The experimental platform was initially chosen to be indirectly coupled to the stepper motor, since the indirectly coupled method represents the worst case scenario in terms of controlling the motion of the machine. The reader can see from Figure 25 that the slide table can be driven directly if the Stepper motor is fixed into place at point C, or indirectly driven if the stepper motor is fixed into place at points A or B.

The experimental platform has many sensors incorporated into its design, rotational encoders are fixed to points one to four, and a linear encoder is attached along the side of the slide table. A photograph of the encoders can be seen in Figures 26-27.

The encoder attached to the end of the slide table (No. 4) has a resolution of 500 pulses per revolution, giving a static resolution of 4 microns. In Figure 27 the other encoders can be seen in more detail. Viewing from left to right the viewer can see the linear encoder (No. 5), one of the two shaft encoders attached to the drive shaft, (No. 2 & 3), each consisting of an encoder disc with four read heads. Finally the reader can see on the far right an encoder attached to the shaft of the stepper motor (No. 1). The encoders used on the drive shaft have a resolution of 1024 pulses per revolution, giving a static resolution of approximately 2 microns when translated to a linear movement. The rotational encoder attached to the end of the stepper motor shaft has a resolution of 5000 pulses per revolution, giving a static resolution of 0. 4 microns when translated to a linear movement. The linear encoder has a static resolution of 1 micron, or 0. 2 microns when times 5 interpolation is used. Heidenhain Ltd donated the linear encoder (No. 5) and the shaft encoder (No. 1). The encoders used on the drive shaft (No 2 & 3) were intended to be used in another area of research covered by a colleague, so the encoders were purchased with their requirements in mind, namely 4 read heads. Their maximum number of grating lines for their respective size was the deciding factor in selecting the kit encoders (No 2-4).

The linear encoder is used to observe dynamic behaviour of the 'end effector' due the excitation of the machine dynamics by drive system vibrations, and to record the level of displacement caused by 'backlash'. The measured level of 'backlash' was in the order of 30 microns. For the experimentation an initialisation movement was first made in a particular direction to remove the effects of 'backlash'. All subsequent movements were made in this direction, thus eliminating the effects of 'backlash'.

The encoders fixed to the experimental platform allow the user to monitor machine vibrations and reactions completely, from the stepper motor right the way up though the drive train to the end effector. The encoders when interfaced to a suitable data acquisition system allow the user to measure the movement of the slide table to sub-micron levels.

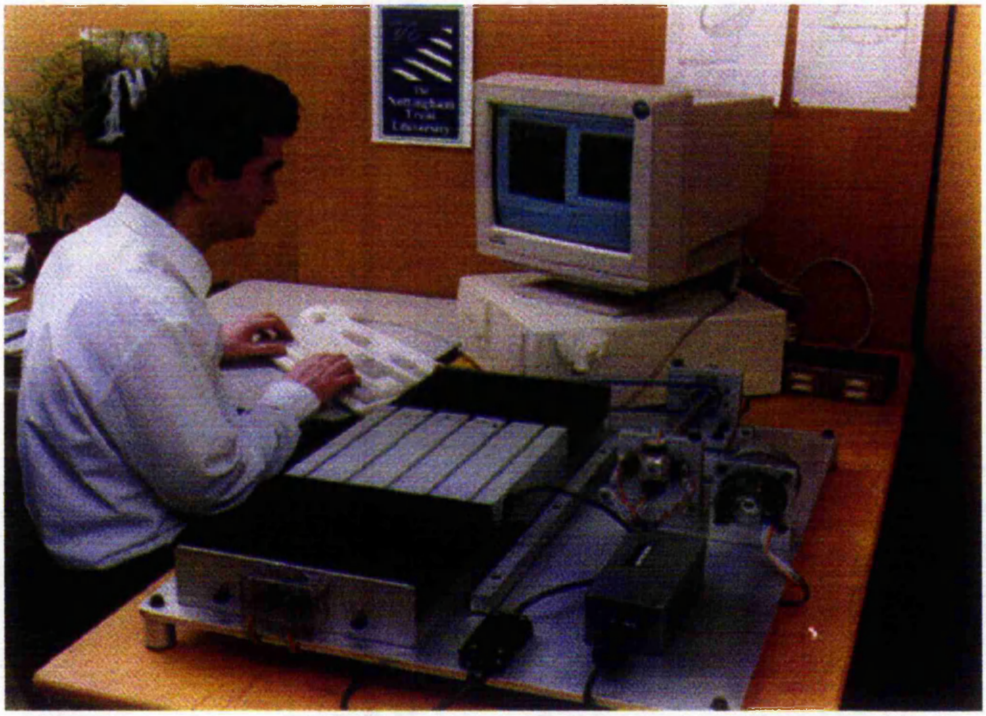


Figure 26 : Photograph of Experimental Platform.

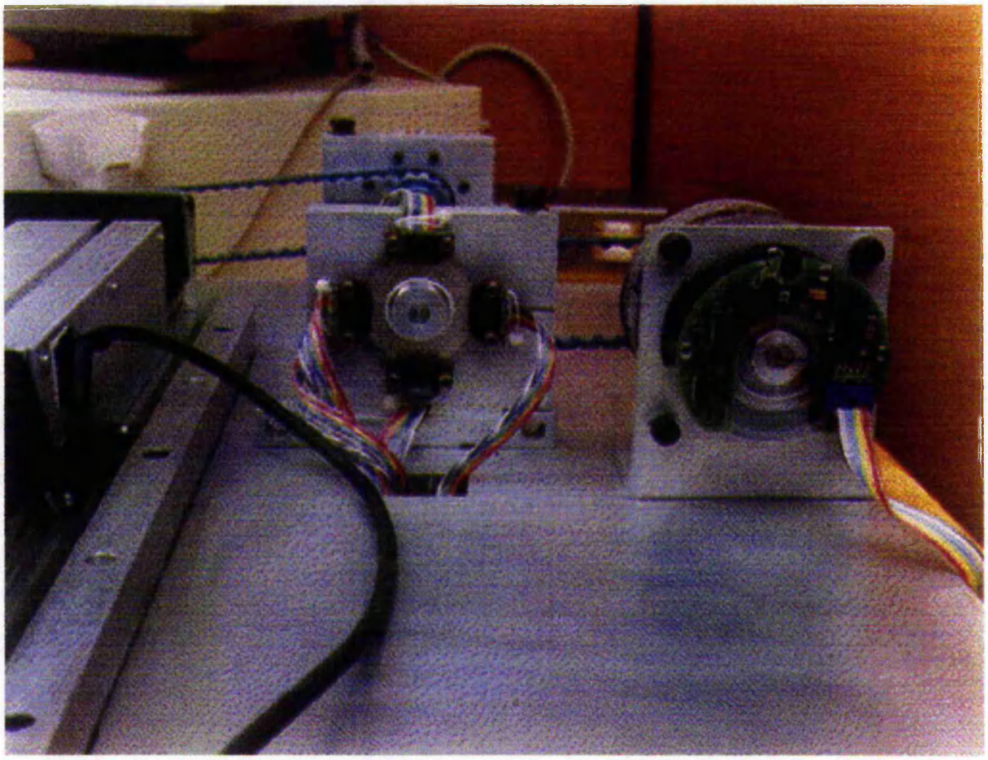


Figure 27 : Photograph of Incremental Encoders.

3.1.2 Description of Data Acquisition System.

The concept of the digital data acquisition board was initially theorised by McManus [11], who needed to acquire data from CNC machining system at a high rate. McManus's research was concerned with the field of Fourier analysis of machine vibrations over the complete working area of a CNC machine. He required a data acquisition system that would capture high frequency vibrations with a minimal amount of quantisation. To this end he prototyped an inexpensive novel data acquisition system using a method of time stamping McManus [11].

One of the fundamental aims of the author's research was to investigate the performance limitations in stepper motor driven machinery. To this aim the author built and instrumented the experimental test rig. Initially it was perceived that the author would utilise the data acquisition board prototyped by McManus. However the board suffered from several fundamental problems and the author had to redesign the board as part of his research. Without a reliable, noise-free data acquisition board the experimentation and therefore the research could not be accomplished. The outcome of this 6 months of work was a 4-layer, reliable, high-speed data acquisition board that fitted within an ISA slot on the PC.

When recording pulse information there are two methods of data capture, pulse counting or pulse period counting, see Figure 28. The author implemented the second method such that when the input pulses arrive the time is logged. This effectively means the data can be recorded as fast as the signal propagates to the board, up to the upper limit set by the clock speed and on board FIFO. Since the clock frequency is much greater than the pulse rate, and taking into account that the encoder resolution is high, then this method of data capture produces the best results. In addition the amount of quantisation error is reduced to such a low level, that it can viewed as negligible (the reciprocal of the clock frequency), provided the clock frequency is sufficiently high. This revised capture system was used to capture accurate information concerning the velocity profile of the slide table whilst moving. The block diagram of the revised system can be seen in Figure 29. The circuit diagrams and logic analyser traces for the second revision of the data acquisition board can be seen in detail in Appendix A.

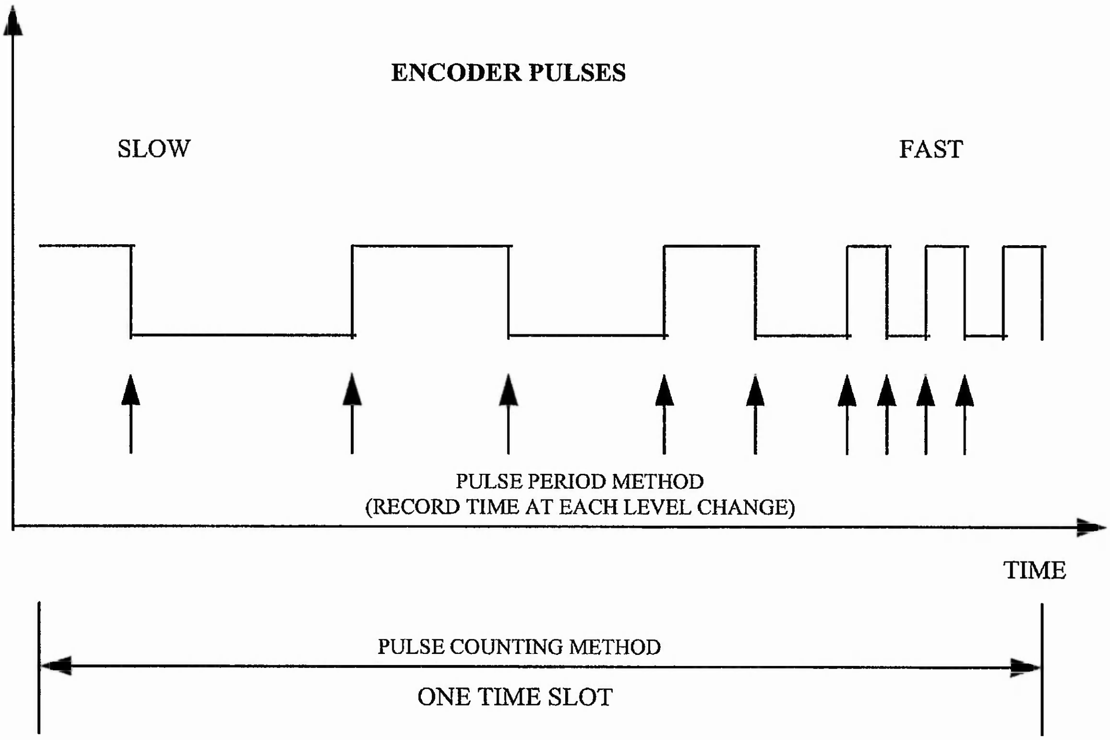


Figure 28 : Digital Encoder Signals.

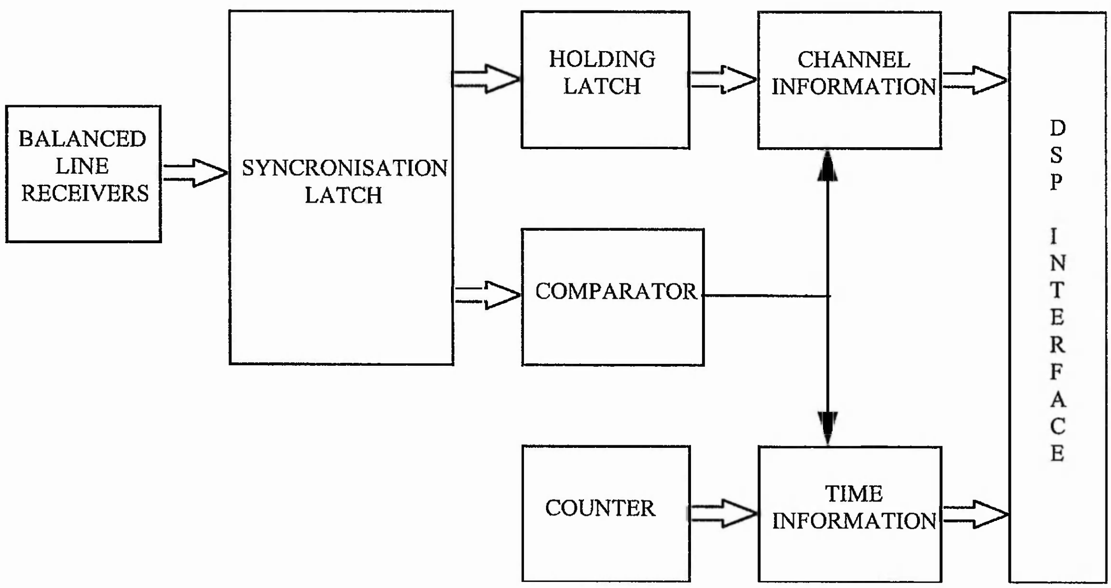


Figure 29 : Functional Block Diagram of Data Acquisition Board.

3.1.3 METHOD OF OPERATION.

There are eight balanced RS 422 channels that form the input to the board. A description follows of the operation of the board but only considers one input channel being used.

A pair of complementary digital signals, typically from an encoder, is fed into the detection board. These signals are passed through a differential line receiver that minimises skew and random noise. The skew is minimised by passing the signal through a schmitt-trigger. The noise is minimised through the use of twisted pair cable & the common mode rejection capability of the differential amplifier. The output from the line receiver is then fed into a latch where it is synchronised to the on-board clock. The latch is also used to eliminate Meta-stable conditions. The signals are then fed from the latch into a comparator and another latch. The second latch is used for speed purposes and to hold the channel information before it is loaded into the channel FIFO. The second latch is clocked on the negative edge of the clock signal to prevent data being accidentally overwritten when new data arrives on the next positive clock edge.

The comparator contains a built-in latch that is used to contain the previous channel value. The comparison is made between the current and previous channel values. If there is a change of state on any of the input lines then EQU (equal) line falls low, and when the clock edge next falls low the write circuitry initiates a write cycle to the FIFO recording the channel the change occurred upon. One set of FIFO circuits records the input signal so that the user can tell which signal changed. The other pair of FIFO circuits have their input lines connected to a sixteen bit counter. When a write signal is issued the current count is loaded into the FIFO, this is used to record the exact number of clock pulses that have past when the change occurred.

The counters are cascaded together in a carry-look-ahead fashion to form a sixteen bit counter. The carry from the most significant byte is used as a signal to indicate that the counter has 'rolled over'. This signal is latched and recorded to allow the user to know precisely (to the nearest 25 nanoseconds) the absolute time from initialisation.

Thus the digital input signals are time stamped when they initiate a change of state, either from a logic one to zero, or vice versa. This time can then be used to calculate the time period between the changes, hence the frequency of the digital signal.

When the FIFO's become Half full they signal an interrupt to the Host system to empty the FIFO's. The FIFO's can be written to or read from asynchronously. The system can either communicate to the PC ISA bus via an interface card or via a high-speed link to a host Parallel D. S. P. system.

3.1.4 Description of the Pacer Cadet Machine.

Pacer Systems Ltd. are a manufacturer of commercial production CNC machines. They utilise stepper motors in an open loop manner as the main actuators for their machines. They offer a large range of machines from the CADET with an active work area of 1 square metre to the K2 with an active work area of over 6 square metres. The author instrumented a CADET machine since this is the most internationally sold of Pacer's machines. A photograph of the machine is shown in Figure 30.

Pacer's CADET machine[1] is a most versatile CNC router / engraver. Designed with advanced engineering technology to operate in a 'windows' environment as simply as a plotter. The CADET's standard features include vacuum hold-down of material, automatic removal of swarf, variable speed spindle for routing and floating head for constant depth engraving. The transmission system on the CADET machine is typically a stepper motor attached via a pulley to a leadscrew. A pulley ratio of 1:1 is typically used. The experimental test rig was designed in such a way to reflect this.

The x axis of the CADET was chosen to be the axis to be instrumented, the stepper motor for the x axis is the most loaded of the two stepper motors used in this system. A 2000 pulse incremental shaft encoder was attached to the rear of the stepper motor. The linear encoder supplied by Heidenhain Ltd. was attached to the bed of the machine such that the movement of the end effector could be measured. The pulley belt for the y axis was then detached for safety reasons.

[1] Literature & photograph supplied by Pacer Systems Ltd.

The results obtained from the CADET machine can be seen in section 3.3, again a small sample of the results is shown the rest of the results forming a database, allowing a correlation between the CADET and the test rig to be formed. The correlation was that the rough motion present on the test rig was of the same order of magnitude as the rough motion present on the CADET, and other such machines when they were subsequently tested. The main difference between the results was the presence in the test rigs case of the velocity fluctuations due to the misaligned pulley. This was subsequently changed.

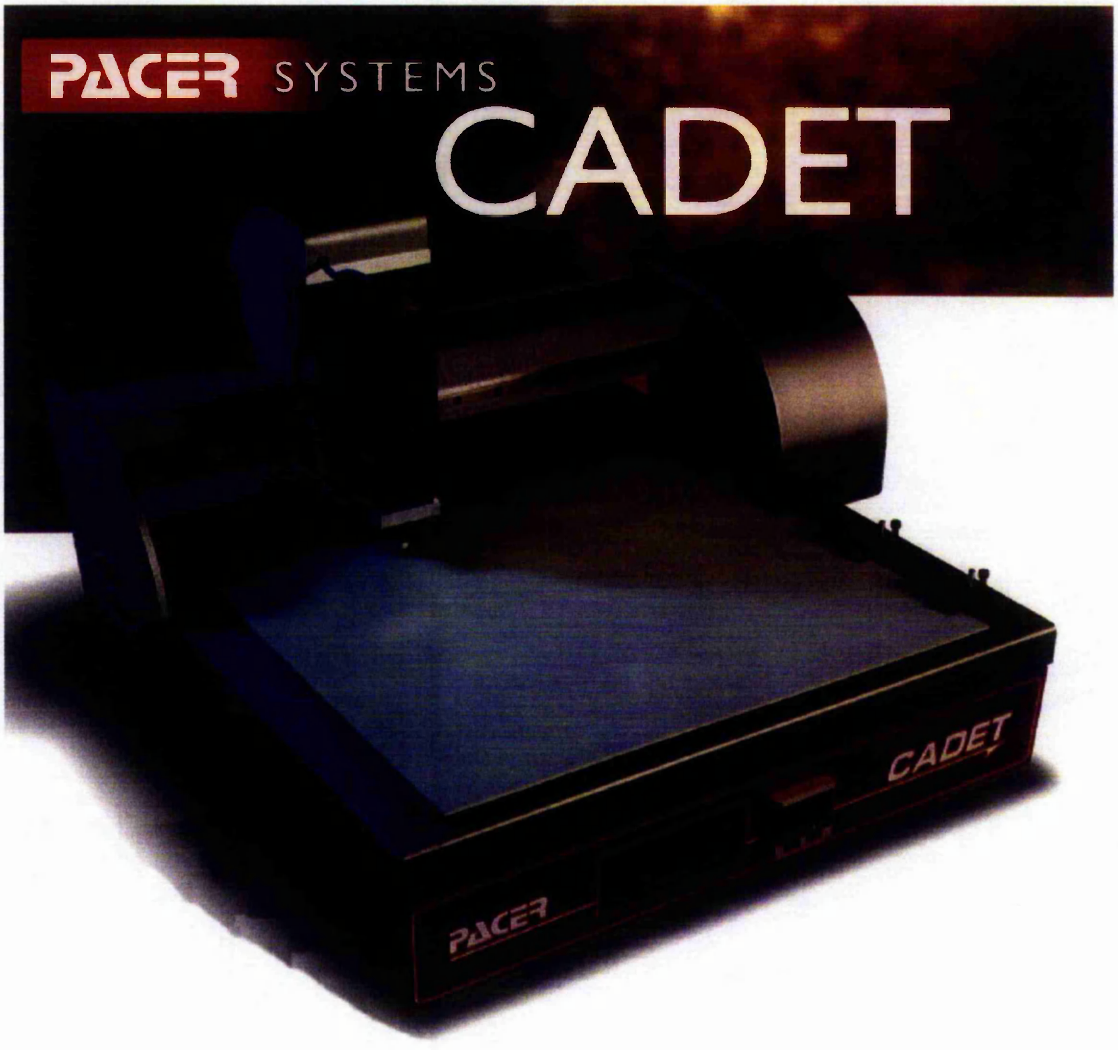


Figure 30 : Photograph of Pacer Cadet Machine.

3. 2 Experimental Results.

3.2.1 Discussion on Velocity Profile Results.

Axiomatic Technology Ltd provided the Stepper motor controller used to obtain the various velocity profiles. They have been developing stepper motor controllers and other products based on stepper motors for a number of years. The company manufactures stepper motor controllers for producing signs, lettering and logos etc. The main controller is called the Versatile Motion Controller or VMC for short.

The VMC board accepts the extended HPGL command set, from which the cutter paths are interpolated by linear or circular interpolation. The controller provides continuous path control, so that the two main axes are closely synchronised in both position and velocity. For a given path segment the dominant axis for the given move is chosen to be the master axis. The required velocity profile is calculated and placed into memory to be used as a look up table. Since the required move is mostly dictated by the velocity/acceleration of the master axis the slave axis is constrained in order to maintain synchronisation. The velocity profiles for the axes are split up into interpolation segments, such that velocity and distance are synchronised. The length of these segments depends upon the given acceleration ratio and velocity for the given move. A velocity profile segment taken from the start of a move is shown in Figure 31. The x-axis is the master in this case, and is shown as being the higher of the two velocity profiles. Since the velocity on the slave axis is dominated by the master axis it is often coarse in nature with frequent changes, this can be seen in Figures 31 and 32. This is the nature of continuous path systems as discussed within first two chapters, where one axis is constrained in velocity and distance by the primary moving axis. The author tested another continuous path machine controller produced by an alternative manufacturer. The velocity demand output from this controller was similar in nature to the output from the VMC board.

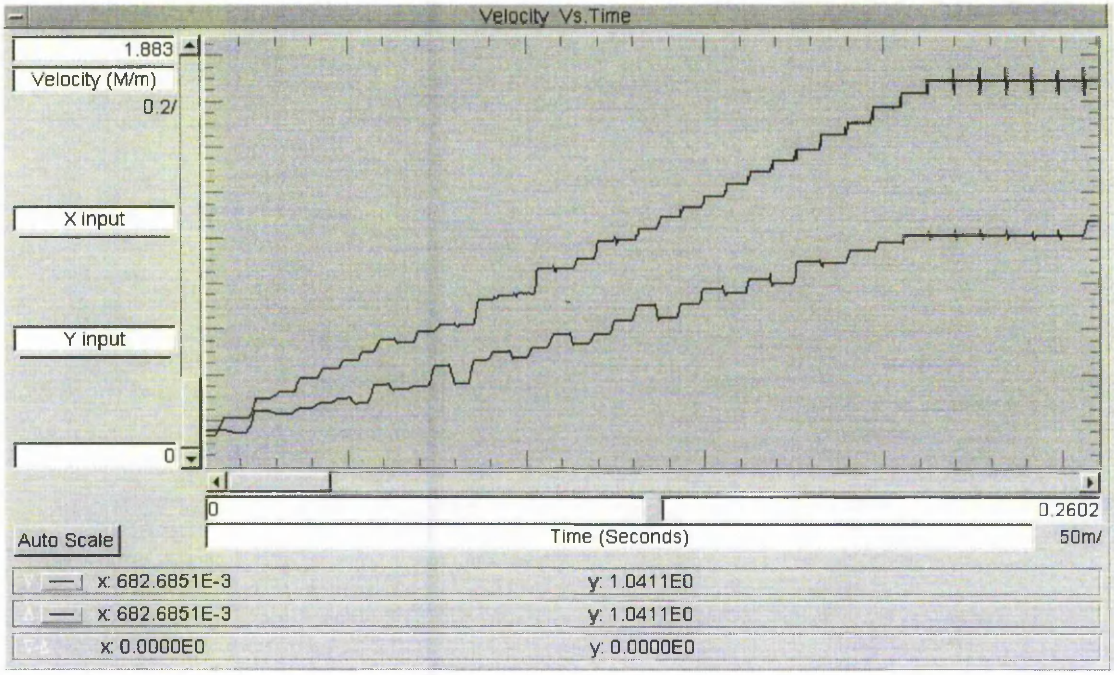


Figure 31 : Velocity Profiles for X & Y Axes.

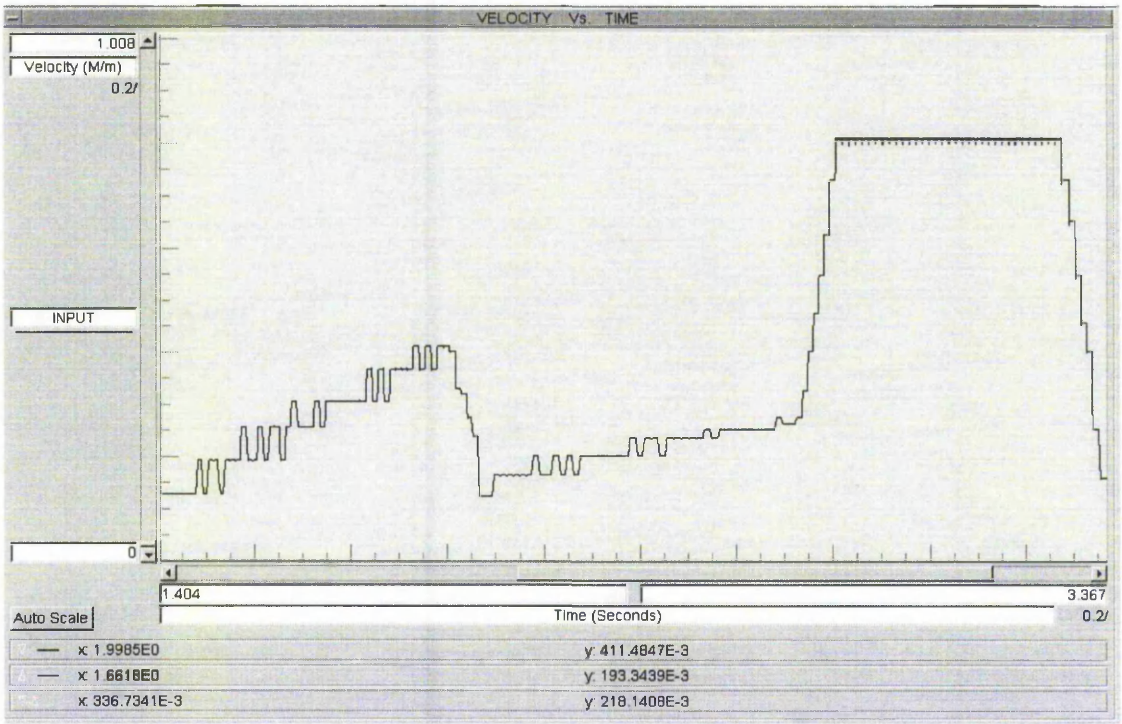


Figure 32 : Velocity Profile for Slave Axis.

Steiger [2]. postulated that theoretically these frequent changes in velocity have the potential to cause large position errors and are the cause of jerky motion in stepper motor driven machinery, which in turn can lead to the machine stalling. Stalling is defined as being when the load torque exceeds the available torque of the stepper motor, the motor is pulled out of synchronism with the magnetic field and the motor stalls. The hypothesis is that the jerky motion reported by Steiger et al. causes the performance limitation in multi-axis stepper motor driven machinery. This jerky motion is the product of the machine dynamics being excited by the interaction of acceleration and interpolation. The excitation of the machine dynamics will cause displacement errors whilst path following. The author calls these dynamic displacement errors since there is no static loss of position. Plotting the derivative of a quantity, in this case plotting velocity rather than distance provides an effective way of measure the error or change of the quantity with respect to time. To further investigate the matter, experiments need to be conducted to measure the magnitude of these excitations, the results taking the form of velocity versus time graphs.

Measuring the velocity error solely due to the interaction of acceleration and interpolation is compounded by the fact that many other factors can contribute to the excitation of the dynamics. Upon analysis of velocity/path data the author concurred with Steiger's theoretical hypothesis that the interaction of acceleration and interpolation, resulting in rough motion on the secondary axis is the predominant cause for the dynamics to be excited. However for a given shape the master and slave axis may be interchangeable. This means that although the surface speed of the machine remains consistent, each part of a shape is unique in terms of the velocity on each axis. To maintain consistency between the experiments the author measured the velocity fluctuations present during the acceleration phase of the beginning velocity profile. The acceleration phase is consistent since the velocity steps consist of pre-determined values stored in a table. This way the consistency between experiments is maintained since the data acquisition board only logs information when a state change occurs, in other words only when the machine moves. The author decided to change the parameters on the controller such that the normal velocity profile ramp became highly ramped in nature, with changes in velocity similar to those experienced on the slave axis. A sample velocity profile for the slave axis can be seen in Figure 33.

When a velocity profile is needed the VMC board starts off typically at a velocity step of about 0.13 M/min. The stepper motor does not need to simply start from zero because of the pull-in rate of the stepper motor, so the velocity profile starts at a frequency suitable to start in terms of torque, but high enough that needless time is not wasted. Beginning at this start point the VMC board proceeds to step up in frequency (velocity). This continues until the next step would exceed the target frequency, when this happens the controller will use the desired velocity as the measure for the next step.

If the feedrate is set to below 0.13 M/min then the controller will not ramp up to the desired velocity, it will just start at the desired velocity, i.e. if the feedrate was set to 0.05 M/min then the resultant velocity profile will be viewed as a rectangle. The lowest speed acceptable for the VMC board would be 0.01M/min corresponding to 13.3 Hz.

It can be seen from the results that the machine is sufficiently underdamped, this is typical for a system that has been designed to traverse at high speed [30]. In order to travel at high speed the system must be mechanically 'stiff', but friction must be minimised. Viscous friction is the main damping factor in designing a CNC machine. The resulting acceleration/deceleration of the system is quite erratic clearly indicating that the dynamics of the machine have been excited.

The following pages and Figures discuss the various aspects of controlling stepper motors. These include deducing the required torque and the resonant frequencies of stepper motors in hopes of finding the cause and effect of the velocity fluctuations shown in Figure 33.

The reader can see from Figure 33 that the dynamic response of the system fluctuates dramatically, first accelerating fast then quickly decelerating at the same rate, whilst the motor is trying to accelerate. If the motor is trying to accelerate the required inertial mass whilst the machine is decelerating then the required torque is a great deal larger then the required torque to move just the slide table, hence the motor may stall. If a measure is taken of the true acceleration being experienced by the stepper motor, it can be seen that the acceleration is in the order of 22% g (acceleration due to gravity). This is eleven times higher than the rate of acceleration for the demand signal.

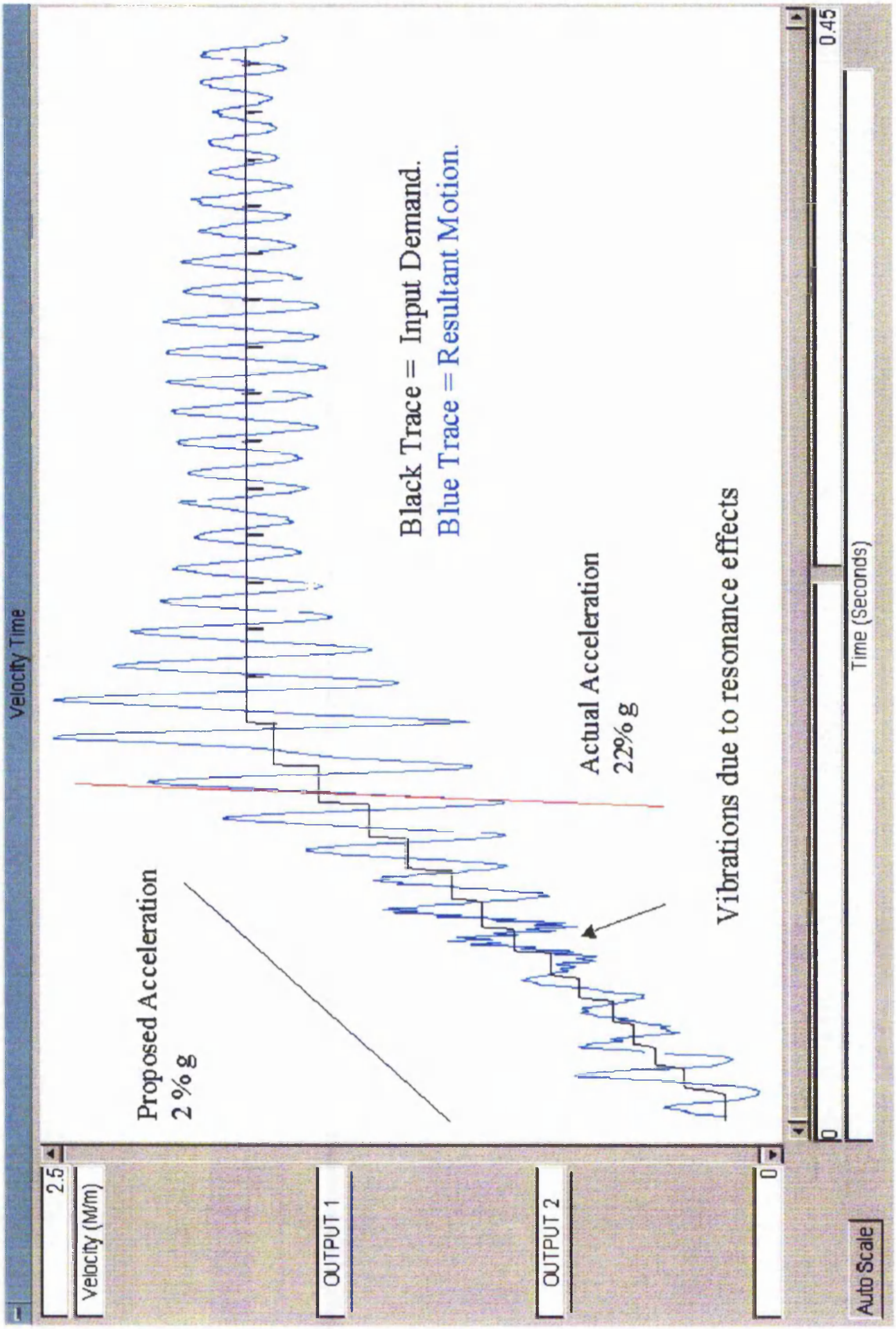


Figure 33 : Standard Velocity Profile for Experimental Platform.

3.2.2 Experimental Test Plan.

The design of the experiments conducted were as follows:

- Resonance Effects : Operation of the system whilst contouring, with the speed set to a figure around the resonant period of the stepper motor. Use of a signal generator & driver to control the stepper motors speed in order to find other resonant periods. Figures for resonant periods taken from Acarnley [27].

- Current Adjustment : Verification of design rule. The rule links the stepper motor current to path errors/ torque levels. Reduce the stepper motor current to reduce path errors. Increase the current to obtain increased torque levels.

- Acceleration : Adjustment of the rate of acceleration , & monitoring effects of such upon path error, velocity fluctuations etc.

- Phase Orientation : Established design rule concerning series / parallel orientation of the stepper motor phase windings. In parallel orientation a wider torque/speed range is achieved, whilst in series greater torque at low speeds is generated

- Micro Stepping : Application of micro-stepping to system under test. Measuring effectiveness of micro-stepping in terms of achievable machine performance.

- Mechanical effects : Performance limitations caused by mechanical errors & disturbances. These include correct alignment of pullies and gear ratios etc.

- Drive Effects : Measurement of the effects that a variety of electronic drive systems have upon the system. Measuring the capability of the drives in full-step / half-step & micro-stepping modes of operation.

- CNC Machine : Validation of the previous results through the instrumentation of a production CNC machine.

- Simulated Loading: Measurement of CNC performance whilst a suitable cutting force is applied against the direction of travel.

Note : The results presented contain additional velocity fluctuations due to the misaligned pulley, this was not intentional and involved a considerable amount of experimentation to prove as such.

3.2.2 STEPPER MOTOR RESONANCE EFFECTS

The resultant motion of the machine can be very rough in nature, as can be seen in Figure 33, this velocity fluctuation leads to path inaccuracies that were previously thought to be due to motor resonance, and possibly by providing the stepper motor with too much current.

Stepper motors generally have two resonant frequencies, due to the nature of the stepper motor. Typically the stepper motor will have a number of phases and a number of teeth, each of these results in a different natural resonant frequency. Acarnley [27] gives a very good account of resonant frequencies of stepper motors and their effects. The resonant frequency of a stepper motor/load can be expressed as :-

$$\text{Natural Frequency,} \quad fn = (T' / J)^{1/2} / 2\pi \qquad (3\text{-}1)$$

Where : -

T' = stiffness of the torque/position characteristic.

J = Stepper Motor / load inertia.

The resonant behaviour of the stepper motor can lead to a loss of motor torque, which could then lead to the system stalling when a load is moved at these stepping rates, which under normal circumstances would be quite tolerable. Therefore it is natural to assume that stepper motor resonance can be a main contributing factor to what the author calls dynamic displacement errors, or more commonly known as rough motion.

If the reader studies the velocity profile shown in Figure 33 carefully, then the effects of actuating the stepper motor at a particular resonant frequency can be seen half way though ramping up. The reader maybe able to notice small fluctuations or vibrations superimposed upon the output waveform at this point. The frequency of these vibrations corresponds to one of the natural frequencies of the stepper motor, approximately 500 - 750 Hertz. When a signal generator was used to operate the stepper motor within the complete range of frequencies then the second resonant period was found. This frequency was discovered to be around 50-60 Hz, suggesting that this frequency may arise initially from mains voltage regulation in the drive circuitry. Essentially there are three systems coupled together, the electronic drive system, the stepper motor and the slide-table/test rig. When extra capacitance was added to the drive in order to remove any ripple voltage and the test repeated the resonance period was still found to be present. The author then removed the pulley belt, removing the added inertia of the slide-table from the load torque, again when the test was repeated the resonance period was still present. The author concluded therefore that the likely cause for this resonance period would be the stepper motor itself. One factor to consider is how the rotor is situated within the stator housing, since there are 50 rotor teeth per revolution this seems to be a probable explanation for the resonance period.

From Figure 33 the reader can deduce that by stepping over, hence avoiding these resonant frequencies then the errors due to these resonant frequencies can also be minimised at best, however with multi-axis machines this is not possible since the resulting speed is a function of two velocities. Effectively the vector sum of the x and y axis velocities, therefore the result can be easily fall into the range of one of these resonance periods. Even with feedback control the controller can only dampen the resonant vibrations within the torque capabilities of the stepper motor.

The results in the form of an FFT, Figure 34 show that the second resonant period (50-60Hz) to be the dominant frequency in terms of the magnitude. The reader can see from Figure 33 that the vibrations at the first resonant period are insufficient to cause anything but the minimal amount of path errors. The magnitude of the second resonant period is greater but unless the CNC machine is under other duress, i. e. complex path following, then these vibrations are again insufficient to cause the motor to stall. This in some way rules out the theory that operation at resonance periods is the main cause of the stepper motor performance limitation. However this is only applicable to the systems under test where the drives and stepper motors used have many times the torque level than the level required to drive the load.

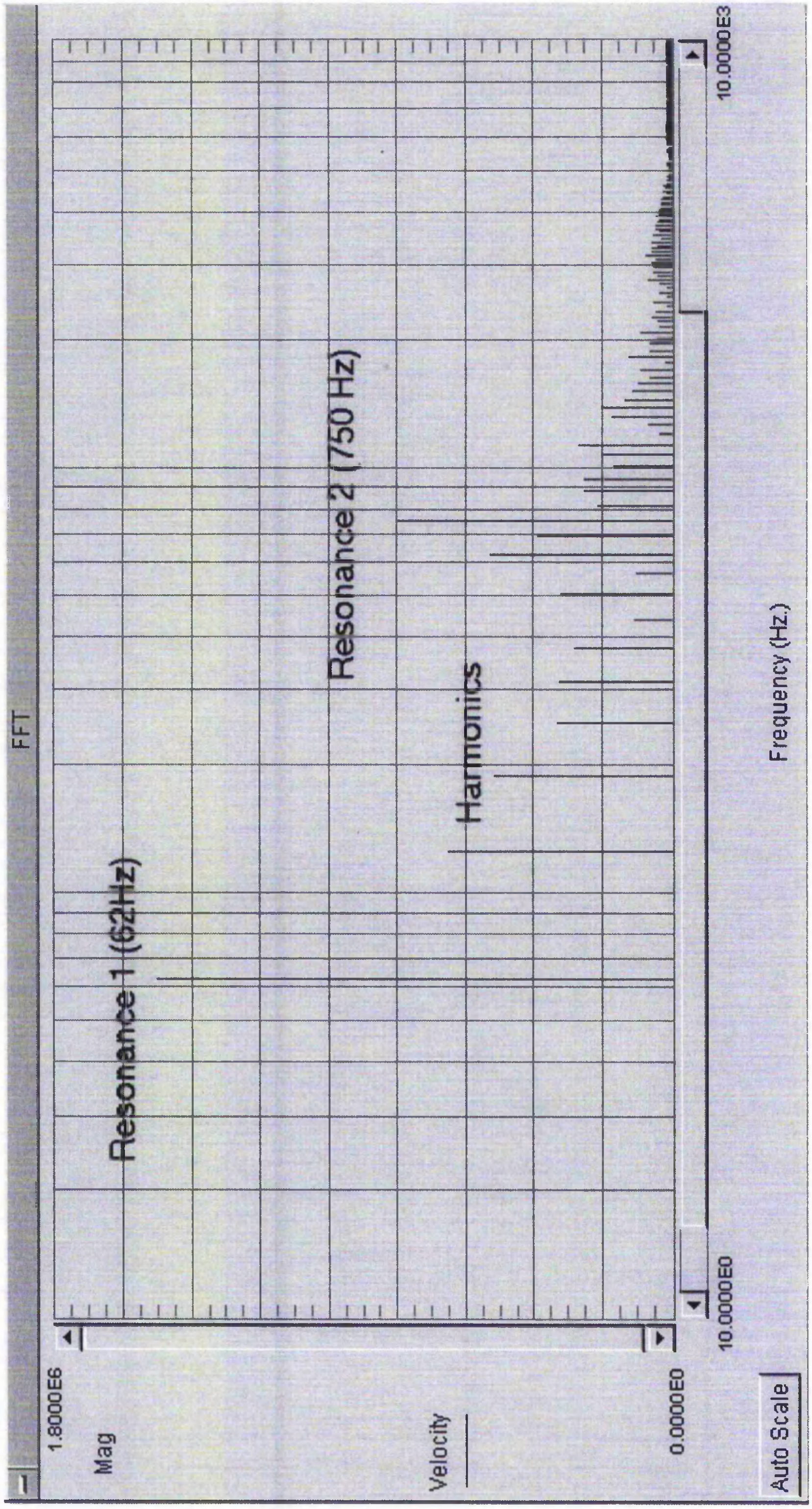


Figure 34 : FFT Manitude Spectrum of Standard Velocity Profile .

3. 2. 3 ADJUSTING THE STEPPER MOTOR WINDING CURRENT

The reader can see from Figure 35, that the dynamic response of the system can be seen to be more damped in nature as the current is reduced. The comparison is made against normal motor operating current, with the top velocity of the machine kept to a constant one meter per minute for both tests. The dynamic displacement error is also consequently reduced.

The reader may take note of the smaller amplitude, higher frequency vibrations that are also superimposed upon the waveform. These higher vibrations can be seen to occur only once the machine has settled down to the required velocity. The higher vibrations are due to the stepper motor being driven in half step mode, where the available torque changes between each half of the full step. The results indicate that whilst the main demand is being placed upon the system, i. e. accelerating to full speed, then these step vibrations are dampened by the increased load. Once this increased load has been removed then the stepper motor has potential to vibrate at the full step frequency (one half of the operating frequency). To conclude there is a relationship between dynamic displacement error and motor current, and reducing motor current can help to minimise this, in our case by 50 to 60 percent. This figure was deduced by comparing the velocity error from figure 33 against the corresponding velocity error with the motor running at 2m/min and reduced current.

Available torque is linked to the stepper motor current, an increase in current will correspond to an increase in torque. The available current for a given motor can be adjusted by changing the resistance of a potentiometer on the drive. In some cases where the demand torque is high (a sufficient mass on the machine) then the stepper motor current could be increased so that the motor could deliver more torque where it is needed (rapid direction change). The machine would also be sufficiently damped by the increase of mass, and viscous damping indirectly due to this increased mass. The reader can see from Figure 36 that as the motor current is increased beyond the point where magnetic saturation occurs, that the dynamic response of the system becomes erratic. The ramping up phase of the velocity becomes a period where the motion of the machine is unpredictable, periods of sudden acceleration and deceleration. This is contrasted against a normal output response where the excitations are sinusoidal based. Although the motor current is increased the peak dynamic displacement error does not increase by the same amount. Due to motor saturation the peak dynamic displacement error also seems to saturate at this maximum figure.

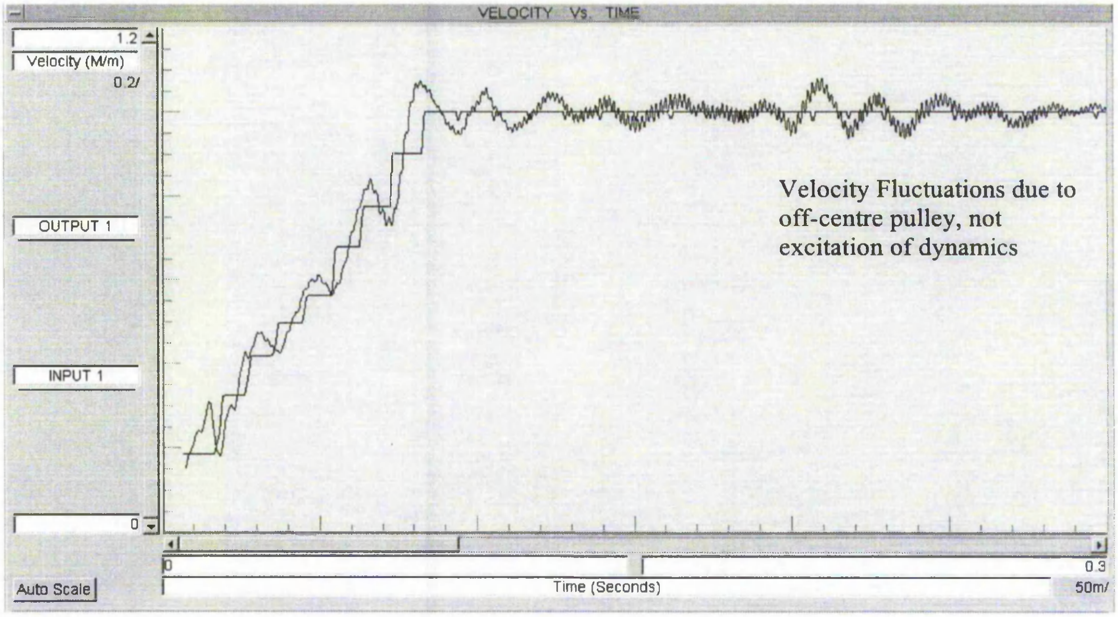


Figure 35 : Velocity Profile with Reduced Motor Current.

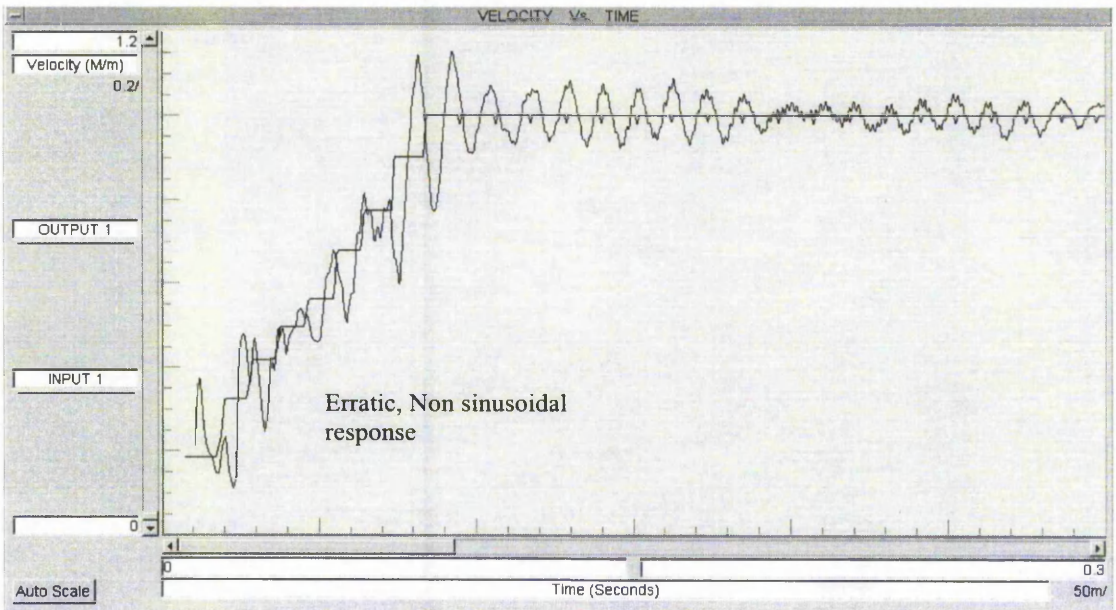


Figure 36 : Velocity Profile with Increased Motor Current.

3. 2. 4 THE EFFECTS OF CHANGING THE ACCELERATION RATIO.

The reader may take the opinion from the results that if the stepper motor was allowed to accelerate gradually then the response of the machine would be less oscillatory in nature. This view of the dynamics of the machine is generally correct, but not entirely. If the acceleration ratio is decreased the machine will have more time to respond to the change, and therefore the machines resultant response should be able to match the input response.

However the acceleration phase is a series of stepped changes, therefore the input to the machine is a step input, with a corresponding underdamped response each time the velocity is changed. Worse still is the fact that now we are allowing the machine more time to respond to this sudden change resulting in a larger number of dynamic displacement errors. The author originally took the view that decreasing the acceleration ratio might help solve the displacement errors, so he conducted a series of experiments to verify his initial thoughts. The author found from the results that the answer is not so obvious. The resultant dynamic response for increasing the acceleration with the normal amount of current energising the windings is shown in Figure 37. The reader may observe that the time for the system to respond is shorter, hence the number of overshoots is lower. However because the acceleration is increased the amplitude of the overshoots is equal or greater than normal.

When the motor current was decreased, and the system response subsequently recorded, it was observed that the combination of the two parameters greatly improved the system response. The system response can be seen in Figure 38. The system response shows that the combination of increased acceleration ratio and reduced motor current can reduce the dynamic displacement error by a factor of 60-70 percent. The disadvantage with using this set of parameters is that only light machining can be conducted, because of the reduced available torque.

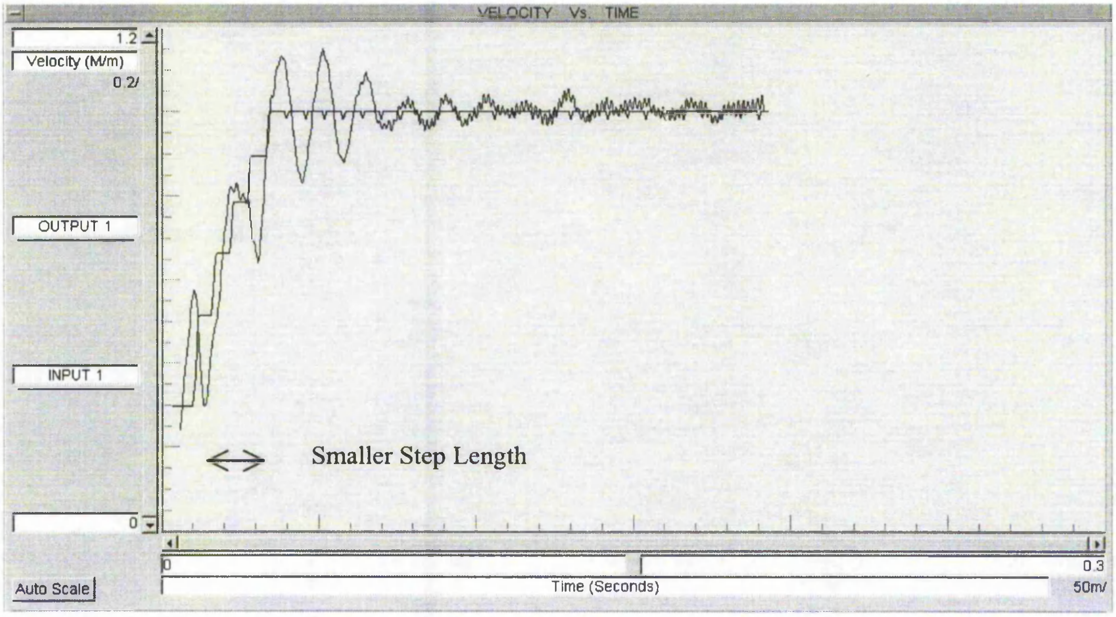


Figure 37 : Velocity Profile with Increased Acceleration.

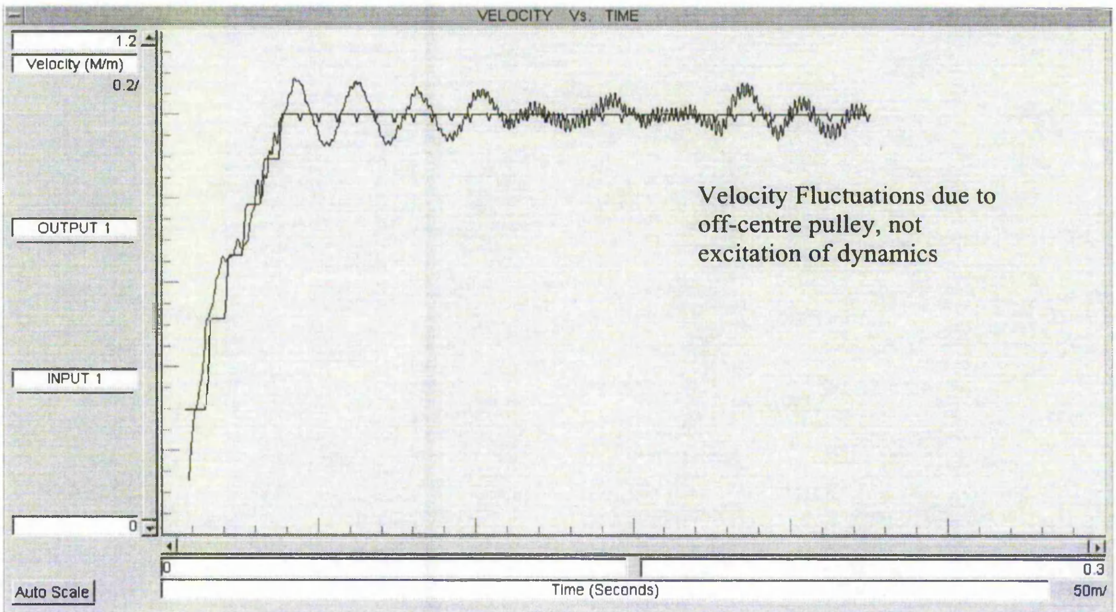


Figure 38 : Velocity Profile with Increased Acceleration and Reduced Current.

3. 2. 5 MICRO STEPPING

The investigative field was increased to include the micro stepping technique of driving stepper motors. Normally to provide a higher degree of resolution a stepper motor is driven in half step mode, meaning instead of switching off one phase and then switching on the next, both phases are energise together for a short duration. This results in the rotor of stepper motor taking up a position half way between the two phase windings, hence the name. The principle of micro stepping goes one step better, micro-stepping works by dividing the current between the two phase by a number of discrete steps. Now the current from one phase is reduced from the maximum, one step at a time until there is no current flowing though the phase. Likewise the current flowing though the second phase increases corresponding to the decrease of the first phase, until the maximum setting is reached. The end effect of this process means that the stepper motor has a smaller step size, hence higher resolution.

In practice micro stepping drives are not so straightforward, some drives offer very good micro stepping capability whilst others compound the problem and offer no measurable benefit. The author in conjunction with Axiomatic Technology Ltd performed tests on a number of micro stepping drives. The system response to the micro stepped input can be seen in Figures 39-41. The reader can see from Figure 39 that due to micro-stepping, the length of the steps of the input velocity profile are smaller, and less coarse. The output response of the system to this input can be seen to be less oscillatory in nature, the changes in velocity become gradual changes. This is clearly evidenced in Figure 40 where the acceleration ratio of the motor has been increased, here the output nearly follows the input demand, this can be seen in greater detail in the magnified picture of Figure 41.

To summarise a micro stepping drive can be used to not only provide a greater deal of resolution but also help to eliminate dynamic displacement error. Nothing is without cost however, the cost of micro stepping is the increase in frequency of the pulse stream provided by the controller. Most controllers have a maximum frequency at which they can be interrupted. Micro stepping reduces the velocity range and available torque significantly.

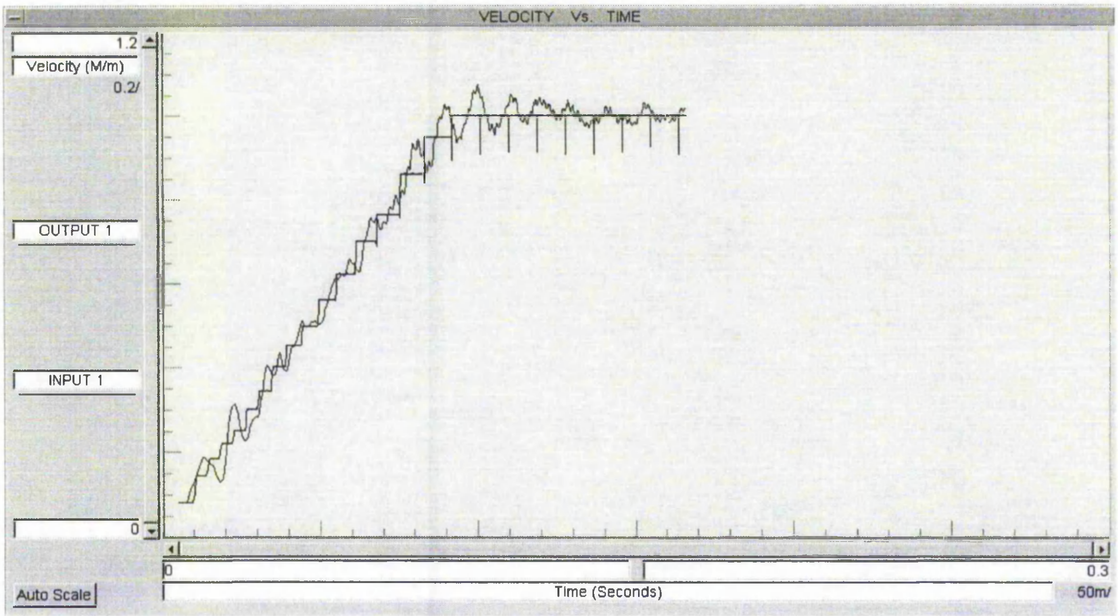


Figure 39 : Velocity Profile using Micro Stepping Drive Unit.

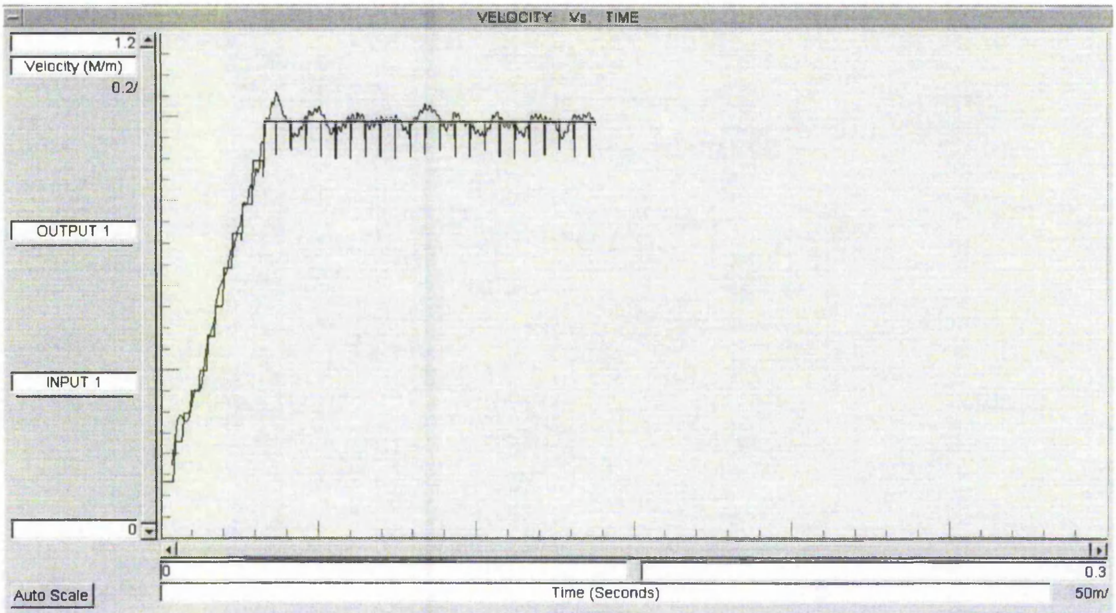


Figure 40 : Micro Stepped Velocity Profile with Increased Acceleration.

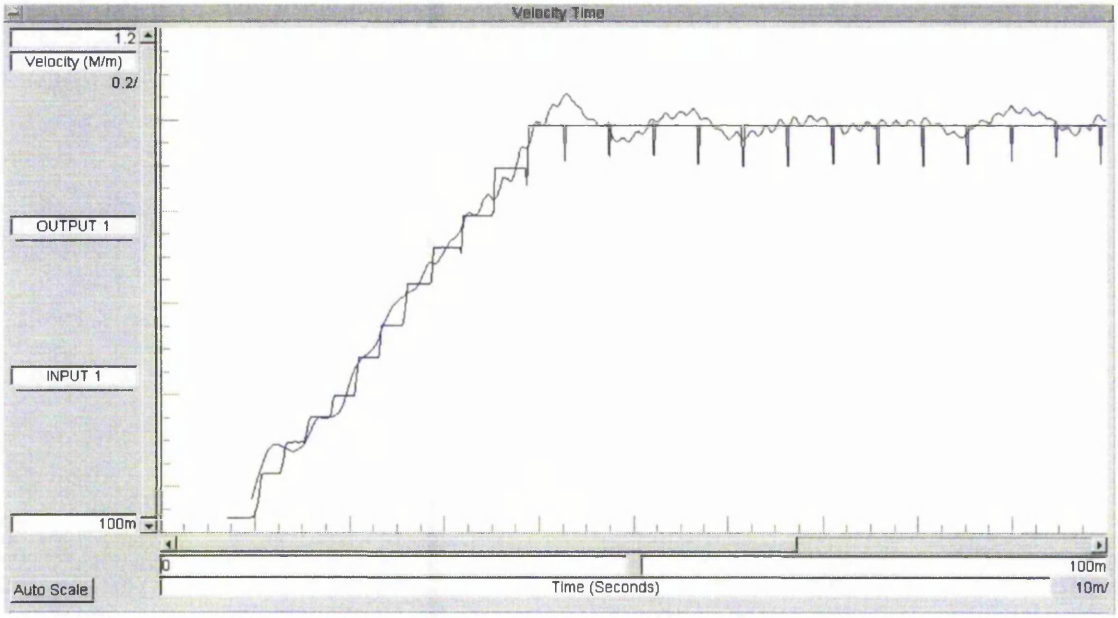


Figure 41 : Magnified Micro Stepped Velocity Profile.

The use of micro-stepping drives can mean that the maximum operating speed is limited by the output bandwidth of the machine controller. In this case the maximum operating speed will be reduced by the same magnitude by which the resolution increased. Therefore if a micro stepping value of twenty is used, i.e sub dividing the resolution by 20 then the maximum speed that the controller can output will also decrease by a factor of 20.

3. 2. 6 STEPPER MOTOR PHASE CONFIGURATION.

Manufacturers generally recommend that for applications requiring a wide speed range then the parallel connection of the phase windings is preferred. This has the additional advantage of minimising the effect of motor resonance. If a series connection is chosen then greater peak torque is achieved, albeit for a much reduced speed range. See Figure 42, a comparison of the two choices, comparing torque against speed.

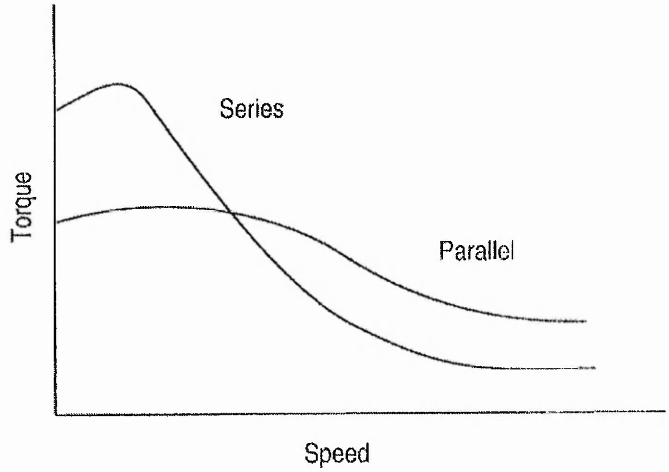


Figure 42 : Comparison of Phase Orientation vs. Torque.

Experimentation has been conducted in the past by Axiomatic Technology Ltd. into the effects of wiring the current phases of the stepper motor in series or parallel orientation, and the effects of increasing the supplied voltage/current with regard to developing increased levels of deliverable torque.

In summary, for contouring systems with frequent speed changes then a parallel phase orientation would be the optimum choice. It can be seen from Figure 42 that the system designer can obtain more high-speed torque from the stepper motor if the windings were wired in parallel orientation. If the voltage is also increased then there is a respective increase in the maximum velocity that the motor can operate under. The benefit of these phase orientations is a matter of choice depending upon the application. For CNC machines high-speed torque is of essence so parallel orientation is normally adopted, with the maximum speed dependant upon the voltage. The power to the drive system is compromised by price of the electronic components. The component cost is exponentially related to the increase in power capacity.

3. 2. 7 EFFECTS OF MECHANICAL TOLERANCES

The reader may have noticed that superimposed upon the results is combination of two oscillations, or a beating effect. The cause of this beating effect is the imperfect nature of the experimental machine platform. The slide table is driven though the process of transferring drive from the motor via one pulley, to a drive shaft and then via another pulley to the slide table. When the pulley centre is drilled and reamed there remains some tolerance error. This non-concentricity causes an end effect when the pulley is mounted onto a drive shaft. The belt, which is attached to the pulley, fluctuates in speed as the outside radius of the pulley changes. This fluctuation in velocity, as we have seen can lead to dynamic displacement errors which are sinusoidal in nature. When the system is fitted with a twin pulley network then each pulley can cause these effects and often add together or subtract from another such that a range of effects can be viewed.

This is the case for the experimental test platform where the two pulleys oppose each other for one half cycle then add together for the next half cycle, this can be seen as a beating effect superimposed upon the waveforms. This can be seen especially seen in Figure 43, where a longer cruise section of the velocity profile is shown. This mechanical noise causes a dynamic displacement error of up to 3-5 microns.

3. 2. 8 INACCURACIES IN DRIVE TECHNOLOGY

Any velocity fluctuations caused by the drive system could not be clearly distinguished from those caused by the interpolator until after the author prototyped a velocity profile generating stage. This circuitry was produced to confirm the findings of Palmin and is discussed within the next chapter. After these improved velocity profiles were generated and fed into the system other interesting effects were observed. These were observed earlier but due to the previous large velocity fluctuations were very difficult to discern. The improved velocity profiles allowed the author to study the effects that the drive system had upon the resultant smooth continuous path motion.

It can be observed from Figure 43, that there are very small fluctuations in resultant velocity. These could only be seen when the data was collected from the rotary encoder attached to the end of the stepper motor. When a section is magnified then the reader can see from Figure 44 velocity fluctuations caused by the drive system. The drive system used was a standard off the shelf drive system with no faults and readily used by a number of commercial CNC machine manufacturers. The drive was operated in half step mode where two phases are energised simultaneously such that the poles of the rotor assume a position that corresponds to halfway between the two phases. What Figure 44 shows is that the current for the phases is not equally matched so one phase obtains more current than the other. The result is such that the rotor assumes a position that is only one quarter of the distance. Then when the energy is fully transferred to the other phase the stepper motor has to move the remaining three-quarters of a step. This can be visualised as a small velocity fluctuation caused by the stepping action, followed by a much larger one. However this causes a displacement error of 5-7 microns, which is lost against the displacement error caused by the excitation of the dynamics which can be up to 30 microns in magnitude.

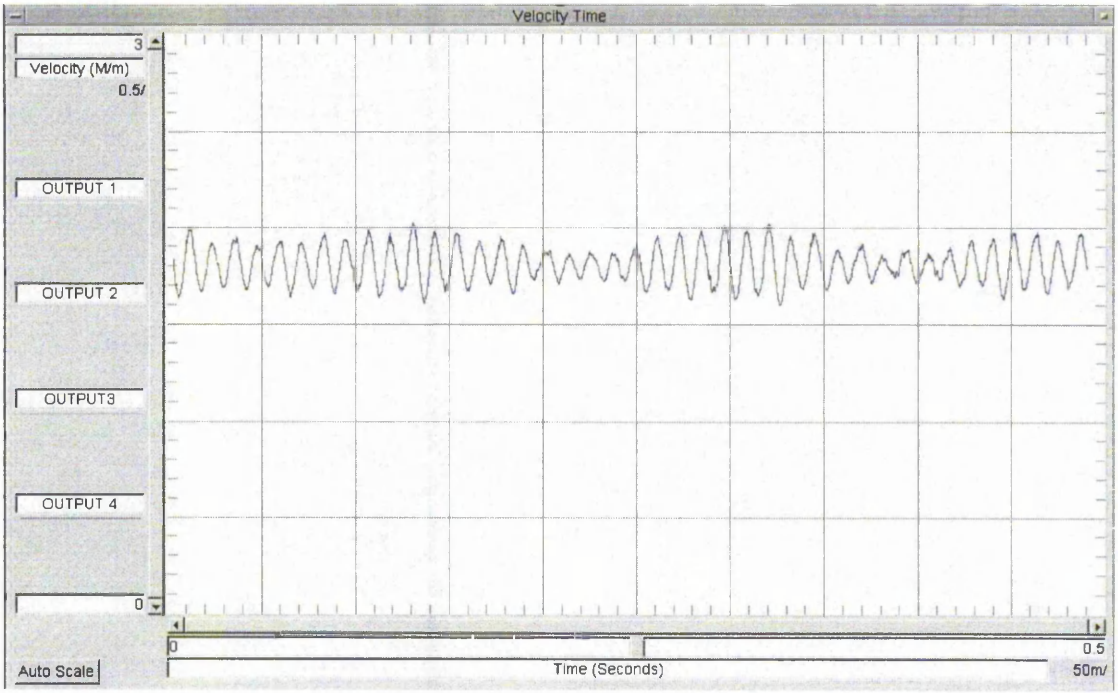


Figure 43 : Velocity Fluctuation due to Off Centre Pulley.

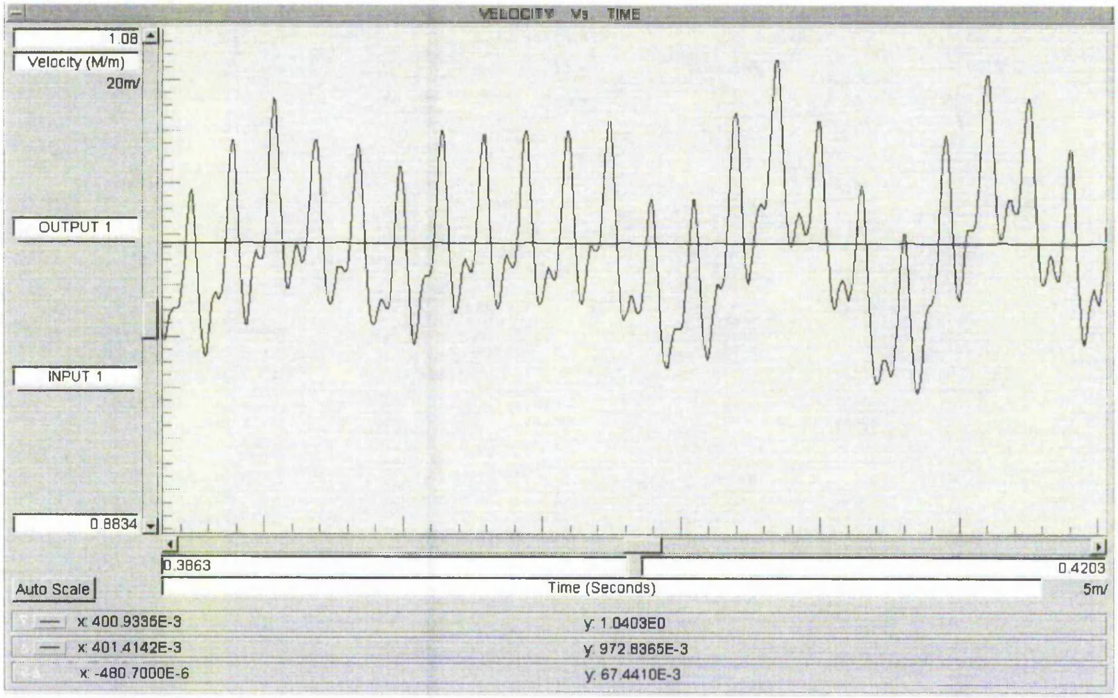


Figure 44 : Magnified Velocity Fluctuations due to Drive Inaccuracies.

3. 3 Production CNC Machine Results.

3.3.1 Velocity profile from unloaded Cadet machine.

The velocity profile shown in Figure 45, was taken from results obtained via the linear encoder attached to the end effector of a Pacer Cadet machine. The machine had two encoders attached to the x-axis drive transmission. A shaft encoder was attached to the opposite end of the rotor shaft (a double shaft motor was purchased by Pacer to accommodate this placement). This encoder was an 'off the shelf' sealed encoder unit with a resolution of 2000 lines per revolution. The Haidenhain linear encoder, taken from the experimental test platform took the role of the second encoder. This encoder was set to a resolution of 1 micron and was attached to the tool-mounting spindle, i. e. the end effector. The linear encoder glass was attached to the machine bed securely, then the read head was moved along the y-axis to the required distance to obtain readings, (0. 5mm). The encoder glass was placed such that this distance remained constant over the complete length of the encoder glass, the bed was also skimmed to ensure the glass was level. The pulley belt connecting the y-axis stepper motor to the leadscrew was disconnected to prevent accidental damage to the encoder glass.

With the encoders securely attached the machine was set to machine along a straight line. The objective of this experiment was to verify, if the measured velocity fluctuations from the Cadet machine were in the same order of magnitude as those measured from the experimental test rig. The results show that if the Pacer Cadet machine is subjected to the same input velocity profile, under no load conditions, it does suffers from velocity fluctuations within the same magnitude as those measured from the test rig.

These fluctuations also recede once the accelerating phase of the velocity profile is over. The reason they do recede and do not have a cyclic bias is that the pulleys attached to the lead-screw/stepper motor shaft are machined to be concentric to the shaft in question. The pulleys attached to the experimental test platform were machined such that they can easily be placed on and off, to allow for ease of experimentation. The drawback is that all the results obtained from the experimental test platform have a cyclic bias to the velocity, caused by a non-concentric pulley.

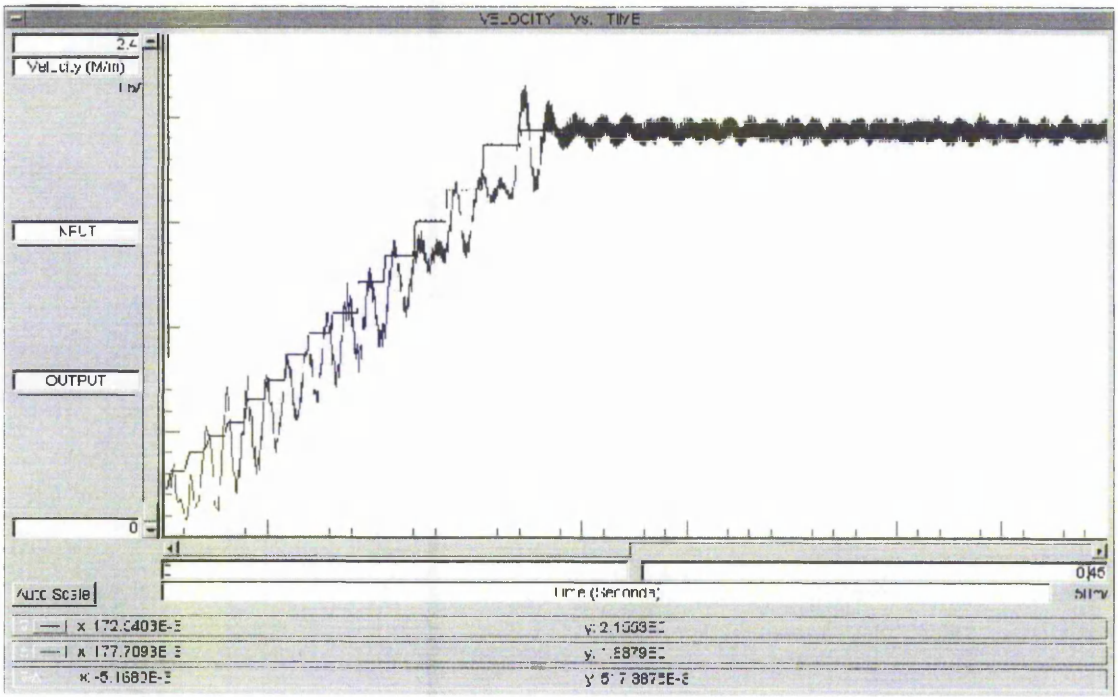


Figure 45 : Velocity Profile from Unloaded Cadet Machine.

The peak-peak deviation from the input velocity can be measured to be in the order of 0. 5 M/min. This amounts to a 25% fluctuation in velocity. If the peak velocity fluctuation is measured, from Figure 40, in terms of acceleration then it can be seen that the actual acceleration that the machine is experiencing is 22 % G. rather than the theoretical 2 %. This goes along way to explain why stepper motor driven machinery is often limited to one tenth of its real performance capabilities.

Once the accelerating phase has finished, and the dynamics have settled the reader can see from Figure 45 that there is a banding effect super-imposed upon the results. Upon magnification the reader can see from Figure 45 that this banding effect is a vibration at the full-step frequency, or half of the operating frequency. In comparison to Figure 44 the reader can see that in Figure 46 there is no distortion of the waveforms caused by drive inaccuracies. The reason for this is simply down to the motor/drive combination. The drive used by the Cadet machine is produced to a higher tolerance than the drive system used for the experimental test rig.

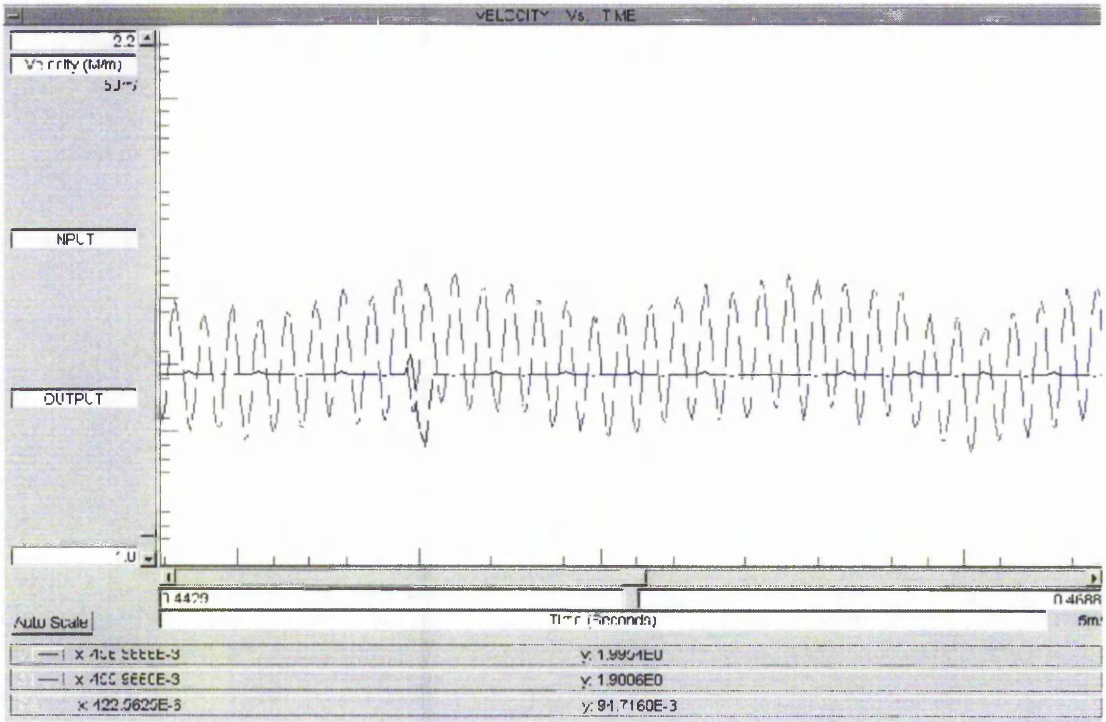


Figure 46 Magnified View of Velocity Fluctuations.

3.3.2 Velocity profile from loaded Cadet machine.

After the validation phase of the testing was complete, one question remained; what happened to the resultant velocity under load conditions. Testing the end effector under machining conditions was not possible due to many factors, but mostly time. The machine used for experimentation was a production machine due to be delivered to a customer, hence Pacer was unwilling to sell a second hand machine. In conjunction with a design engineer from Pacer who had been experimenting with predicting load forces, a simple load test was designed. The test in question involved a mass of 5 Kg, string and a pulley. The mass was tied to the string, with the other end attached to the end effector. The pulley was used such that the weight was suspended vertically to the side of the machine. The reasoning, was to subject the end effector to a side force of approximately 50 Newtons in the opposite direction to that of the motion. The design engineer derived this Figure from early experiments that he had conducted. The velocity profile data obtained from the linear encoder can be seen in Figure 47.

The reader may observe from Figure 47, that the sidewards loaded of 50 Newtons does in no way damp out the velocity fluctuations seen in Figure 45. It can be seen from Figure 47 that the simulated machining load actually prevents the velocity fluctuations from receding quickly, doubling the time it takes to settle to a background level. The magnitude of the velocity fluctuations however hardly increases, remaining at around 25 – 30 % deviation. This was in some ways unexpected since the engineers at Pacer believed that their machines could not possibly deviate in velocity by this amount and that the operation of machining would reduce this by a significant amount. The author thought that the fluctuations might diminish slightly but the compounding evidence concerning the limitations in stepper motor driven machinery suggested that this would only be minor.

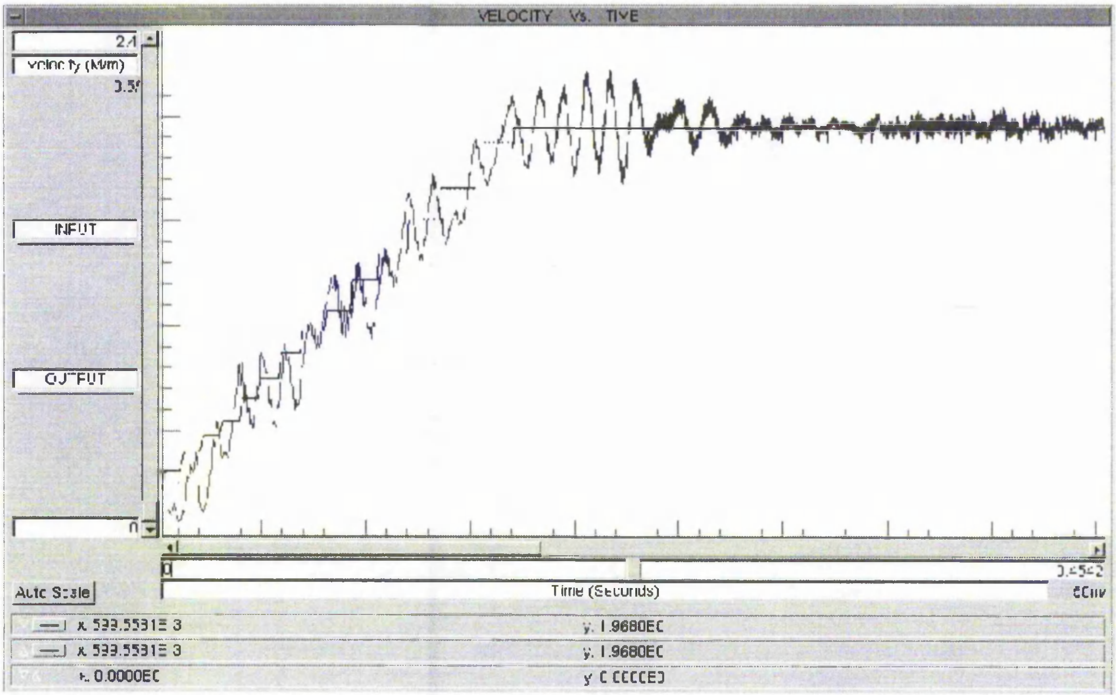


Figure 47 : Velocity Profile from Loaded Cadet Machine.

3.4 Significance of Experimentation.

The results of the experiments conducted on the test rig have shown that the principal causes of the performance limitation in multi-axis machines are the excitation of the machine dynamics. This excitation manifests itself as velocity fluctuation. These fluctuations cause the performance limitation by reflecting additional load upon the motor. The consequence of this reflected load is that the system has less available dynamic torque availble due to the velocity fluctuations caused by the excitation of the dynamics. This means that the available speed and acceleration are greatly reduced, see Figure 48. In Figure 48 the y-axis represents torque and the x-axis represents velocity. The maximum performance of the stepper motor is represented as the plotted curved line. The difference between this line and the line representing the machine load indicates the amount of reserve torque the motor has before it will stall. This reserve torque is used to accelerate the stepper motor. Hence from the graph the reader should see that with the reflected load the stepper motor has a limited amount of reserve torque in which to accelerate, and that the top speed has been significantly reduced.

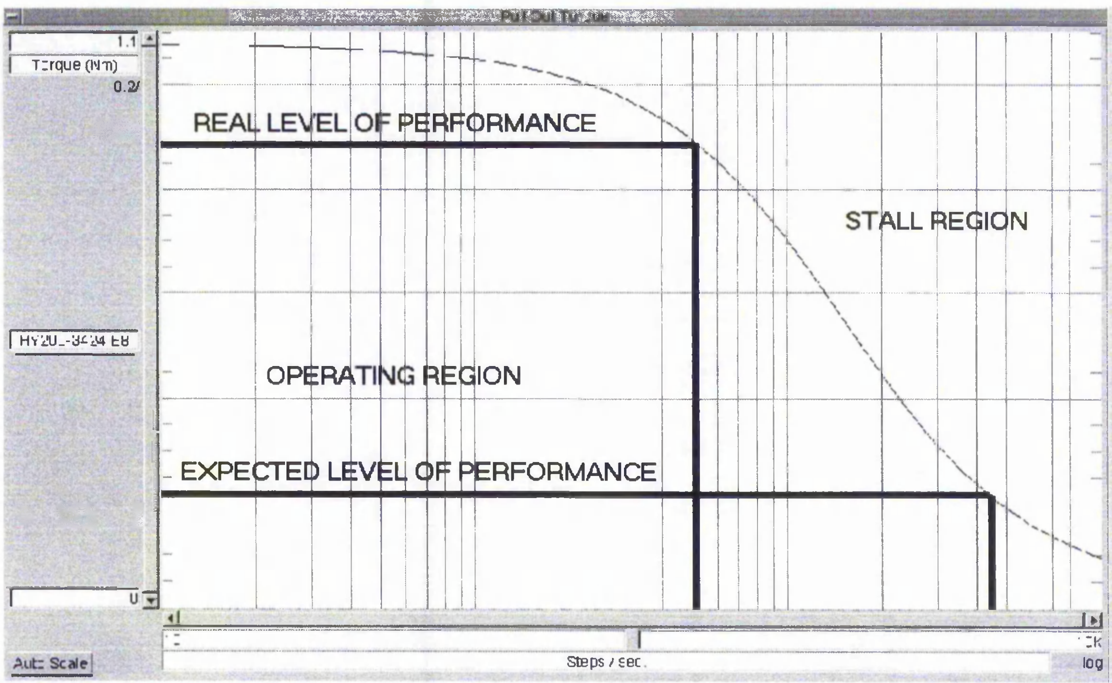


Figure 48 : Effect on Available Torque by Reflected Load on Stepper Motor.

The results obtained from the experimental test rig were verified by instrumenting a production CNC machine. This verification confirmed the findings that rough motion causes a performance limitation in multi-axis stepper motor driven CNC machinery. Arising from the experimentation is also the validation of several design rules concerning rough motion. The results show there is a relationship between dynamic displacement error and motor current, and reducing motor current can help to minimise this. This was quantifiable in the fact that operating at a level of 66% of the nominal current reduced the displacement error by one half. If the acceleration is changed such that the rise of the first overshoot corresponds to the next step demand then the results show that there is insufficient time for the dynamics to be excited. This can be seen in Figure 38 (page 84), where the current has been reduced to dampen the final step response. This technique in a part measure meets the requirements of finding a way of overcoming the performance limitation. However each drive and control would have to be tuned to each particular machine, and this is hardly a solution. Another disadvantage is that only light machining can be conducted, because of the reduced available torque.

Another technique that meets with some success is that of micro-stepping or having a low gear ratio. They result in the same thing, smaller resolution of step distance. This means that the interpolation steps are smaller, hence less excitation of the dynamics. However the disadvantage is that the maximum top speed is remarkably reduced by the bandwidth of the controller. Since the distance steps at the end-effector are finer in resolution then a greater number have to be generated to maintain the same speed in terms of metres per minute, hence there is still a limitation in performance.

In Summary the reader should hopefully see that the excitation of dynamics caused by the interaction of acceleration & interpolation, does present a problem in stepper motor driven machinery. The author has verified the postulations of Steiger et al. by the experimentation conducted upon the test rig. The author went a step further and validated the experimental work by instrumenting a production CNC machine supplied by Pacer Systems Ltd. The author concludes the experimentation showing how, by changing system parameters, that the rough motion can be lessened to some extent, but still with no definite solution available. This lead the author to model the experimental test rig in order to find a solution and to also confirm the findings of Palmin et al. [50].

Chapter Four

System Modelling & Motion Profiles

Chapter Four System Modelling & Motion Profiles

4.0 The Process of Simulation.

The Process of simulation is by definition the use of a system, often a computer, to artificially create the appearance of another. The essential feature is that the action must be capable of being represented by some mathematical function or model. Simulation allows designs to be analysed and modified without having to go to the effect, expense, and time of building a prototype. Simulation can also be used to perform analyses that are not always possible or desirable on a physical design such as worst-case component tolerance sensitivity analyses and simulating faulty circuits.

The author turned to simulation to experiment with various velocity profiles, and to confirm the findings of Palmin [50]. The process of simulation was chosen for the last reason presented above, in that the author did not want to damage the experimental test rig. There are a number of modelling approaches that can be used with control systems. Whereas mathematical models based on the chemistry and physics of the system represent one alternative, the typical machine control model utilises an empirical input/output relationship, the so-called black-box model. These models are found by experimental tests of the process. Once the parameters have been determined by systems identification, one can proceed to analyse the overall system dynamics, the effect of different controllers in the operating process configuration, and the stability of the system, as well as obtain other useful information.

Once the various ideas were tried and tested on the simulator the author validated the theories by developing a velocity profile generation system. This system in the form of a digital signal processor generates the appropriate pulses for the drive system of the experimental test rig. Thus the designed velocity profiles can be tested in practice. The author used this system to confirm the findings of Palmin [50].

4.1 Detailed Description of System Modelling.

Mathematical models in the form of polynomial equations can be used to describe any number of different objects from curves to the mechanical behaviour of CNC machines. To obtain more useful information from the experimental test platform, that cannot be readily obtained from the results, a transfer function would be needed. The transfer function of a system describes the relationship between input and output of the system and can be used to obtain the magnitude of output response to a given input. The bode plot, frequency plot and the pole/zero plot can also be obtained from the transfer function. The transfer function is derived from the mathematical model of the system, and as such provides the control engineer with in-depth knowledge of the system. The accuracy of the data is dependant upon how accurately the system is modelled.

Analysis of any control system can be achieved by one of two ways. The first method is to build a mathematical model of the system in the form of differential equations, then apply Laplace transforms to the model to determine information such as the pole placement etc. This typically involves prior knowledge of the systems physical parameters, often involving in-depth knowledge of the materials and internal construction of each part of the system. The alternative method is to use systems identification. Systems identification is the process of approximating the systems parameters for a given computer model by measuring the input/output responses through experimentation. This provides the engineer with a powerful tool with regard to understanding the system in question, provided they can capture information from their system.

The use of data modelling and representation tools such as Hewlett Packard's Visual Engineering Environment (HP VEE) and Mathworks MATLAB allow the design engineer to analyse the system in question in quasi real-time. HP VEE is a graphical programming language for creating test systems and for solving engineering problems. The user can select from a variety of standard mathematical formulae, and special user supplied functions can easily be added. Shown in Figure 49 is the block diagram for generating a FFT for a given input signal.

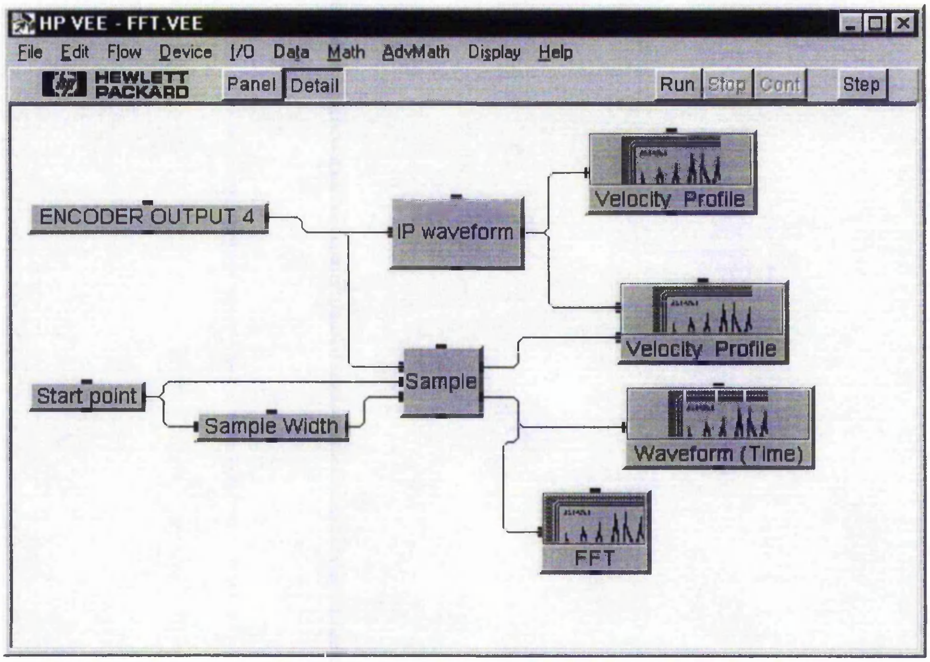


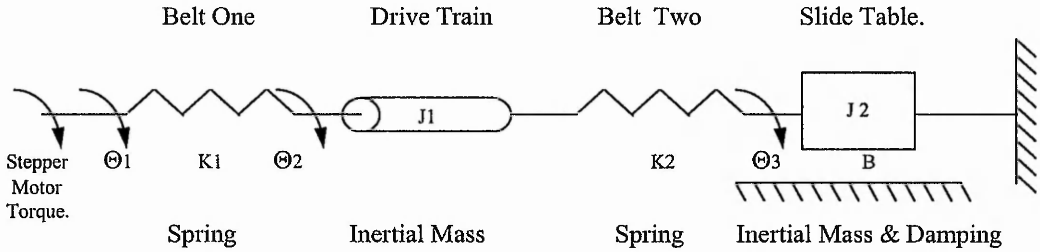
Figure 49 : HP VEE Interface.

In the Matlab environment in order to build a model with the systems identification package the user firstly needs the input demand and the output response signals. Secondly the user needs to know the order of the mathematical model required. The computer can be used to assist in this prediction, but a far better approach is to build a mathematical model by analysing the construction of the system. This way the user can guide the prediction more accurately than an unassisted approach. The model to be chosen should be based upon the nature of work to be conducted, is a general behaviour desired or should a more complex model be chosen. Researchers such as Kenjo [28] have shown how the electrical characteristics of a stepper motor can be modelled as a fourth order differential equation.

The author wanted to model the general behaviour of the experimental rig rather than a precise model which describes everything including elements such as the control and drive electronics. The reason for this is that the excitations predominantly arise from the general construction of the experimental rig rather than the drive. The second reason is the level of precision; a complex model may form the sole basis of a research project but will be of little use if the model is incorrect. However a generalised model takes less time and still provides the engineer with about 80% of the information that could be obtained from the complex model. The author analysed the nature of the experimental rig in terms of control elements (mass, spring, damper) and represented the system in a physical entity diagram shown in figure 50. This diagram can then be represented as a quadratic equation. At this point the order of model is determined, however the constants still remain unknown. From this information the author determined that for the generalised behaviour of the experimental rig then a model of approximately third order would be sufficient. From further analysis the author concluded that most CNC drive trains can be adequately modelled as a low order system (a lead screw indirectly coupled to a stepper motor drive can typically be modelled as a third order system).

Once the order was determined then the objective was to provide a set of discrete time sampled data to the Matlab environment in order for the system parameters to be identified. The author resampled the discrete distance data from the normal velocity profile found in chapter 3, Figure 33 to convert the data into discrete time based data. In the process of systems identification it is often advised that only a section of the data is used to determine the parameters. This means that after the parameters are determined then the rest of the data can be used as a verification test. This was accomplished and the reader can see this in greater detail in appendix B. Several choices are available for the Matlab user for modelling the data with regard to the model order and the method by which the data is analysed. From the theoretical analysis of the experimental rig, the author decided upon a third order model. Model prediction can be obtained from Matlab by analysing a percentage of confidence versus model order graph. The theoretical analysis provided confidence in the choice since both Matlab and the theoretical analysis suggest that a third order model would suffice. The analysis method that proved to be the most successful in terms of confidence levels was the least squares approach, so this method was chosen. Once these two pieces of information are known then the parameters can then be determined by applying the chosen method and data to the model. The mathematical model of the experimental rig was then determined. The various ways of representing this model, i.e transfer function where determined by applying various Matlab functions to the model. These can be seen in appendix B. The transfer function seen in figure 50 was taken from the modeling process and represents the experimental test rig.

Mathematical Model of System.



$$\text{Transfer Function} = \frac{51s^2+1269s+19165}{s^3+11s^2+2703s+19064}$$

Figure 50 : Mathematical Model of Experimental Test Platform.

Once a model has been established it is possible to produce a simulator of the modelled system. This is conventionally achieved by taking the Laplace transform and converting it to a Z transform by the means of a Bi-linear transformation. From the Z transform it is a small step to convert the transform into a difference equation that can directly produce a numerical simulation. This standard method is embodied within the simulink toolbox for Matlab. The author used this toolbox to build a simulator from the mathematical model shown in figure 50. The input was then fed into the simulator, which produced excitations similar to those experienced on the real machine, Figure 51. Shown in Figure 52 is a comparison between the actual machine output, taken from the results in Figure 33 (page 77) and the output from the simulator. The simulator output is the darker of the two lines.

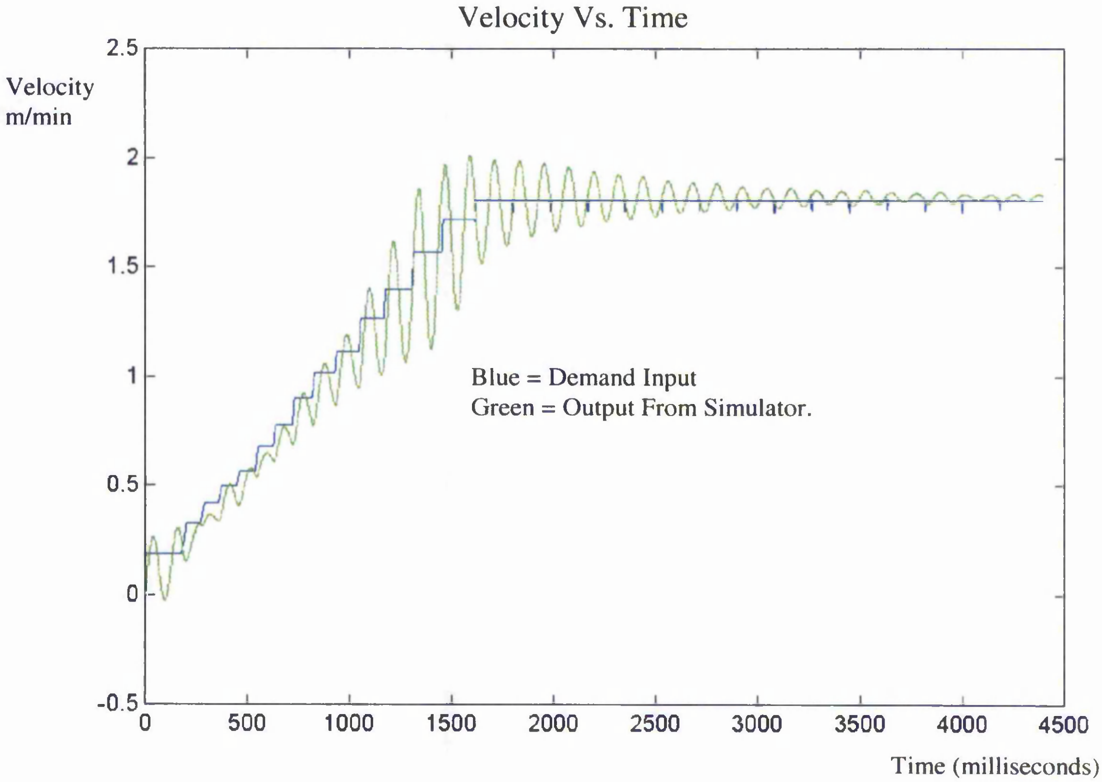


Figure 51 : Simulated Output of Experimental Test Rig.

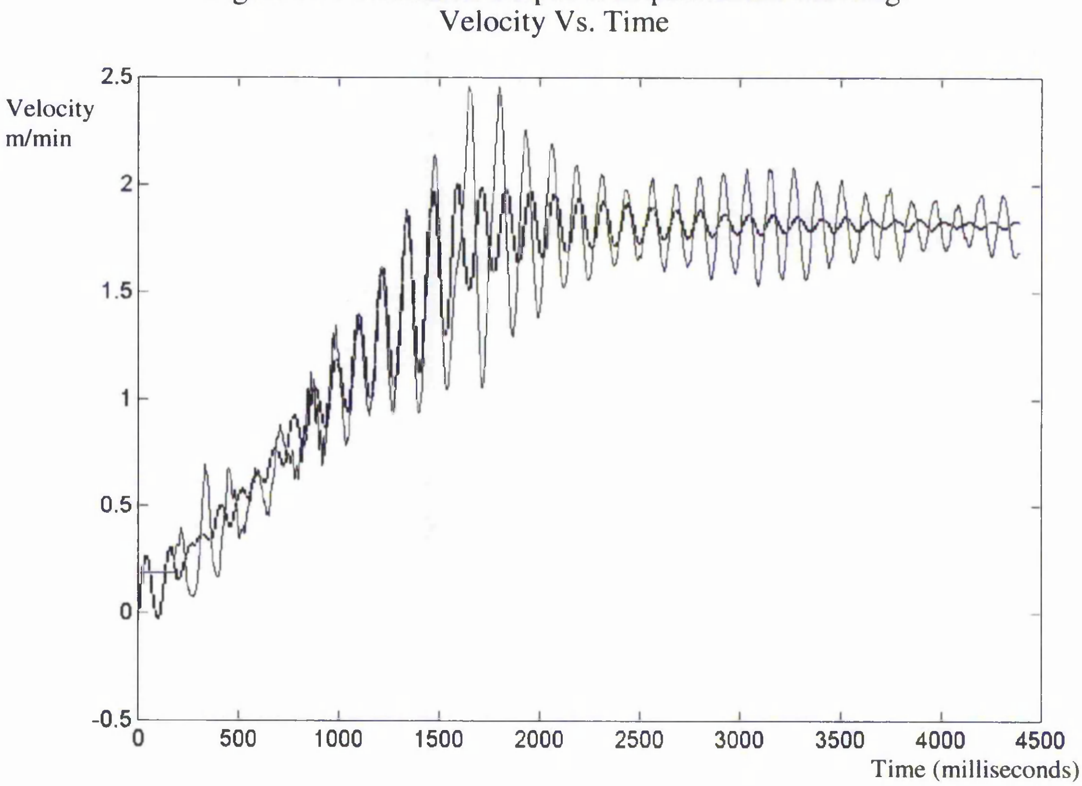


Figure 52 : Comparison of Simulated Output Vs. Actual Output.

The reader can see from Figure 52 that the simulated output response generally matches the actual output response. The two outputs do intentionally differ slightly because the simulator output did not model the effect of the non-concentric pulley. The effect of the non-concentric pulley can now be clearly seen in Figure 52 by contrasting the two responses. This simulator was used to study the effects various other velocity profiles, as mentioned by Palmin[50], had upon the system. These included linear, exponential, inverse exponential and parabolic. The author then developed a digital based velocity ramp generator incorporating a digital signal processor. This meant the data from the simulator could then be cross-checked against the actual output from the experimental rig for the various profiles. The results from the simulator matched the actual results. To avoid clutter and confusion the simulator results are not shown, however the reader can see the results from the ramp generator in figures 55 & 56 later within this chapter. This generator develops timing pulses at the required frequency necessary to generate the required velocity profile for the experimental test rig. Additional experimentation was then carried out to confirm the predictions arising from the simulator. These included amongst others, measuring the obtainable level of performance for parabolic velocity profiles.

4.2 Description of Velocity Profile Generator.

The software based machine velocity profile generator was developed on a digital signal processor. The DSP in question is the Texas Instruments TMS320C50, Figure 53. The DSP belongs to the integer range of the Texas Instruments TMS320 DSP range. The DSP was chosen for a number of reasons. Firstly because of its fast mathematical computation, being able to multiply in a single clock cycle. Secondly the DSP can be purchased in an evaluation package called an EVM board. This board contains suitable interfacing to allow the DSP to communicate to host systems, namely the personal computer.

The DSP features an on-board timer, an essential part of any controller that intends to transmit pulse information at precise periods of time. The internal block of the TMS320C50 DSP can be seen over-leaf in Figure 54. The DSP has a total of sixteen input output ports that have a data length of sixteen bits. The DSP also supports JTAG information, which allows the user to emulate the DSP in software, see Figure 53. These powerful features coupled with the relatively low cost makes the TMS320C50 DSP ideal as the basis for an embedded microprocessor system, that can be added too as the need arises.

The machine velocity control algorithm was coded in a mixture of C code and assembler as was needed. The DSP communication software was written in C code since this was not taken as being time critical, whilst the algorithm to generate the resultant velocity profile was coded via the use of the DSP assembler since it was time critical.

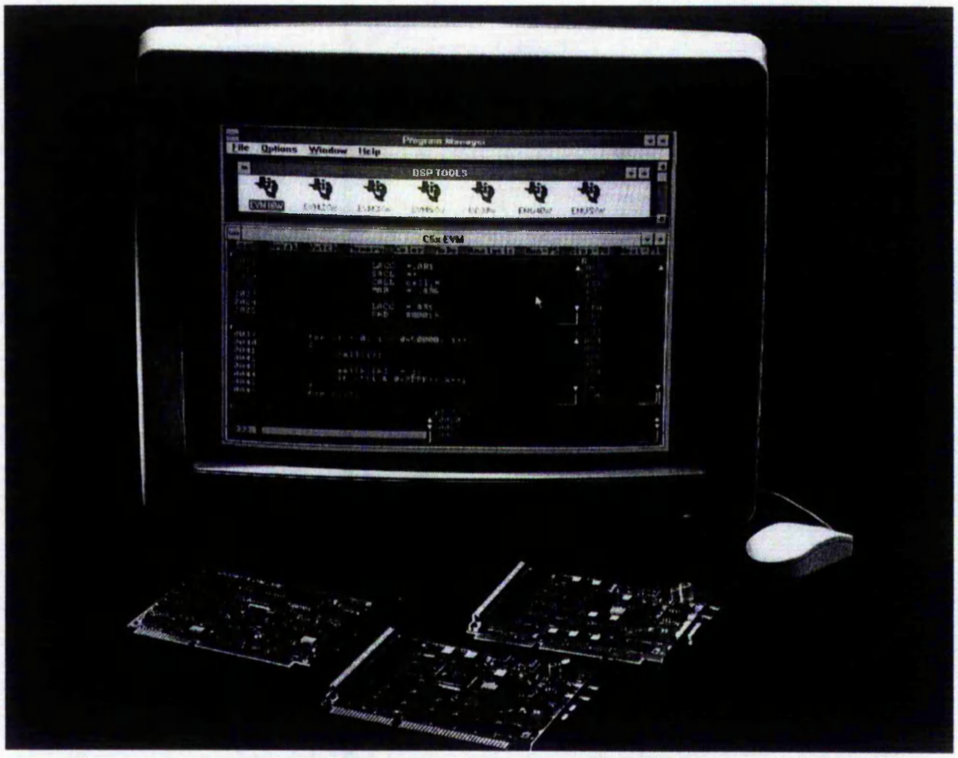


Figure 53 : Texas Instruments TMS320C50.

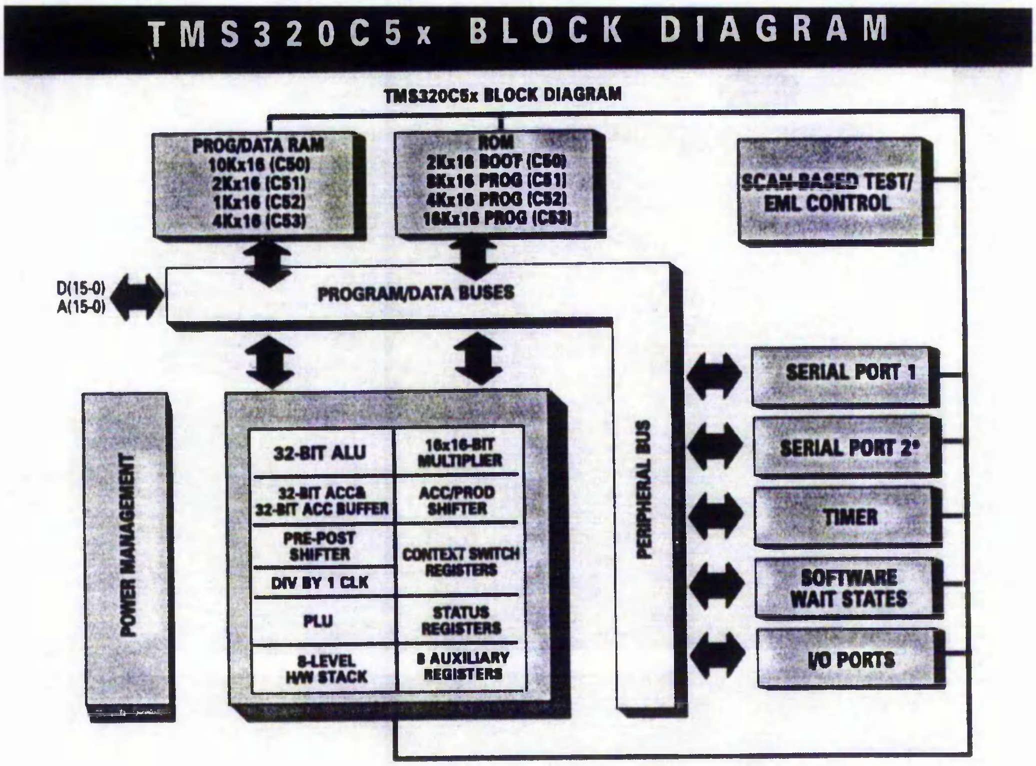


Figure 54 : Block Diagram of TMS320C50.

The velocity profiles are generated in the following manner: -

1, First the distance and direction are sent to the DSP.

2, The number of pulses that comprise the total distance, accelerating phase and decelerating phase are calculated.

3, The accelerating & decelerating phase is then generated, given the maximum velocity.

4, The first PFM signal is calculated and loaded into the timer, when the timer counts down to zero, the signal level of the output port is changed and the next value is loaded into the counter.

5, Once all the PFM signals have been generated the system returns to remain idle until another message is received.

The signals generated from the output port are pulse frequency modulated, where the frequency of the pulses changes in relation to the speed demand signal. The machine velocity control algorithm can generate one of three velocity profiles as decided by the user with a single key press. The distance, velocity, direction and acceleration can be changed manually from the keyboard or automatically from the software. The machine velocity control algorithm was developed to measure the scope for improvement in terms of resolution and accuracy that improved velocity shaping could provide. The following pages contain short discussions on the two velocity profiles found to be of relevance to the author's work.

4.2.1 LINEAR VELOCITY PROFILE

The linear velocity profile was designed along the lines of what most people view to be the typical velocity profile. The velocity profile used as a comparison against the interpolation synchronised velocity profile shown in Figure 33 (page 77).

The velocity profile has two stages of acceleration, almost identical in nature. The first stage is featured at the start of the velocity profile shown in Figure 55. This stage accelerates more rapidly than the second stage in an attempt to bypass the motors natural frequency. This is contradictory to various ideas of two stage velocity profiling that suggest accelerating slowly at first to overcome system inertia and then accelerate faster. The velocity profile shown in Figure 55 accelerates quickly at first since this is the region where it will have the highest torque. As the speed increases the available torque decreases, to match this condition the acceleration changes slightly such that the demand torque does not exceed the available motor torque. The velocity fluctuates at the two main changes of acceleration, the start of the velocity profile, and the end of the acceleration phase. This indicates that these areas need to be smoothed.

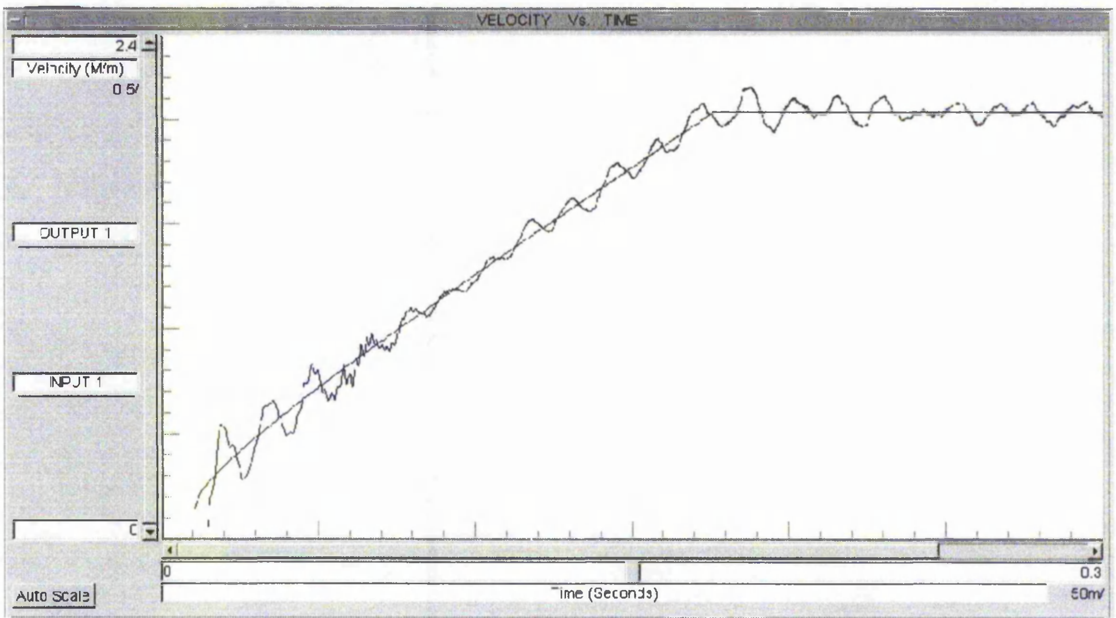


Figure 55 : Linear Velocity Profile.

4.2.2 PARABOLIC VELOCITY PROFILE.

The parabolic shaped velocity profile was designed based upon the work of Palmin[50]. The velocity profile is based on a parabolic curve expression can be readily changed to suit the machine in question, simply by adjusting the base acceleration. The parabolic shaped velocity profile can be seen in Figure 56. The velocity profile starts off at the standard set starting speed then as the motor begins to move the acceleration changes. The acceleration is low at first, then quickly becomes progressively higher until the motor has accelerated up to a speed approximately equal to 60-70% then the acceleration decreases until it becomes negligible.

The parabolic velocity profile represents a significant improvement over the velocity profile shown in chapter 3 , Figure 33. The resultant dynamic displacement error can be reduced by over 80% by utilising this velocity profile - down from 30 microns error to 5-6 microns. The remaining percentage is taken as being mechanical noise due to the off-centre pulleys.

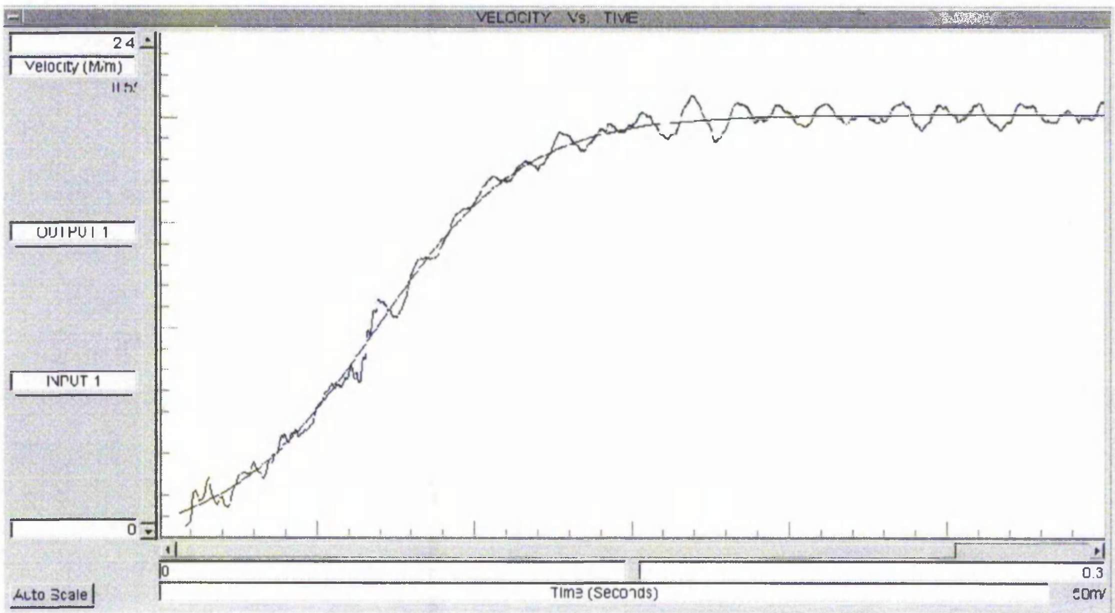


Figure 56 : Parabolic Velocity Profile.

4.3 Smoothing Motion

The process of generating smooth motion is essential to the requirements of multi-axis continuous path motion. It can be seen from Figures 55 & 56 that smoothing the input velocity profile results in less excitation of the dynamics. Velocity fluctuations are the result of this excitation, hence they are consequently reduced. In turn the machine does not accelerate/decelerate as sharply, which means the load torque is much lower. With a lower load torque the machine has more available reserve torque, from which greater machine performance can be achieved.

Unfortunately in multi-axis continuous path machinery the velocity is constrained by the process of interpolation as a consequence of synchronising distance with time. The author considered these points and decided that if an additional open-loop stage could smooth the velocity then both requirements would be met. The author tried various stages upon the simulator using the Simulink tool of Matlab. This simulation led to the idea and then development of the novel inverse velocity filter algorithm. The reader can see the results of adding this additional stage to the simulator in Figure 57. It can be seen that the resultant velocity profile is greatly smoothed, thus supporting the effectiveness of such an approach. The solution was implemented in hardware and is discussed within the next chapter.

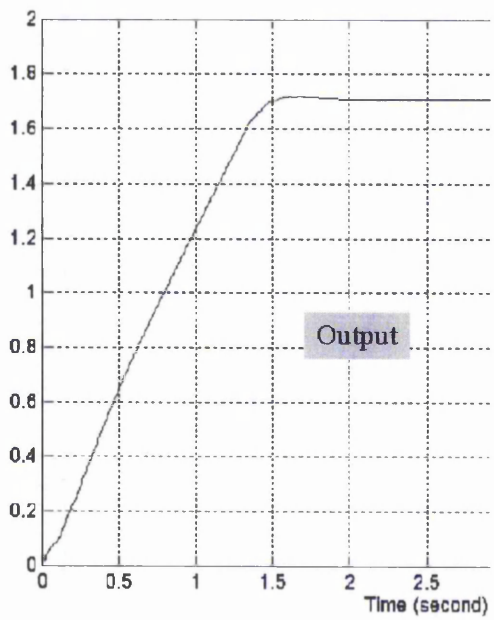


Figure 57 : Output Response from Simulator with Filter Stage.

Chapter Five

Novel Inverse Velocity Algorithm.

5. 0 Solutions for Smoothing Rough Motion.

To obtain optimum performance from multi-axis continuous path stepper motor driven machinery then it is clear that the resultant path must be described as accurately as possible by the path-planning and interpolation algorithms. It is also clear that this resultant motion should be as smooth as possible in order to minimise the displacement errors caused by the excitation of the dynamics, therefore achieving the highest utilisation of the available torque (chapters 2 & 3).

Considering the aspect of smooth motion then it can be seen from the earlier experimentation that there are many parameters that can effect the resultant velocity of multi-axis machinery. The author has shown through simulation (see chapter 4), that if the appropriate additional stage is added to the output of the controller then the input velocity to a given axis can be smoothed in real-time.

A number of options were considered on how to achieve this, two such examples are filtering and feedback control. The filter solution, shown in Figure 58, will essentially be a low-pass filter thereby smoothing the sharp changes of acceleration. If the changes of acceleration are smoothed then the excitation of the system dynamics will be lessened, which in turn means greater performance from the system.

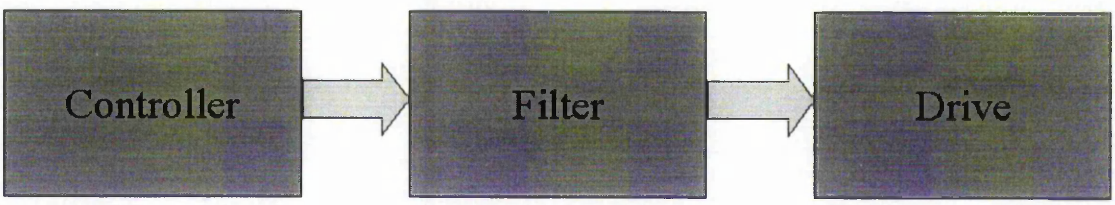


Figure 58 : Filter Solution

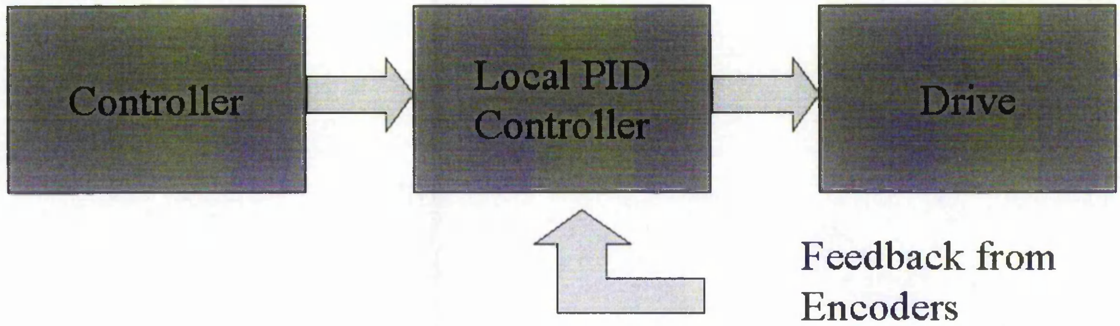


Figure 59 : Local PID Controller Solution

The feedback controller used in the simulation was a local PID controller, since this represents the typical controller used in classical control theory. The local PID controller solution, shown in Figure 59, coupled with the system as a whole, will behave in a similar fashion, provided the parameters are set to heavily damp the input signal. This is essentially an overdamped system. The velocity will be fed back to the PID forming a localised velocity control. Tuning such a system would be difficult as it would require a special profile that continually went up and down a given step length. The PID controller relies on the system and drives properties to provide active damping of the velocity, thereby smoothing any velocity fluctuations, and in turn increasing the performance of the system. Shown in Figures 58 and 59 are block diagrams showing how these methods can be applied to the existing controller. In the case of continuous path motion the control of velocity is not an independent process, the path also needs to be controlled and synchronised to the motion, see chapter 2. In order to gain smooth continuous path motion then the above methods need to be assessed not only in terms of smooth motion but also in terms of resultant path error. This leads to the consideration of other factors such as phase and quantisation errors.

In control terms, processing a signal may produce a phase shift, which normally depends upon the frequency of the signal. For the two solutions presented above, a uniform phase shift will manifest itself as a time delay and provided this time delay is uniform for all the axes then there will be no problem since there is no difference between the relationship of the two axes. However a non-uniform phase shift would mean that the velocity for each axis could be shifted by different amounts depending upon the current velocity of the axis, this could result in a loss of position, resulting in unacceptable path errors.

The quantisation of the input demand also presents problems. The signal is quantised in both the distance and time domain. In the distance domain then rounding as a result of quantisation errors may result in the addition or subtraction of pulses. For example if ten samples of value 2.9 are taken then the total number of pulses should equal 29 not 30 which would be the result due to rounding errors. In order to minimise path errors due to rounding, then the input signal should be quantised as little as possible. The time domain presents similar problems in terms of velocity, the resultant velocity may suddenly change and excite the dynamics, leading to dynamic displacement errors.

With both of the sampling methods discussed in chapter 3, the level of data quantisation is linked to the sampling frequency and the resolution of the encoder. When all the other options like interpolation, over-sampling and higher order derivative approximation are considered the choice becomes which method presents the most accurate data for the given application. The accuracy of the signal is dependent upon the level of quantisation of the signal. Since the sampling frequency is often restricted by bandwidth of the transmission medium, most sampling frequencies fall below the 1MHz range. For the pulse period sampling method then the effective sampling rate is equal to the system clock frequency provided the hardware is capable to handle the transmission of data at this speed. This means a lower level of quantisation is achieved when the pulse period method is used, in this case by up to 80 times lower than using the pulse counting method sampling at 500KHz. For example if we sample a 250 KHz signal at 500KHz then the quantisation error is equal to +/- ½ sample pulse, (2 micro seconds). However if we sample at 40 MHz (maximum clock frequency of data acquisition board) then the quantisation level for a signal is still +/- ½ sample pulse but the sample pulse length is now only 25 nano seconds, a magnitude of 80 times better. Therefore for our given application the pulse period method presents the best sampling method.

One aim of the research is that the solution should provide much of the benefits of feedback control but without the incurred cost of such a method. Feedback control is expensive in two ways. Firstly transducers have to be fitted to form the feedback element since most manufacturers of open-loop CNC machines do not fit encoders. Secondly the cost increases as additional components are added to the controller. Feedback control is typically used for one of two reasons, firstly the actuator is not open loop stable, and therefore information concerning its current state is needed or secondly it is used to remove the effect of external disturbances upon the system. The first reason is not particularly valid for stepper motors since they move in discrete steps and are open loop stable. Only the second reason offers the possibility of any benefits. In light machining operations, where the load is does not have sufficient mass to cause excess wear of the mechanical components, and machine tool chatter is not excessive then these disturbances are minimal. It is the author's belief that if the effects of the periodic disturbances, i.e. an off-centre pulley could be monitored then a way could be found to compensate for them whilst still operating in an open loop fashion. This would require routine monitoring for the instances where greater precision or performance is required, but for the current application these parameters do not vary greatly. In essence better control through better understanding, a hybrid control residing between open and closed loop control. A machine used for light machining could be instrumented to fully understand the range of disturbances that could result in the machine generating path errors. This data could be assessed and the controller could then be adjusted to minimise the significant causes of those errors.

The options were considered and found that the filtering method would yield the highest merits. The level of quantisation will be kept to a minimum by using the pulse period counting method of data capture.

5. 1 Description of Novel Inverse Velocity Algorithm.

In linear systems mathematical models can be expressed as either a Continuous or Discrete-Time system. Continuous-time system models can be considered as representations of analogue filters whilst discrete-time systems representations of digital filters. Since the author's field is concerned with digital control then the design of a digital filter, such as a finite-duration impulse response filter, (FIR filter) would seem to be the best solution. FIR filters have several benefits namely that they are stable, they can have linear phase characteristics and they can be efficiently realised in hardware[78].

A block diagram showing a FIR filter solution is shown in Figure 60. The input to the filter is data sampled at discrete time intervals. These data samples are applied to coefficients respective to the present time and the time they arrived. The products of the data and coefficients are summed and then divided by the total number of samples to form an appropriate filtering effect over the data. If the coefficients are all one then the filter acts as a moving average filter. A moving average filter behaves in the same manner as a low-pass filter when applied to discrete time data.

Conventionally FIR filters rely on the input data to be sampled at discrete time intervals. Since the quantisation level needs to be minimised and the only way of currently achieving this is to use the pulse period method of sampling then additional circuitry would be needed before and after the filter to convert unit distance into unit time and vice versa. This is inefficient and the signal will be significantly degraded resulting in path errors due to quantisation and rounding. An alternative approach put forward by the author is that of the inverse velocity filter. Essentially the FIR filter operates on time based data, but a filter using distance based data would be equally valid, albeit its characteristics would vary with speed. Unit time and unit distance measurements are not directly interchangeable due to reciprocal nature between the two so although both are filters, they are conceptually different. These differences can be seen in Figures 60 and 61 where instead of measuring discrete-time information the filter accepts discrete-distance data.

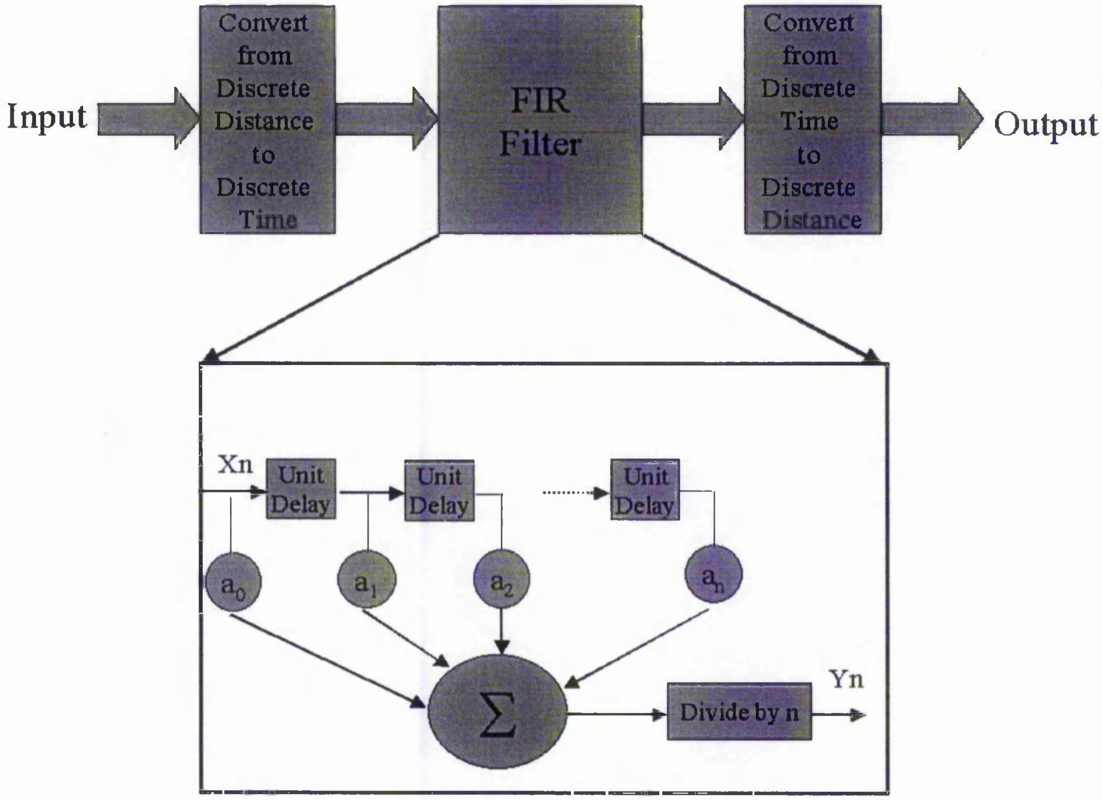
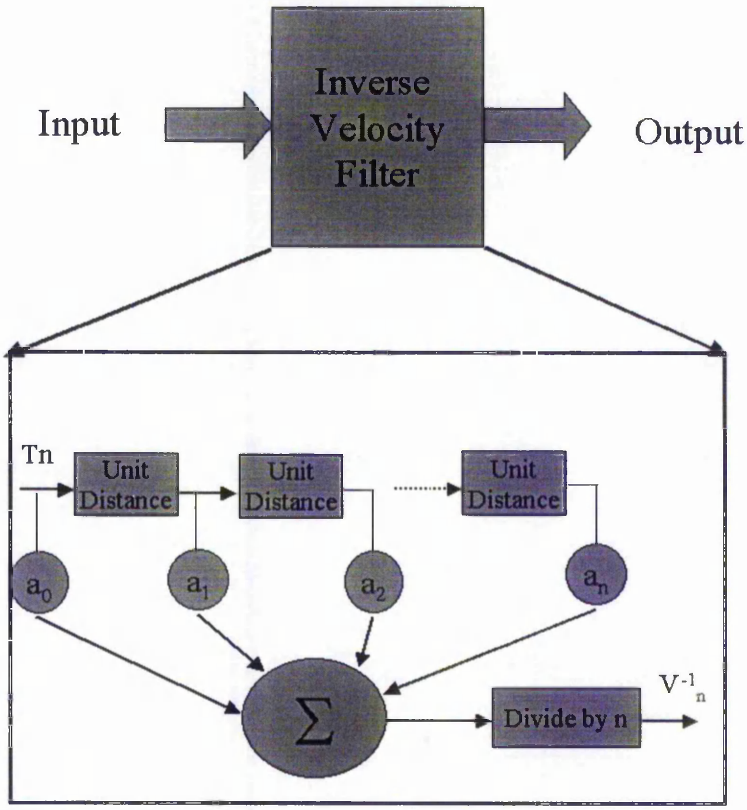


Figure 60 : FIR Filter.



Tn = Time at distance n.
V^{-1}_n = Inverse Velocity at distance n

Figure 61 : Inverse Velocity Filter.

The operation of the inverse velocity filter is similar to the operation of the FIR filter. First the data is collected on a pulse by pulse level, applied to the relevant coefficient and summed. This figure is then divided by the total number of samples, this forms the time period of output pulse. Since the filter is affecting change to the time period of the output pulse then the filter is effectively not filtering velocity but its inverse hence the name inverse velocity filter. The inverse velocity has the benefit of real-time operation & low quantisation of input data. The disadvantage with this technique is that the characteristics of the filter will vary with velocity, but this could be compensated for if the characteristics of the filter were changed at fixed velocity levels. In implementation a span of sixteen points were chosen. The reason for this figure is based upon the number of stepper motor pulses that form the coarse ramp steps, being between 10-20 in number. In mathematical terms then dividing by 16 can easily be achieved in binary if the number is shifted to the right 4 times. The characteristics can change to include more points such as 32 or 64 which produces too much time delay at low frequencies but would suit the high speed or high frequency ranges.

5.2 Mathematical Description of Inverse Velocity Filter

In the field of engineering one way of expressing information is in the form of a mathematical assessment since it forms a concise description of the process or problem area. The main point to consider when assessing the inverse velocity filter is that the filter itself behaves as any other digital filter as shown in Figures 60 & 61, i.e. the mathematical processes upon the data remain the same. The difference between the techniques becomes a matter of correct interpretation of the data. For the typical FIR filter the data will be discrete time so the typical FIR filter will be represented by the following mathematical formula.

$$Y_t = \frac{a_0 X_t + a_1 X_{t-1} + a_2 X_{t-2} \ldots\ldots + a_n X_{t-(n-1)}}{n} \quad - \quad (5.0)$$

Where Y_t = Current Filtered Output.

a_x = Constant depending on filter algorithm.

X_t = Input value at time (t).

t = Time.

n = Number of Filter points.

Therefore the mathematical representation for the inverse velocity filter can be represented by the following formula. Since the inverse velocity is fed in and the output is the inverse velocity then the reciprocal has been removed from both sides to aid clarity.

$$Vo_d = \frac{a_0 vi_d + a_1 vi_{d-1} + a_2 vi_{d-2} \ldots\ldots + a_n vi_{d-(n-1)}}{n} \quad - \quad (5.1)$$

Where Vo_d = Current Filtered Output Velocity.

a_x = Constant depending on filter algorithm.

$\mathbf{Vi}_d$ = Input Velocity at distance (d).

d = distance.

n = Number of Filter points.

5.3 Implementation & Results.

The aim of the research is to investigate the cause of performance limitations in multi-axis continuous path stepper motor driven machinery. The reader can see from chapter 3 how micro-stepping, low gear ratios and reduced current levels can be employed to significantly improve the performance of multi-axis machinery. However these methods share the same the disadvantage that the peak velocity or acceleration is restrained. The concept of the inverse velocity filter is that by smoothing the velocity on the slave axis of a multi-axis CNC machine then any performance degradation due to the excitation of the dynamics will be sufficiently reduced, without any velocity/acceleration restraint. The filter will smooth the velocity profile by performing a moving average over the inverse of the velocity. A moving average filter means that all the coefficients are set to one, thus eliminating the multiply operation, thereby increasing the speed of operation. Several ways of weighting the coefficients were tried since it made sense to weigh the coefficients such that the current input value has more effect upon the output than the very end measurement. In practice however it was found that these weightings made very little significance over the fact that the velocity was smoothed.

The solution was implemented in software within a DSP. The DSP is interfaced to the digital acquisition board developed by the author. This board provides both the acquisitions of the input data stream as well as the circuitry for the generation of the output pulses. The buffering of the input signal is provided in the form of FIFO's aboard the data acquisition system. The author decided to contrast the output of the filter against the benefits that the methods suggested in chapter 3 would yield. The reader can see from Figure 62 the input waveform to the machine with the drive current reduced, smaller velocity steps and a low gear ratio of 3:2, essentially the optimum you could hope for without the filter solution. This is contrasted with Figure 63 showing the filtered input velocity profile & resultant motion. These velocity profiles show the extent of improvement the research has yielded when contrasted against the velocity profiles shown in chapter 3. Since the filtered solution is compared against the 'optimum' unfiltered velocity profile the benefits are hardly visible to the eye, therefore a mathematical analysis technique had to be applied. Standard deviation is an accepted way of assessing the noise upon a signal. The standard deviation in velocity error was taken for Figures 62 & 63 to demonstrate the benefits of the filter. The standard deviation for Figure 62 is 351 mm/minute & for the filtered velocity profile in Figure 63 is 178 mm/minute.

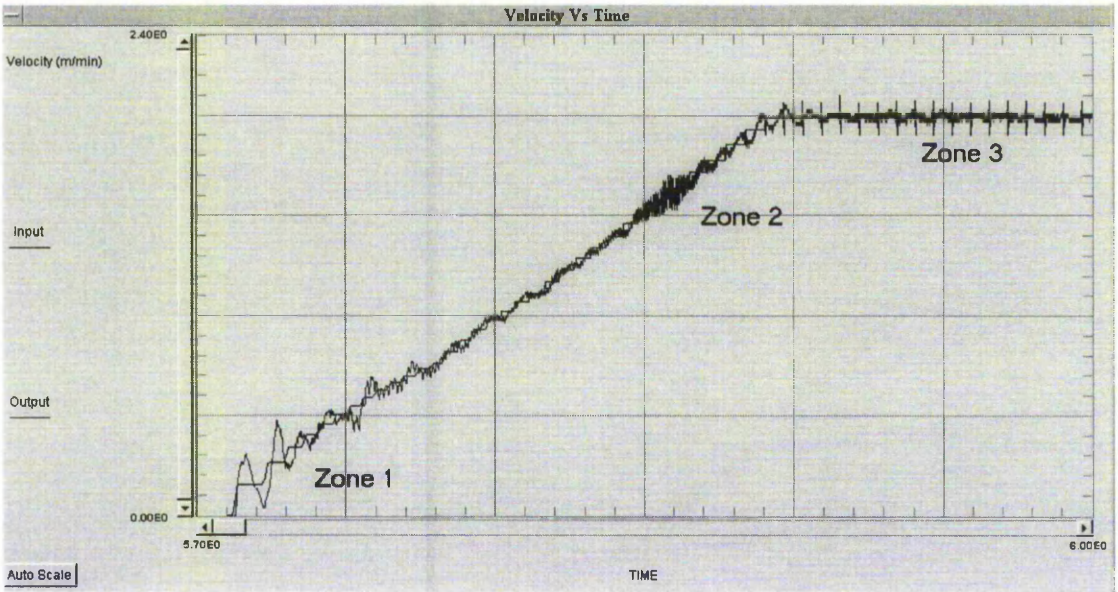


Figure 62 : Velocity Profile without Filter.
(Velocity = 2m/min)

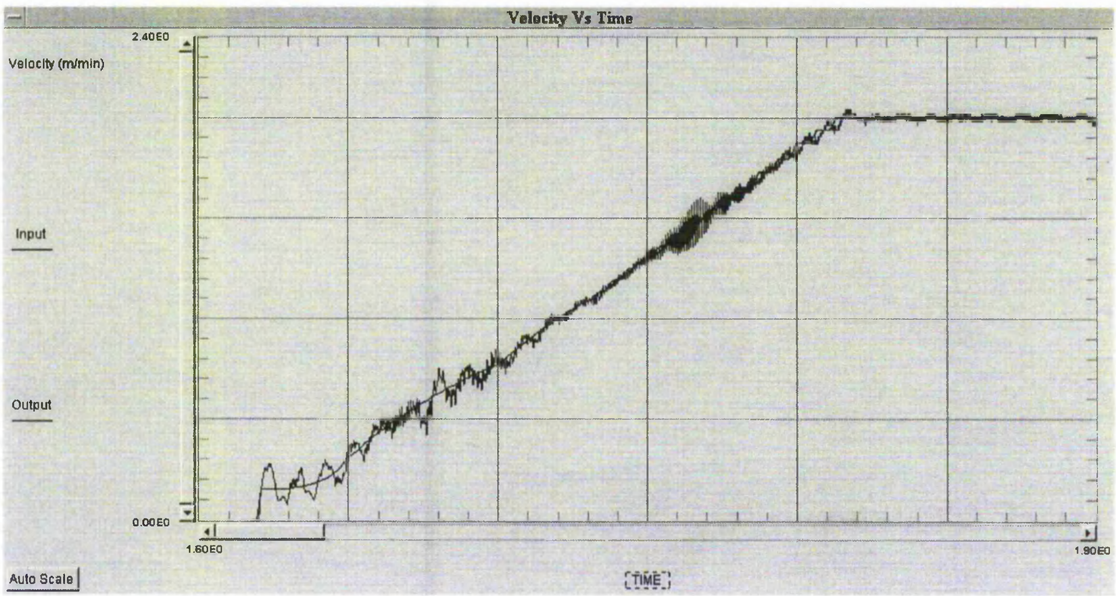


Figure 63 : Velocity Profile with Inverse Velocity Filter.
(Velocity = 2m/min)

The reader can see that in zone 1 both the filtered and unfiltered velocity profiles suffer from the initial start-up velocity step. The difference between the two is that the filtered solution smoothes the second and subsequent impulses, giving the motor time to recover, whilst for the unfiltered velocity profile, the second impulse simply causes the dynamics to excite further. The reader can note that for the region indicated as zone 3 that the sharp spikes caused by interrupt latency on the controller are smoothed. The decrease in standard deviation clearly indicates the benefit the filter has upon the resultant velocity of the system for a standard velocity profile. However the reader can see from zone 2 that by solving one problem often another is generated. In this case the smooth velocity profile means that when the pulse frequency matches one of the two natural frequencies of the stepper motor, the resonance vibrations may be larger due to the longer duration spent operating around this frequency. To measure the effectiveness of the filter under continuous path conditions then an appropriate shape was selected. The reader can see this shape from Figure 64, showing the visual front end of the computer aided manufacturing software produced by Axiomatic Ltd for use with the Pacer systems CNC machinery.

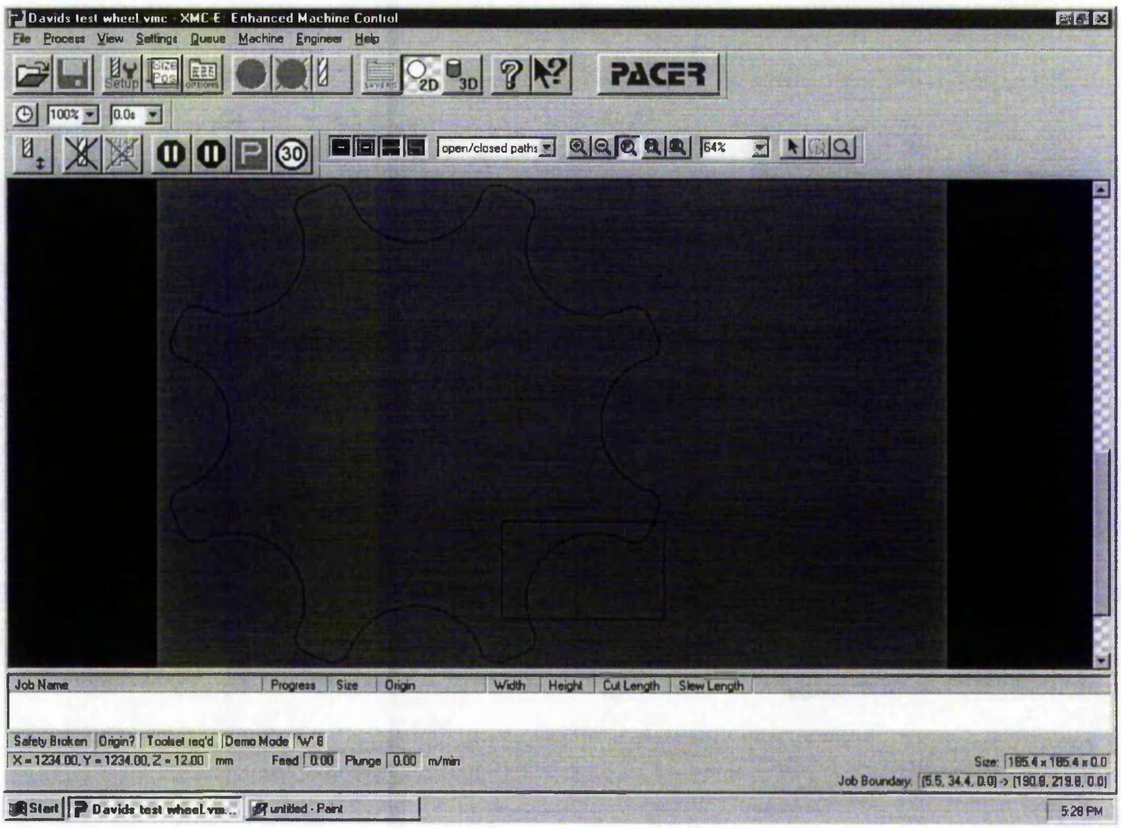


Figure 64 : Visual Front End of XMC-E.

The drawing represents a bottle separation gear used to separate bottles for drinks bottling companies. An engineering company using a Pacer Cadet machine produces these gears. Although the shape does not seem complex, it does represent the common problems associated with continuous path systems. The shape has a long low curvature arc followed immediately by a short high curvature arc. These arcs are composed of short straight line segments, which also presents difficulty on how the shape is to be treated, either as an arc or as short lines [2].

The velocity profiles for both the x & y axis where captured for the area shown in the box section, these can be seen in Figure 65. To provide clarity the output response is not shown.

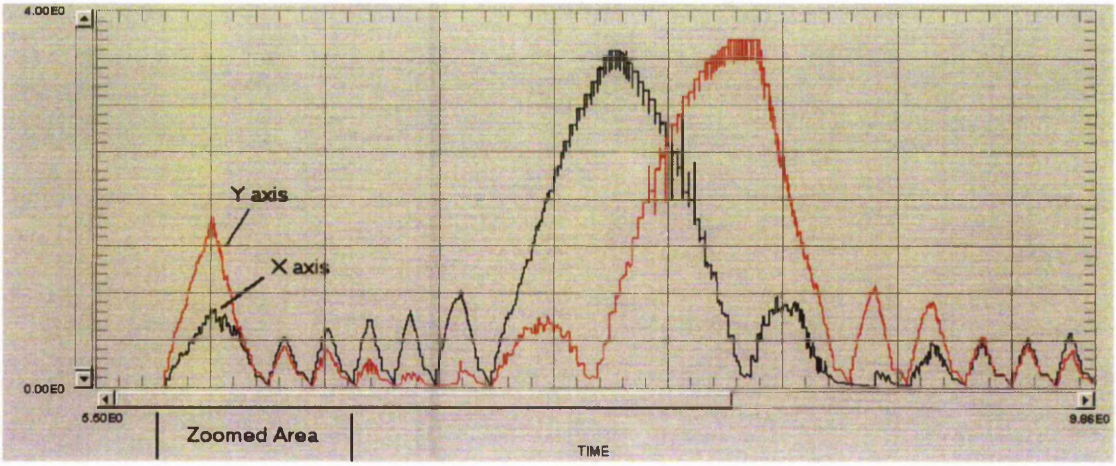


Figure 65 : Velocity Profile for Bottle Gear.

Initially the first vector is predominately in the y direction so the x-axis becomes the slave axis. As the machine moves around the small corner the master and slave axis change such that the y-axis becomes the slave axis. A section was taken and magnified to show the coarse velocity on the slave axis, this can be seen in Figure 66. Again only the input demand is shown to provide clarity. The filter was then applied to the system to smooth the x-axis, The smoothed velocity profile can be seen in Figure 67. The y-axis of the graphs show a velocity span of 0-0.8 metres per min,(0.1m/mm per div) whilst the x-axis represents a span of 0.84 seconds

The reader can see that the velocity profile has been considerably smoothed, therefore the dynamics will be less excited, enabling a higher achievable level of performance.

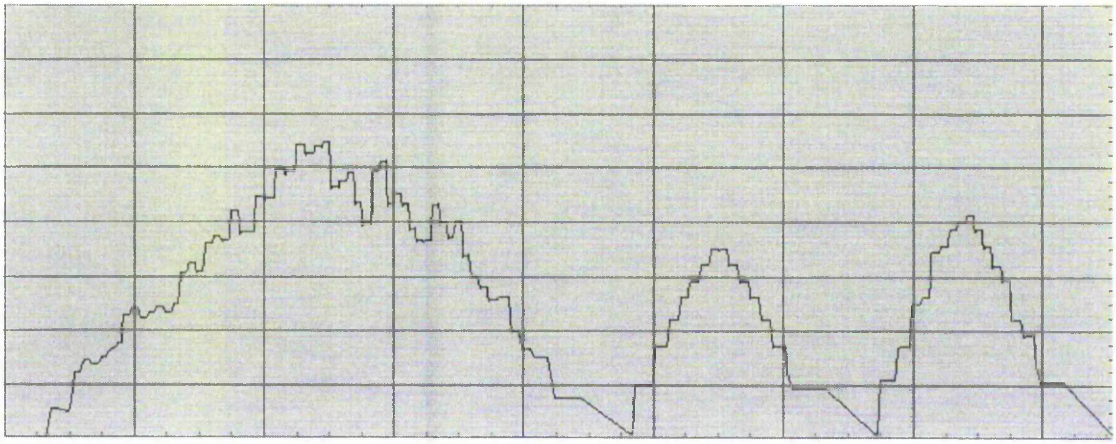


Figure 66 : Zoomed Area of Gear Velocity Profile.

(Y axis – 0.1m/min/div, X – 0.1 seconds/div)

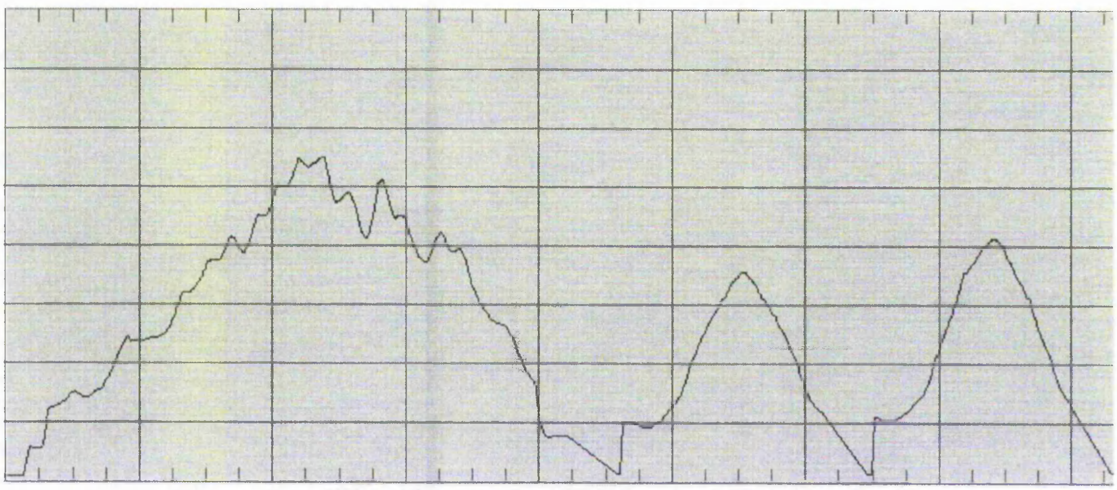


Figure 67 : Filtered Input Demand for Gear Velocity Profile.

(Y axis – 0.1m/min/div, X – 0.1 seconds/div)

5.3 Summary.

This section describes how a rough velocity profile can be smoothed by the use of a novel filtering technique. The novelty of the technique arises from the fact that the input to the inverse velocity filter is not discrete time but discrete distance data. This concept changes the properties of a standard FIR filter to the extent that the resultant filter is a different technique altogether. It can be seen from the practical results presented in the form of velocity profiles, that the algorithm does smooth the velocity profile as expected from the simulation presented in chapter 4.

The disadvantage with this filtering approach is that the filter characteristics change in proportion to speed. If both axes are traversing at the same speed then there is no problem since both signals are delayed by the same amount. However if they are traversing at different speeds then they will be delayed by different amounts. This difference in delay will result in path errors. In practice the author found, as expected that increasing the number of data points increased the amount of phase delay. It was found that using 16 data points filtered the velocity profile significantly whilst only imparting a minimal phase delay. The question arises whether the path error caused by the phase delay is still below the previous levels caused by rough motion.

There are numerous ways in which to measure error upon a signal, the standard deviation method was chosen due to its wide acceptance as an error measurement. The best method to evaluate the filter is in terms of the increase in performance and the decrease in path errors. A measure of the performance was made and plotted in the form of acceleration vs. velocity graph, see Figure 68. The improved velocity profile shown in Figure 62 was used as the base comparison vs. the filtered velocity profile shown in Figure 63. For each of the three speed levels the acceleration was slowly increased until the machine stalled. The tests were repeated several times to avoid discrepancies.

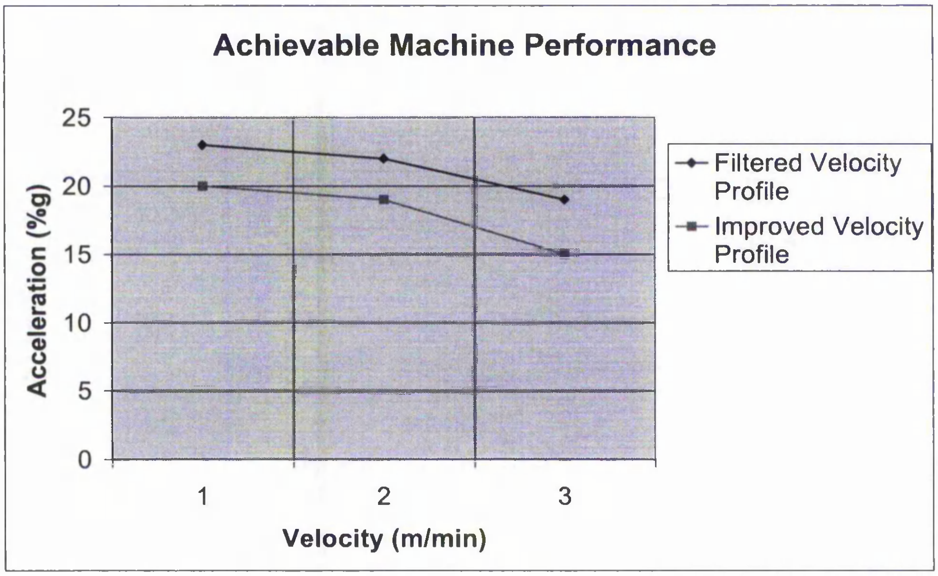


Figure 68 : Achievable Machine Performance Comparison.

The graph clearly shows an increase in the amount of acceleration that the machine can be subjected to. This graph is not a true performance graph however since the y-axis is acceleration not torque, if the load torque is known precisely then this axis can be converted into torque. Due to the additional load being reflected back to the stepper motor caused in part by velocity fluctuations, the practical load torque is far greater than the calculated value. A more formal method of assessment needs to be found in order to gauge the effectiveness of the solution. The author turned to the work of Palmin[50] and Acarnley[27] concerning measuring the torque capability of stepper motors. This work is detailed within the next chapter.

Chapter Six

Assessment of the Algorithm via the Measurement of Dynamic Displacement Errors.

6. 0 Dynamic Displacement Errors

In continuous path systems when the path is interpolated into chord segments then any difference between the desired path and the interpolated path can be expressed as displacement errors. The author calls these errors static displacement errors since they are independent from any CNC machine and are caused by the interpolation/path-planning algorithm. The author uses the term dynamic displacement error to symbolise any additional displacement errors that are caused predominantly by the excitation of machine dynamics whilst the path is being machined.

The excitation of the dynamics cause a limitation in stepper motor driven machinery by causing a reflected load to be placed upon the motor. Considering Newton's first law, if the dynamics are excited then it can be seen that the velocity fluctuates, with increased levels of acceleration. This increased acceleration has the same effect as adding greater mass, the equivalent force required to move the machine is increased, reducing the range in which the machine can be operated within. Therefore an assessment of the machines performance can be derived from the measured dynamic displacement errors.

Acarnley [27] describes how external load torque, perhaps caused by friction, give rise to a small error in position when the motor is stationary. The motor must develop enough torque to balance the load torque and the rotor is therefore displaced by a small angle from the expected step angle. The resultant 'static position error' depends upon the external torque, but is independent of the number of steps previously executed, i. e. the position error is non-cumulative.

Acarnley presents the case that by knowing the load torque, peak static and static torque/rotor position characteristic then the static position error can be calculated. The load torque however is a theoretical value, which only considers the construction of the stepper motor and machine it fails to take into account any dynamic behaviour of the components. The author builds upon the work of Acarnley by showing how the true dynamic load torque can be derived from measuring the position error under dynamic conditions.

6.1 Development of Dynamic Displacement Error Measurement Technique.

Manufacturers generally supply information about the torque producing capability of a stepper motor in the form of a static torque vs. rotor position graph, showing the torque developed by the motor as a function of rotor position for several values of winding current [27]. A typical set of curves presented by Acarnley is shown in Figure 69.

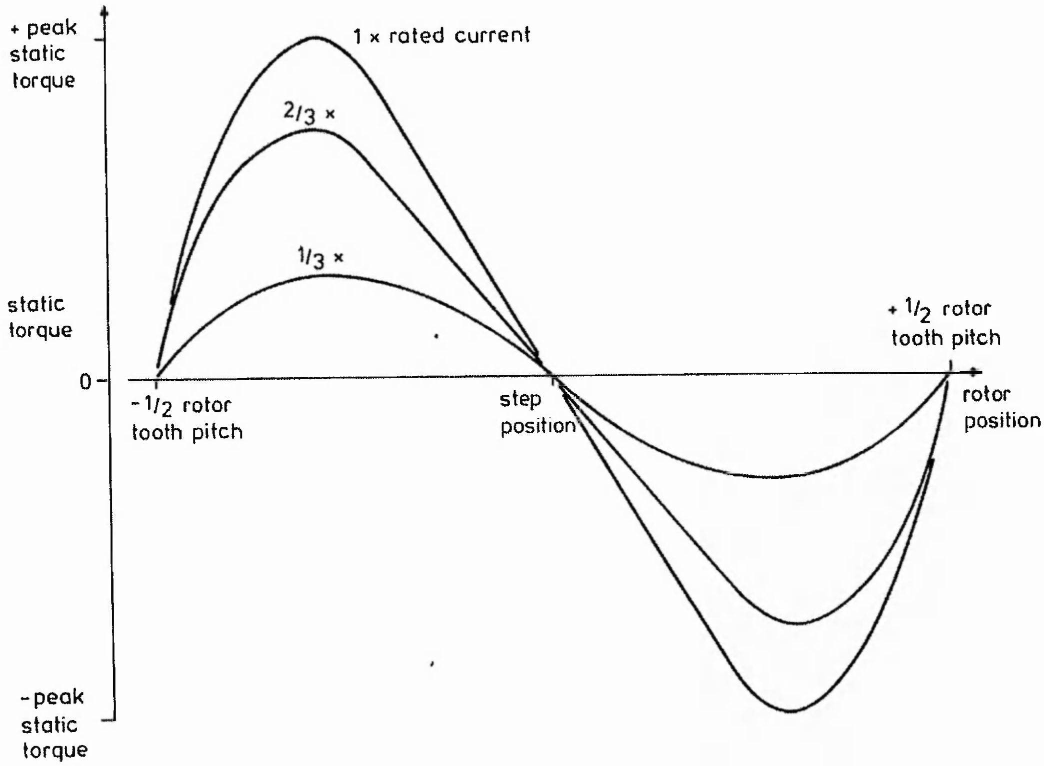


Figure 69 : Static Torque/Rotor Position Characteristic at Various Phase Currents. (Taken from Acarnley)

At the step position the appropriate sets of rotor and stator teeth are completely aligned and the motor produces no torque. If the rotor is slightly displaced from the step position a force is developed between the stator and rotor teeth (Harris et al. [79]) giving a torque which tends to return the rotor to the step position. A rotor displacement in the negative direction produces a positive torque and a positive displacement results in a negative torque. The static torque /rotor position characteristic repeats with a wavelength of one rotor tooth pitch, so the rotor only returns to the correct step position if it is not displaced by more than half a rotor tooth pitch. For hybrid motors used throughout this study this corresponds to two full steps or four half steps. For larger displacements the rotor and stator teeth become aligned in stable equilibrium at a distance which is a multiple of the rotor tooth pitch from the required step position [27]. This can be seen in Figure 70.

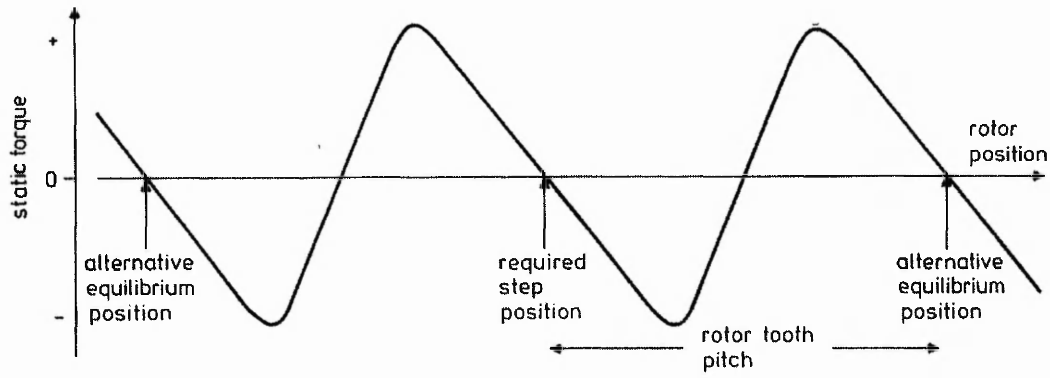


Figure 70 : Static Torque/Rotor Position Characteristic over Several Rotor Positions. (Taken from Acarnley)

The shape of the static torque/rotor position characteristic depends on the dimensions of the stator and rotor teeth, as well as the operating current. Prediction of the characteristic from the internal geometry of the motor is a complex electromagnetic problem (Ertan et al. [80]) which from the author and Acarnley's point of view can be safely left with the motor designer. The maximum value of static torque is known as the 'peak static torque'. It is often quoted as a single value corresponding to the rated phase current, even though it is a function of the phase current.

If an external torque is applied to the motor then the rotor must adopt a position at which the motor produces sufficient torque to balance the load torque and maintain equilibrium. The maximum torque which the motor can produce, and therefore the maximum load which can be applied under static conditions is equal to the peak static torque [27]. If the load exceeds the peak static torque the motor cannot hold the load at the position demanded by the phase excitation.

A load Torque produces a static position error, which can be deduced directly from the static torque/rotor position characteristic. An example taken from Acarnley can be seen in Figure 71 which shows the characteristic for a permanent magnet motor with eight rotor teeth and a peak static torque of 1.2 Nm at the rated phase current. With a load of 0.75 Nm the motor must move approximately 8 degrees from the step position, until the torque developed balances the load.

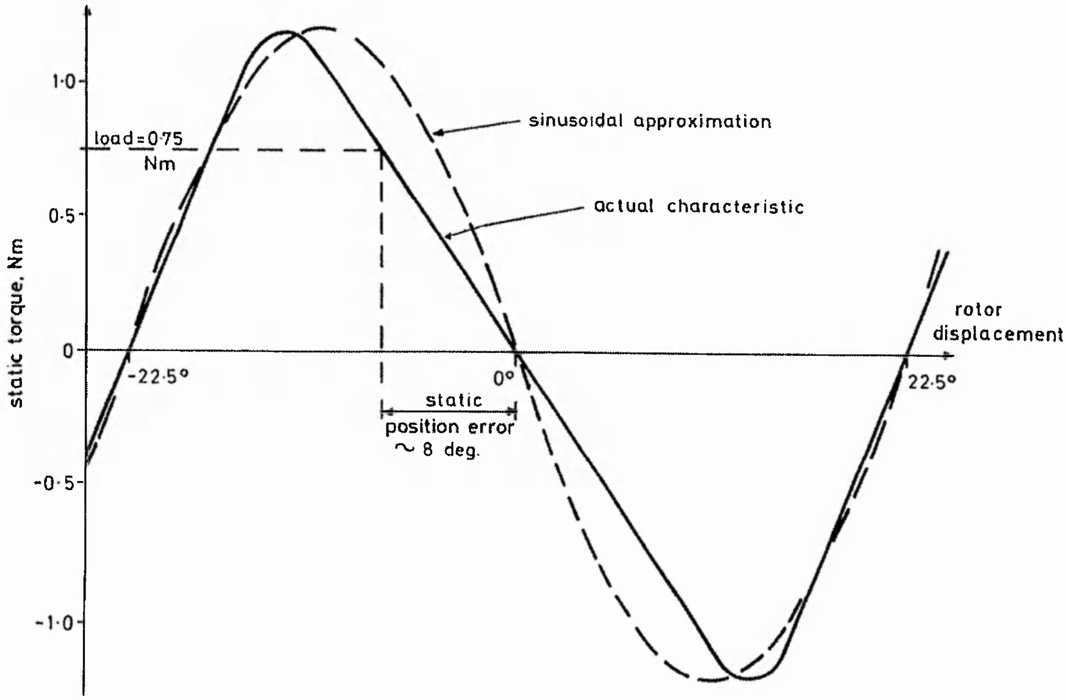


Figure 71 : Derivation of the Static Position Error from the Static Torque Characteristic. (Permanent magnet Motor example Taken from Acarnley)

The same theory applies to hybrid stepper motors except that the period would be 1.8 degrees rather than the period of 45 degrees shown in the example. Acarnley states that an estimate of the static position error can be obtained if the static torque/rotor position characteristic is approximated by a sinusoid. For a motor with p rotor teeth and a peak static torque T_{PK} at a rotor displacement θ from the step position, the torque developed by the motor is approximately:

$$T = -T_{PK} Sin(p\theta) \tag{6-1}$$

When a load T_L is applied the rotor is displaced from the demanded position by the angle θ_e, at which the load and motor torque's are equal:

$$T_L = T = -T_{PK} Sin(p\theta_e) \tag{6-2}$$

and the static position error is :

$$\theta_e = \frac{Sin^{-1}(-T_L / T_{PK})}{p} \tag{6-3}$$

Therefore increasing the peak static torque, either by choosing another motor or by using a different excitation scheme can reduce the static position error. Increasing the number of rotor teeth would also be effective in reducing the static position error, but for hybrid motors this is typically 50.

If we consider the case of position error whilst the machine is moving, i. e. dynamic rather than static then the error increases in proportion to the torque required to accelerate the inertial load. See Figure 72. If this required torque exceeds the peak static torque then the stepper motor will stall. It is proposed that the additional position error can be measured using a suitably high-resolution encoder, in our case 5000 poles per revolution. The purpose of the encoder is to measure the change in rotor position as the machine is accelerated and decelerated, thus to remove any chance of backlash the encoder should be fitted directly to the stepper motor, See Figure 73.

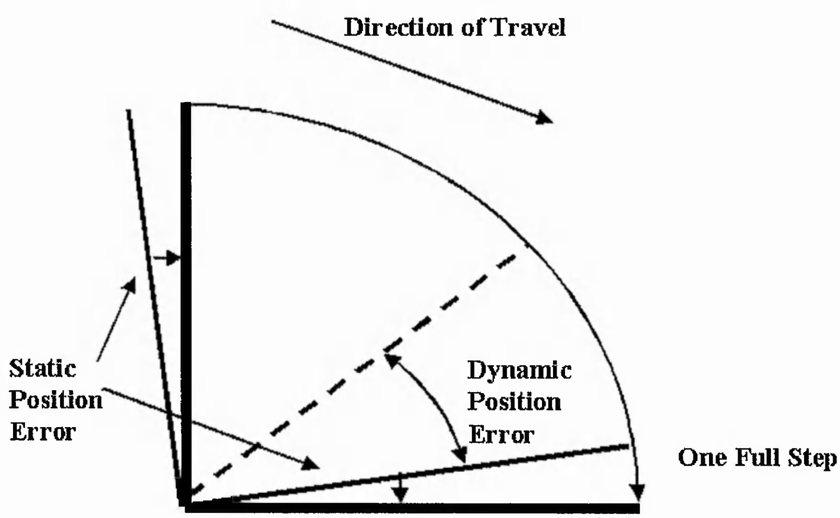


Figure 72 : Determining Dynamic Position Error.

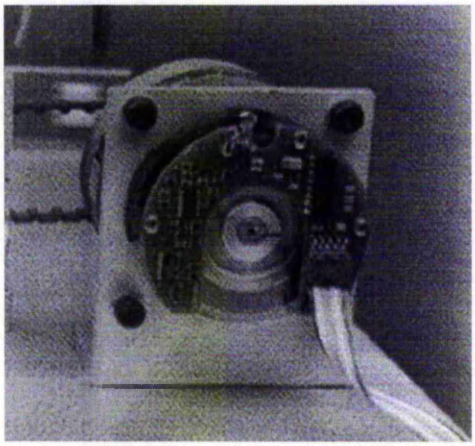


Figure 73 : Encoder used to Prototype DDE Technique.

The change in position error will reflect the additional load placed upon the stepper motor whilst the machine is moving. The lower limit will be the static position error whilst the maximum error will be the same for stalling conditions, i. e. two full stepper motor steps. The true position error can be found by adding the dynamic position error to the static position error. The static position error can be calculated as presented before or it can be measured by moving backwards and forwards. When the stepper motor changes direction it must travel twice the static position error to settle back into an equilibrium position representing the opposing load torque. It is proposed that the static position error can be determined by moving backwards and forwards, and dividing by two the additional distance moved by the stepper motor, see Figure 74.

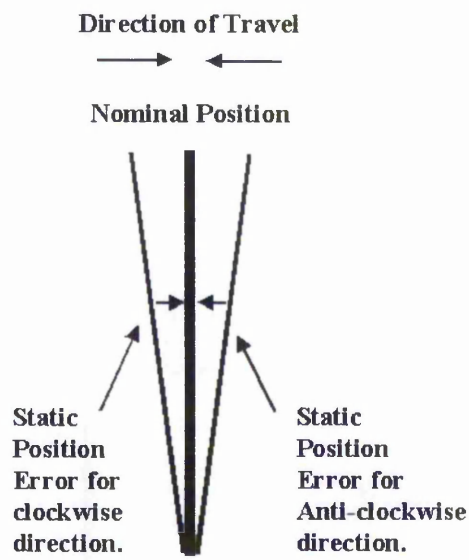


Figure 74 : Determining Static Position Error Through the use of Encoders

The load torque can then be determined by reading the corresponding torque level from the static torque/rotor position characteristic or by applying Acarnley's approximation, equation 6-1 and using the true position error figure. The load torque determined is now an instantaneous value since it is dependant the dynamic position error for the given step in question. These instantaneous values can be plotted and can form the basis of stall detection and correction measures.

6.2 Performance Measurement derived from the Measurement of Dynamic Displacement Errors

If the combined static & dynamic displacement error is plotted against each step that is taken (distance error vs distance travelled), then the resultant path error caused by rough motion can be determined. See Figure 75, The Y axis represents distance error against the X axis representing distance traversed. The distance error is expressed in terms of the number of full-steps that the stepper motor lags behind at that instant. It can be seen in Figure 75 that by the end of the acceleration period that the lag is equal to one full step or two half-steps. From the path planning and interpolation point of view the maximum interpolation error is typically ½ of a unit step be it a half or full step. The reader can see from Figure 75 that the dynamic position error causes path inaccuracies exceeding 4 times this level, up to 2 full steps. Once the inertia required to accelerate the load is taken away the reader can see that the torque of the motor tries to return the rotor back to the nominal level. The frictional components of the machine coupled with the current velocity also create an opposing force, causing a constant lag in position whilst traversing at that particular velocity. The superimposed sinusoid under inspection was found to be caused by the encoder disc not being concentric to the motor shaft. This concentric error was measured with a clock gauge and from the experimental results proved to match the error shown in the graphs below.

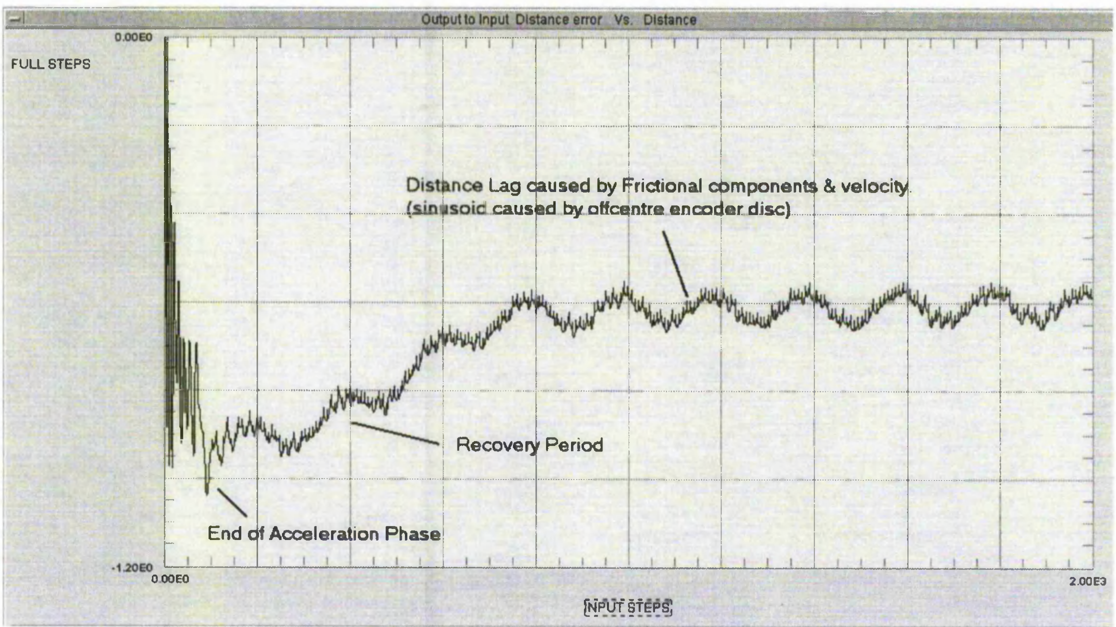


Figure 75 : Position Error vs. Distance Travelled.

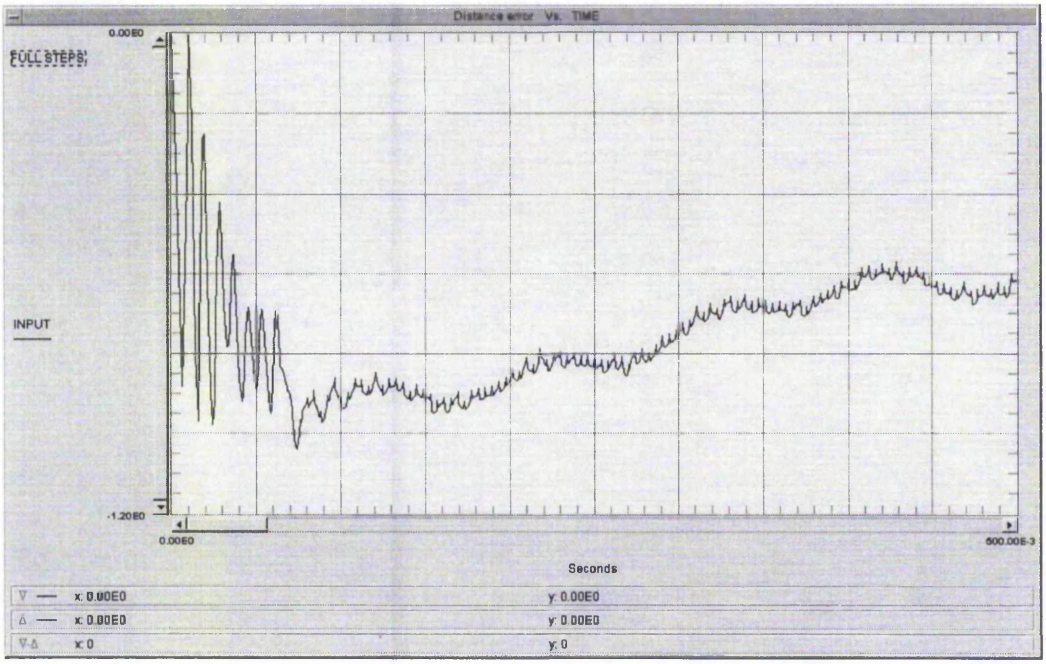


Figure 76 : Expanded View of Position Error vs. Distance Travelled

The reader can see from Figure 76 an expanded view of the graph shown in Figure 75. In Figure 76 the fluctuations in distance caused by the machine whilst accelerating are now more visible. The reader can see how the velocity fluctuations caused by the interaction of interpolation and acceleration cause displacement errors of nearly two motion steps (2 half steps = one full step), even when traversing at relatively slow speeds. Ideally the error during the acceleration phase should gradually increase as the machine goes through the acceleration phase, what the fluctuations show clearly is that during the acceleration phase the dynamics are excited.

If data is captured whilst the machine is traversed at three different speeds and the results overlaid then the reader can see the effect of the frictional lag discussed previously. See Figure 77. The reader can see that in light machining where the machine mass is not great that positional errors caused by frictional components form a major part of the overall position error. The Machine was traversed at 1, 2 & 3 metres per minute, with ever increasing levels of frictional position error.

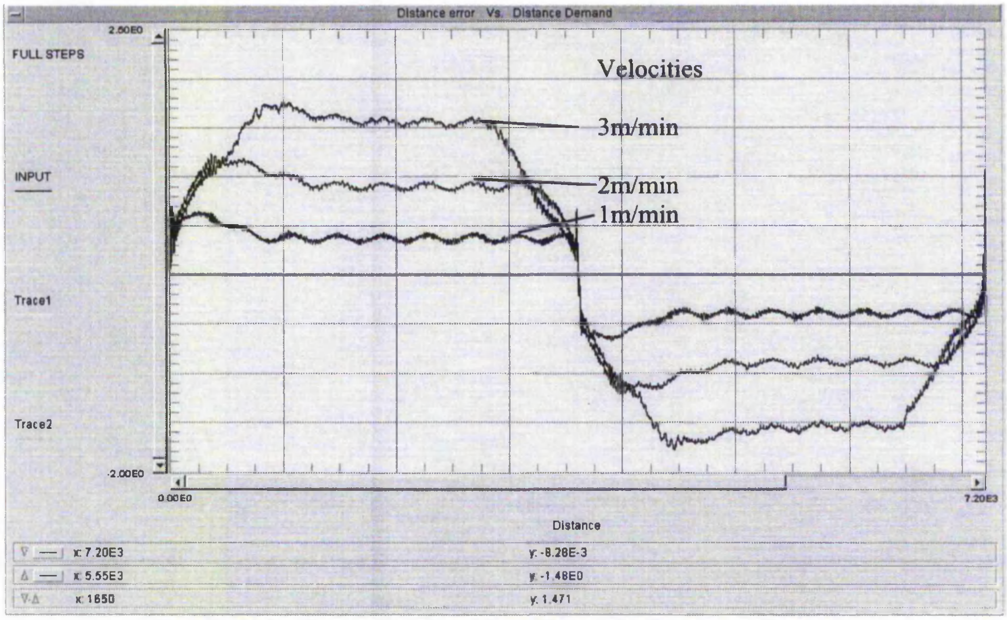


Figure 77 : Dynamic Position Error for Various Speeds.

If the acceleration is raised and the machine is traversed such that stalling may occur, then the maximum dynamic position error is reached. The author achieved this state then reduced the acceleration down by 1 percent to measure the dynamic position error. The reader can see from Figure 78 the dynamic position error caused by accelerating the machine at 19 percent of G, at a velocity of 3 metres per minute. The dynamic position error reads as 1.98 full-steps. When the machine is accelerated at 20 percentage of G then the machine will exceed 2 full steps and it will stall.

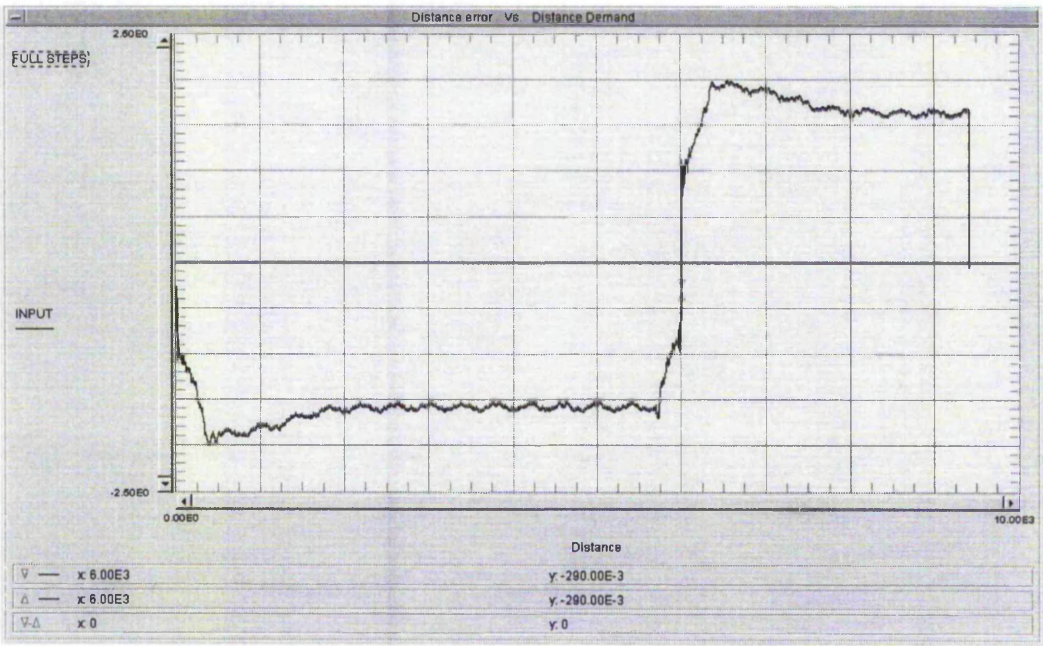


Figure 78 : Dynamic Position Error for Near Stall Conditions.

6.3 Effectiveness of Filtering Solution to the field of Smooth Continuous Path Motion.

From the dynamic position error graphs it is possible to measure the effectiveness of the filtering solution by contrasting the position error caused with and without the filter. To demonstrate the successfulness of the research a typical velocity profile that would have been generated by the controller at the start of the research was contrasted against the filtered velocity profile. The encoder pulleys were aligned to be concentric to the shaft & the machine was traversed along the same stretch of lead screw in the same direction to avoid inaccuracies. The results can be seen in Figure 79 & 80, which show an expanded view of positional error caused by the acceleration phase. The scale of the Y-axis is equal to ¼ of a full motor step. The reader can clearly see the fluctuating positional errors caused by the rough motion, and how by smoothing the motion that the error has decreased down to the expected theoretical level, thereby improving the performance of the machine considerably. The fluctuating distance error means that the stepper motor is pulled out of position and stalls before the expected level, by smoothing the velocity the displacement error decreases meaning the machine can be traversed at greater speed or acceleration before it reaches the stall region.

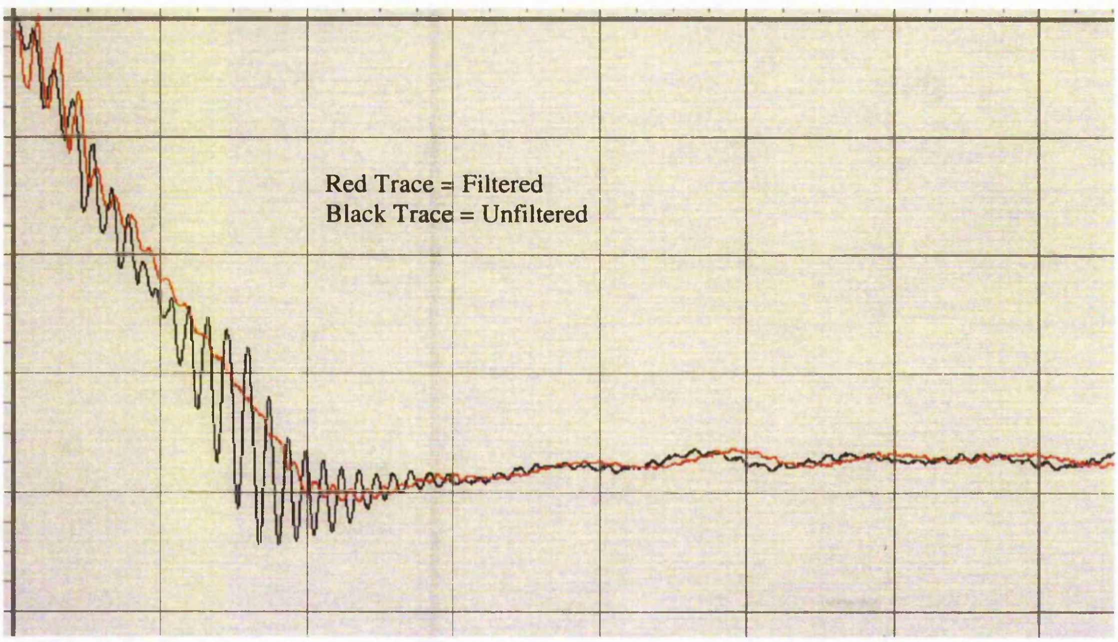


Figure 79 : Expanded view of Dynamic Position Error for Filtered and Unfiltered Velocity Profiles.
Y-axis = Distance Error (¼ Fullstep /div) , X-axis = (Distance travelled) 200 Fullstep /div.

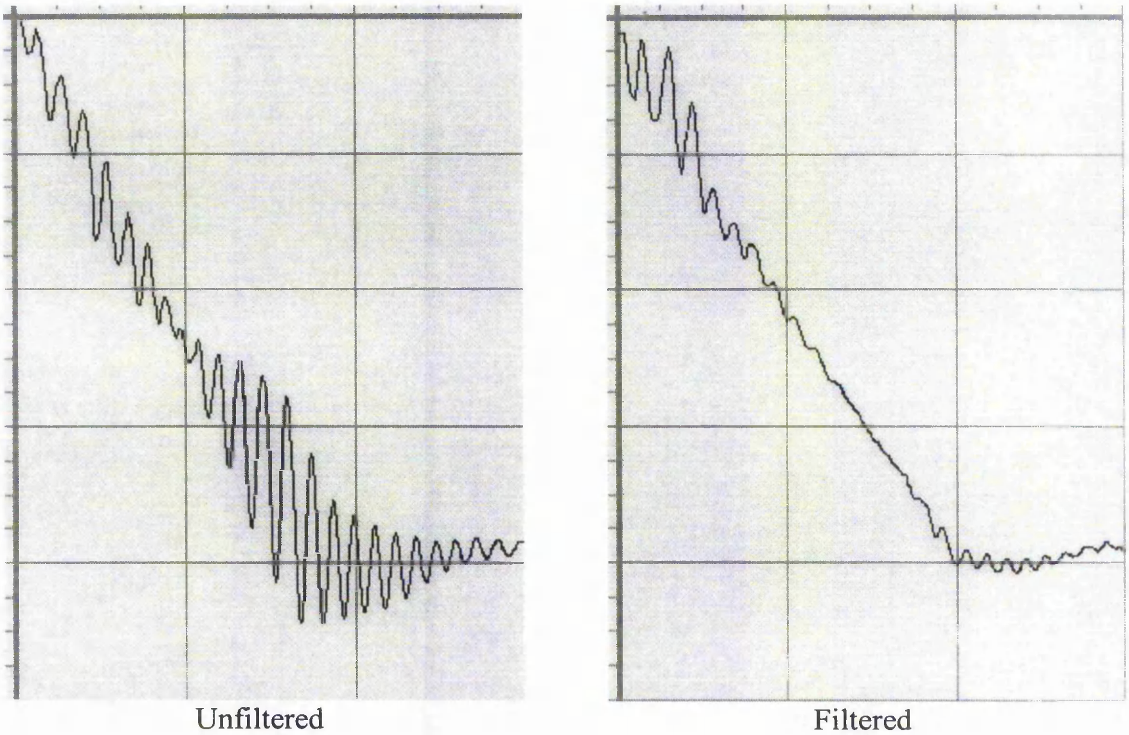


Figure 80 : Separated view of Dynamic Position Error for Unfiltered and Filtered Velocity Profiles.
Y-axis = Distance Error (¼ Fullstep /div) , X-axis = (Distance travelled) 200 Fullstep /div.

Chapter Seven

Discussion.

Chapter Seven Discussion.

7.0 New Techniques & Algorithms.

The report has presented several ways of improving the performance of continuous path stepper motor driven multi-axis machinery. The reader can see the benefits of such methods by monitoring the velocity fluctuations of the output response in the form of velocity time graphs.

The generation of smooth velocity profiles for multi-axis continuous path motion is a complex real-time computational task. The generation of interpolated chord segments coupled with synchronised motion generation frequently leads to situations where the machine dynamics are excited. The author has demonstrated that the performance of multi-axis machinery could be improved if the velocity profiles were smoothed as an additional post-process to the main controller. The act of smoothing the velocity profiles such that the machine dynamics are not excited means that the machine performance is effectively doubled. Instead of only being able to traverse at 2m/min, now the machine has greater acceleration potential and can traverse at 4 m/min.

The Inverse velocity filter uses discrete distance sampled data. This is novel in control terms since most controls systems are continuous or discrete time systems. The reason for using discrete distance based control is based upon the early sampling theorems. Originally back in the fifties when a quantity needed to be measured it could be measured at regular time intervals or the time could be recorded when the quantity changed state. If encoders were used to measure distance then they would be called pulse counting (discrete time) and pulse period counting (discrete distance) based sampling respectively. If the quantity changed state frequently then the pulse period counting method would produce a sample at each change, requiring vast amounts of storage space. In the fifties this was very expensive so the pulse counting method was used. In the pulse counting technique the approximation would have a tolerance of +1-2 pulse counts, depending upon when the counters were queried. This led to great research interest into finding higher order approximation theorems in order to gain a more accurate approximation, such as the research conducted by Forsythe[50].

This meant that the pulse period counting method was used for only slow changing signals, such as daily rainfall measurements where the amount of data to be stored would be low. However times change and what once held true sometimes changes such that the reverse is true. Today memory is cheap & it is more cost effective to produce a high precision timing circuit then it is to etch microscopic lines onto an encoder. For both methods the approximation is dependant upon the resolution of the encoders, putting aside methods such as the one presented by Forsythe [50] then the higher the resolution the better the approximation. For the discrete time method the quantisation error is dependent upon the update frequency, the higher the update frequency the greater the quantisation error. In the case of discrete time, the signal source might have a high sampling frequency to satisfy the Nyquist sampling rate but have a lower update frequency. In the case of the discrete distance method the quantisation error is dependent upon the inverse of the sampling frequency, such that as the sampling frequency increases then the quantisation errors diminish.

Therefore it is the belief of the author that the pulse period counting method is the best of the two methods provided the encoder resolution is sufficiently high (typically above 1000 pulses per revolution) and the signal frequency is not excessive (below 5MHz). The reader can see evidence of this in the clarity of the results shown in chapter 3 where output response is accurately approximated. The use of this method for the filter means a decrease in positional error due to quantisation, when it is contrasted against a conventional FIR filter. The main disadvantage being a variable phase shift due to an increase in input frequency. This can be accommodated provided the number of points used in the smoothing algorithm is kept low. Thus a compromise of smoothing vs. phase shift must be found in order to limit the positional error. Academically the weighting to the samples would produce different filtering effects, i. e. such as the weights used in a Butterworth filter. However these effects are negligible over the fact that the velocity profile is smoothed.

The measurement of the dynamic positional error presented in chapter six, and used to validate the filter is important to the manufacturers of stepper motor driven CNC machinery. This method can be used to measure the performance of the machinery in a number of ways, either in terms of positional error or when combined with the static torque characteristic, in terms of the torque used. Techniques such as frictional belts are often used to find the peak torque by increasing the friction until stalling occurs, these methods are useful for rough approximations as they suffer from numerous calculation inaccuracies. These inaccuracies stem from trying to calibrate the friction and failing to take into account the changes in friction along the lead screw. By accurately measuring the rotor displacement and referencing this to the peak static torque characteristic then a more accurate approximation of torque utilisation can be found.

7.1 Effectiveness of the Research for the field of Stepper Motor Driven CNC machines.

During the progress of the research the author has seen the performance of the Pacer Cadet machine being improved from a maximum top-speed of 2 metres/minute to a maximum top-speed of 4 metres/minute as a direct result of the group research conducted by the author and Axiomatic Technology Ltd. The feedback provided by the author in the form of velocity vs. time graphs has facilitated the improvements in machine performance conducted by the Axiomatic staff. During this time there have been developments made in other aspects of stepper motor machinery. A stepper motor manufacturer has developed a stepper motor that contains a viscous liquid inside the motor to dampen the high frequency vibrations developed by the stepping action of the stepper motor. Testing conducted by a stepper motor driver company (Stebon Ltd.), demonstrated that the motion of the motor was smooth and the vibrations caused by the stepping action were considerably dampened. However the conclusion is that the stepper motor is only of limited use, because of the limited performance caused by the viscous friction being added in the first place, Cliff Joliff [85].

The role of electronic damping has grown considerably in the last few years, the inverse velocity filter will hopefully be only one solution amongst many that emerge in the next few years, developed in academia or in industry. The main stem of the research however is not the solution, but the techniques and investigations that were carried out in order to discover the main limitation in performance of multi-axis stepper motor driven machinery. The performance limitation arises from vibrations causing an additional load to be reflected back towards the stepper motor. The reflected load is an unknown quantity and therefore is not used in the theoretical torque calculations. The stepper motor is being actuated at its maximum performance due to the vibrations caused by interpolation. But this is not realised, so the difference between the theoretical and real performance is seen as a limitation.. The Dynamic Displacement technique can be employed to measure dynamic position error, from which the true performance can be derived.

7.2 Aspects of the Research for other Fields of Engineering.

Stepper motors are widely used in a variety of industries. One such industry is the space industry where the digital nature of the stepper motors and the excellent weight torque ratio of the motor provide significant benefits over the choice of servomotors. Stepper motors are widely used as the actuators for the control of the position of the satellite dish. In this case the only constraint is that of accurate positioning of the satellite. There is not even any requirement to have the two or more axes synchronised so independent point to point movement of the axes will suffice. However the main requirement for satellite positional system is precision of movement that can be achieved by the motor. Gear ratios and micro stepping are often used to obtain greater static resolution. This is also true for silicon wafer fabrication where stepper motors are used as actuators for the slide tables. In these cases then the inverse velocity filter is of limited use because the motion has been made smoother at a cost of speed performance. However the knowledge base on stepper motor motion built as part of the research is of importance.

The reader can apply the dynamic position error technique to these situations to calculate positional errors and torque utilisation. The encoder resolution may have to be increased and inaccuracies may be present due to backlash caused by gear meshing but the technique would provide an estimate of the level of torque required. The use of oversized motors to compensate against dynamic disturbances causing stalling may not be very expensive on earth but to send these motors to space will cost considerably more due to the extra fuel cost. Therefore the more knowledge there is concerning the motion of stepper motors the better the design of the system will be.

Chapter Eight

Conclusion & Further Work.

8. 0 Conclusion.

The aim of the research was to develop motion control techniques in order to improve the performance of continuous path multi-axis stepper motor driven machinery. Typically this type of machinery suffers from rough motion, which in turn may lead to path inaccuracies and stalling. This rough motion may limit the performance of the machine such that it may need to be down rated to a fraction of its potential. Previous work by Steiger et al. has identified that interpolation errors, and sharp changes in acceleration (arising from quantisation of time and distance), could theoretically be the cause of stepper motor driven machinery exhibiting rough motion.

The author experimentally verified the theoretical assumptions of Steiger et al. He concluded with Steiger that the majority of the performance limitations were caused by rough motion as a direct result of acceleration and interpolation. The author investigated the extent of the problem by instrumenting a number of CNC machines including a custom-built test rig. System parameters such as acceleration rate and motor current were changed to monitor the effect they had upon the motion. The roughness of motion was expressed in terms of velocity and position errors. The process of experimentation has demonstrated that there are two important factors concerning stepper motor driven machinery. The first is the interpolation algorithm and its respective performance in terms of accuracy and resolution of distance. The other factor is the smoothness of the velocity profile during changes in acceleration and deceleration. These factors are inter-linked and as such a compromise has to be made.

In order to accomplish the research goal it was determined that the approach should be to produce path smoothness through velocity profile shaping. This was based upon earlier work conducted by Palmin et al. , concerning velocity profile generation for single axis machinery. Palmin puts forward the case of maximum torque utilisation via the use of a parabolic velocity profile. It has been shown in the thesis that rough motion predominantly arises from machine excitation caused by the interaction of acceleration and interpolation. This interaction causes an additional load torque to be placed upon the stepper motor in such a way that the machine requires more torque to operate effectively. The maximum acceleration that the machine can achieve is proportional to the remainder of the available torque, thus the performance is limited by the increase of dynamic load torque. Since torque generally falls with increased motor speed there is thus an inevitable reduction in attainable speed. The author confirmed the single axis hypothesis put forward by Palmin, first by mathematical modelling and simulation, secondly by the implementation of a suitable velocity profile. The author found that in the multi-axis situation the hypothesis put forward by Palmin may not be realistic due to the synchronisation between the axes needed for continuous path motion. Depending upon the cutter path the slave axis might exhibit rough motion because the velocity fluctuates. This rough motion may propagate back to the master axis, therefore limiting the master as well as the secondary.

Ideally there are two ways to generate smooth continuous path motion. If close synchronisation is required then a secondary process contained within the controller could smooth velocity profile for each axis. The second method is to develop an interpolation algorithm that considers the smoothness of the velocity for each axis as well as the static motion. This second solution is a complex problem if the axes need to be closely synchronised. To confirm the author's hypothesis regarding multi-axis machinery the author implemented a novel inverse velocity filter to act as the secondary smoothing process. The filter was developed in software residing in a digital signal processor, and as such is independent of the controller used. The inverse velocity filter smoothes the velocity profiles generated by the controller in real time. The novelty of the filter arises from its use of discrete distance sampled data rather than discrete time sampled data. By smoothing the velocity profile in software in a way similar to the velocity profiles generated by Palmin then the filter reduces the effective torque limitation, thereby allowing the stepper motor to use more of its available torque, hence better performance. The process of smoothing the velocity also has an effect upon the cutter path. The hypothesis is that by smoothing the motion, any changes to the cutter path can result in better path accuracy.

Experimentally it has been shown that the interaction of interpolation and acceleration causes a reflected load being placed upon the stepper motor, thereby limiting the performance. The additional load being reflected back from the machinery is difficult to calculate since it relies heavily upon the current condition of the machine, for example what lubricant is used on the leadscrews etc. Building upon the work conducted by Acarnley et al. and Palmin et al. , it became clear that the problem of limited performance in multi-axis stepper motor driven machinery relies on the determining the dynamic torque under load conditions. In order to measure the load requirements placed upon the stepper motor whilst under continuous path conditions the author devised a method of measuring the dynamic torque of the stepper motor and thereby the load torque. This method of deducing the static and dynamic rotor position allows the engineer to calculate the true torque/speed graph for the system under test. This method was used to measure the effectiveness of the filtering solution, which proved the hypothesis put forward by the author regarding improved path accuracy through smoother velocity profiles, resulting in less excitations of the machine dynamics.

Thus the significant performance limitations of continuous path stepper motor driven machinery have been demonstrated. These experiments concerning the motion of CNC machinery have resulted in new knowledge being developed within the field of continuous path motion control. The author has experimentally proved the hypothesis put forward by Steiger, and has adapted the hypothesis put forward by Palmin to include multi-axis machinery. Steps to overcome the performance limitation have led to the development of a novel inverse velocity control algorithm. This solution was realised in hardware and reduces the excitation of the machine dynamics significantly. The research has shown that a higher level of performance in terms of path accuracy and top speed can be achieved if techniques such as the one presented are utilised in their control system.

8.1 Further Work

The author has demonstrated that by applying a secondary filtering process to the primary interpolation/motion control process the engineer can successfully reduce rough motion present in some multi-axis stepper motor driven machines. This holds true for certain interpolation algorithms where the interpolation and acceleration are closely synchronised across the number of axes controlled. In some cases the synchronisation essential for continuous path motion can be achieved indirectly, such as in the case of direct function interpolation. Direct function interpolation is where the shape is represented by a series of parameters and the corresponding motion of each axis is a function of these parameters. This type of interpolation can be categorised in the same way as spline interpolation. Spline interpolation is the process of fitting and fairing a series of mathematical splines to a given shape. Until recently such interpolation techniques have been so computationally intensive that only a limited number of samples can be used on the prototype systems. The underlying problem is that the interpolation algorithm is not calculating static points as in the case of computer graphics, but rather time-constrained motion. With recent changes in microprocessor architecture and related semiconductor components then these systems are moving from the laboratory to the mainstream industrial motion control systems. The use of such interpolation techniques that can maintain path accuracy whilst only maintaining the minimum amount of synchronisation may provide an alternative solution to the one presented within this report. Such a solution could achieve a greater level of machine performance and path accuracy if successfully implemented, however there are currently still many problems to be resolved.

The inverse velocity filter smoothes the input velocity profile and works well under normal circumstances. However in some situations the machine may experience an external disturbance that will excite the machines' dynamics and cause the machine to stall. In these cases then the unit distance feedback control side could be explored. If a FIR filter can be changed in such a way that it works on discrete distance rather than discrete time then this opens up the question of whether other discrete time methods could also be converted to use discrete distance data. The benefit of discrete distance over discrete time is that the signal is not so heavily quantised, which means the measured signal is a closer approximation to the real signal. This also opens up the question of whether or not greater accuracy can be gained for unit distance data if higher order approximations are used. Forsythe and Thomas [47] discuss ways of calculating the best sampling period for discrete time sampling using their polynomial approximation theorem. One avenue to explore in the field of smooth continuous path motion is to see if these techniques could be adapted and applied to discrete distance data. The problem with the discrete distance sampling approach is that as the input signal changes then so does the characteristics of control stage, be it a filter or feedback controller. These parameters may need to be changed on-line to take into account the signal changes.

Another area that can be expanded is the dynamic torque measurement technique. The frictional components of the machine depend greatly upon the current position of the table upon the lead-screw. This means that the static torque is not constant across the length of the machine bed, this may cause the recorded signal to drift due to inaccuracies. In order to gain a more precise reading it would be better to perform a series of scans across the available working area, thereby obtaining the degrees of friction across the whole area of the machine bed. This technique could then be used in conjunction with the path planning data to discover the exact reasons for the CNC machine to stall.

These areas when explored would provide additional knowledge to the field of motion control engineering and provide a useful insight to the condition monitoring of multi-axis CNC machinery. As it is sometimes said "*gaining a PhD. is not the end of the research but the start of a new chapter*"

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Appendices

Appendix A.

Digital Data Acquisition board.

INTRODUCTION

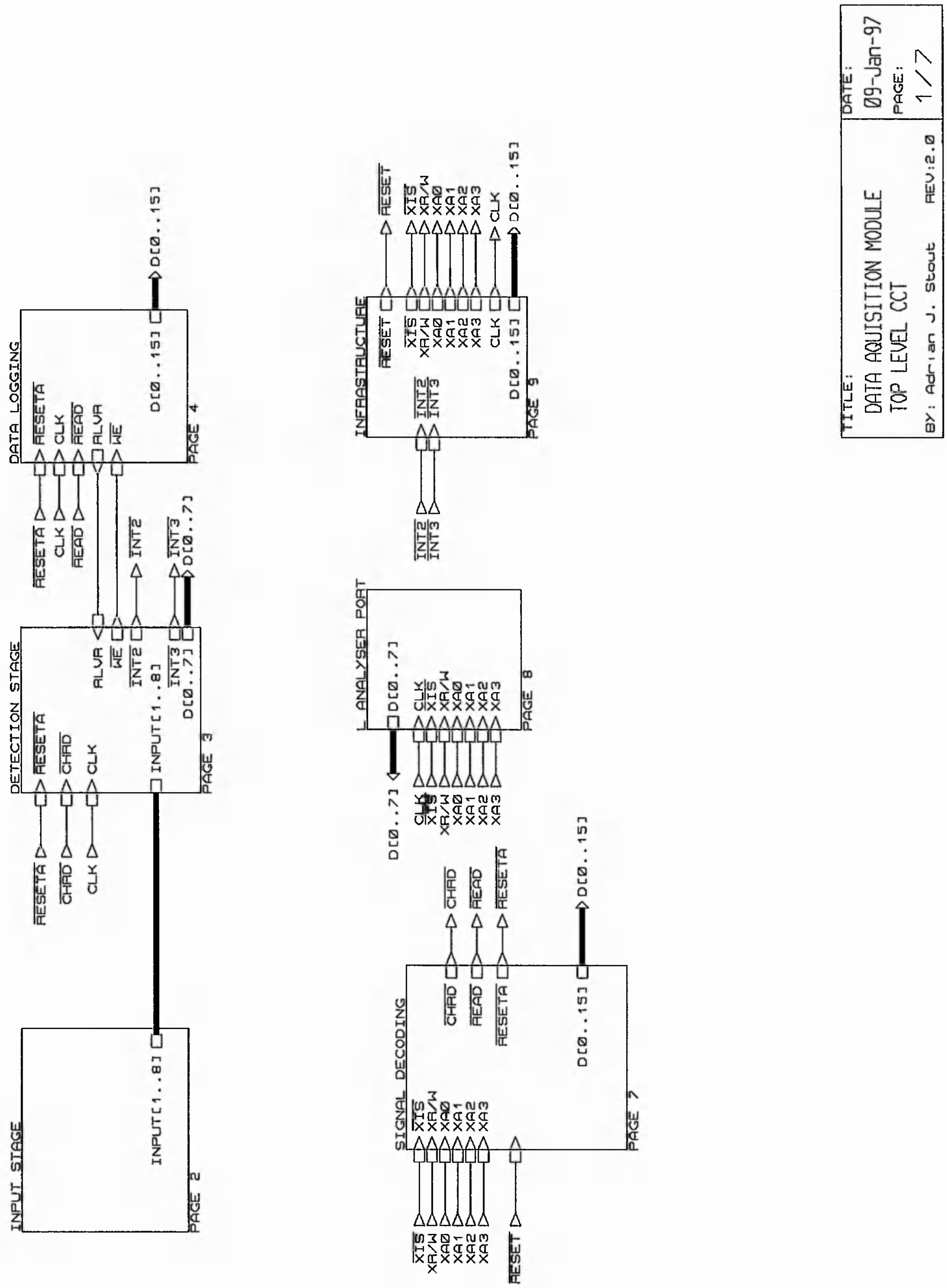
The Digital Data Acquisition Board (DDAB) was initially developed by Leo McManus [11] at the Nottingham Trent University, Department of Computing. The system time stamps digital pulses, from which the length in time, of these pulses can be determined. These pulses can be obtained from a digital incremental encoder, which produces pulses corresponding to the radial movement of a shaft. The other method of capturing velocity/displacement information is to count the number of pulses over a discrete length of time. The first method offers a distinct advantage over the second method, the fact that in today's world it is easier to develop a high frequency precise crystal clock than it is to produce an ultra high resolution encoder disk. Other than this advantage, they both suffer from the same disadvantages since one approach is the direct analogy of the other, simply time and distance swapped.

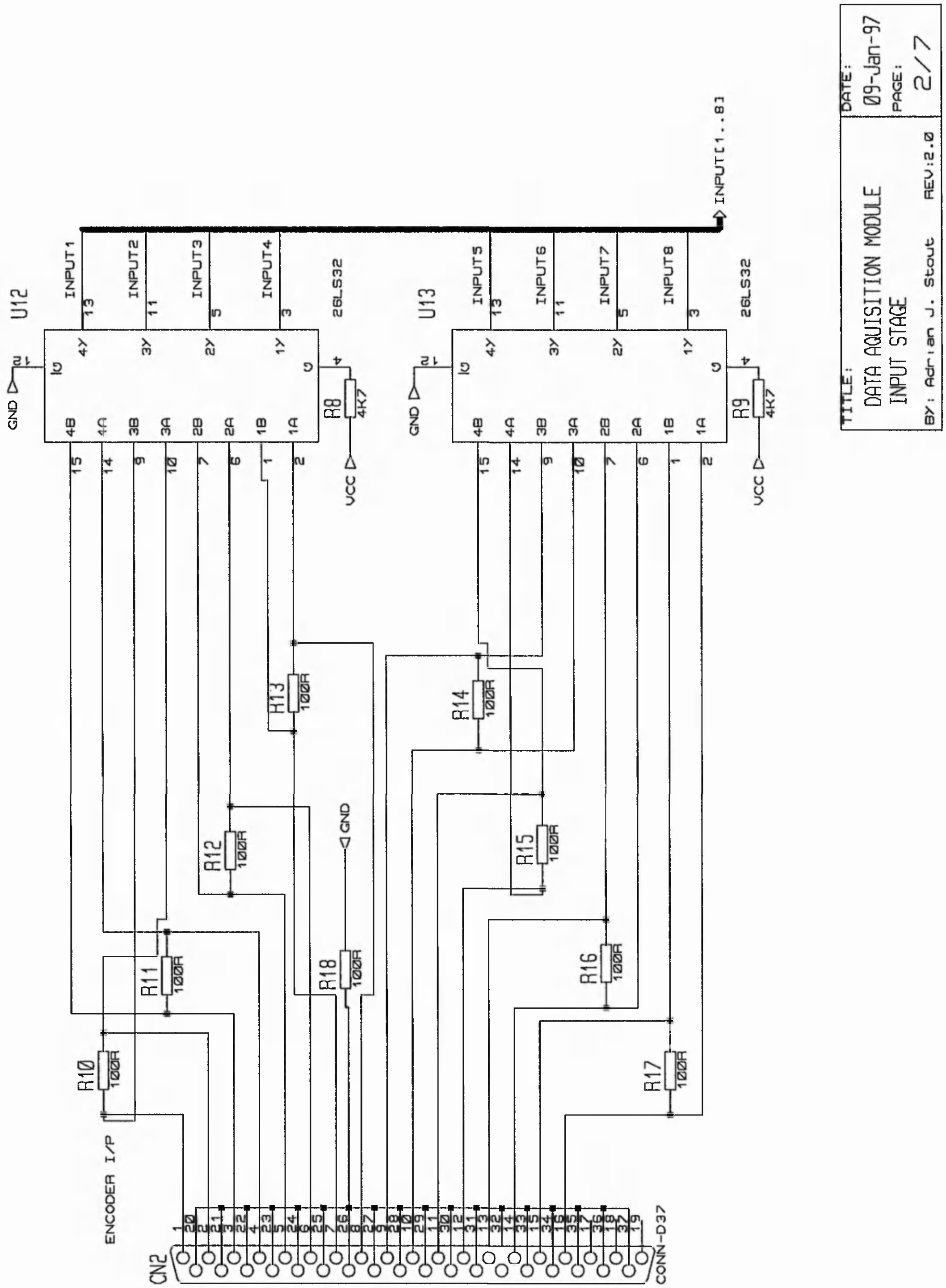
The first method of data capture was initially developed by various independent parties over thirty years ago, as a means of recording velocity information from very low speed applications. This method was used for interpolating the velocity between successive pulses, of shafts with a very slow rotational movement. The drawback with this method is the amount of memory needed for high speed applications, there becomes a point where it takes less memory to store the amount of pulses arriving within a discrete time period that to store the time stamp of each individual pulse. In the past with the prices of memory being very high, the second method of data capture was often used instead of the first. The availability of cheap mass storage memory devices has allowed the researchers at The Nottingham Trent University to design a inexpensive fast digital capture system that overcomes this disadvantage. This development means that now higher frequency pulses can be now be logged and recorded cheaply. The maximum frequency is limited only by the storage devices and the clock frequency.

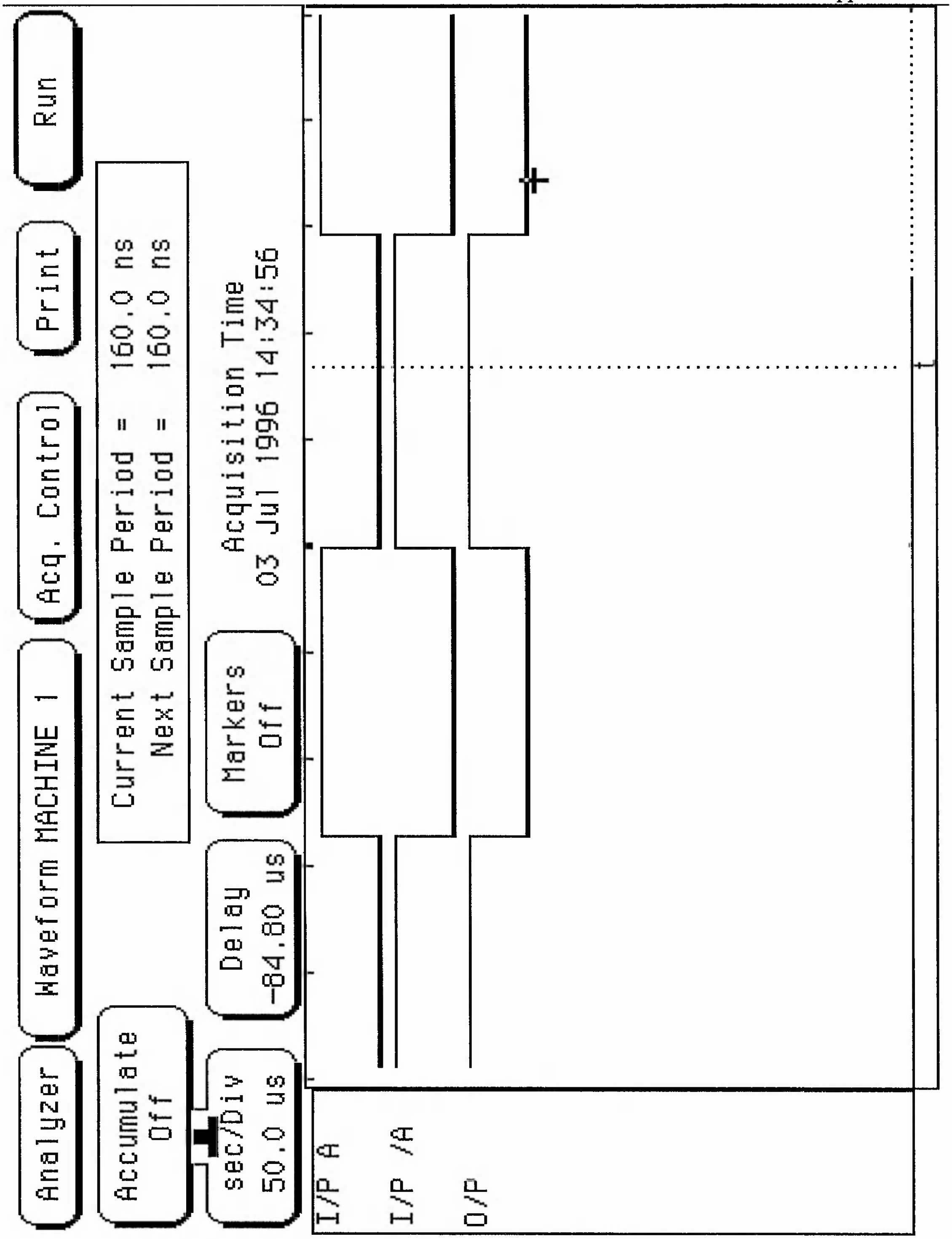
DESIGN.

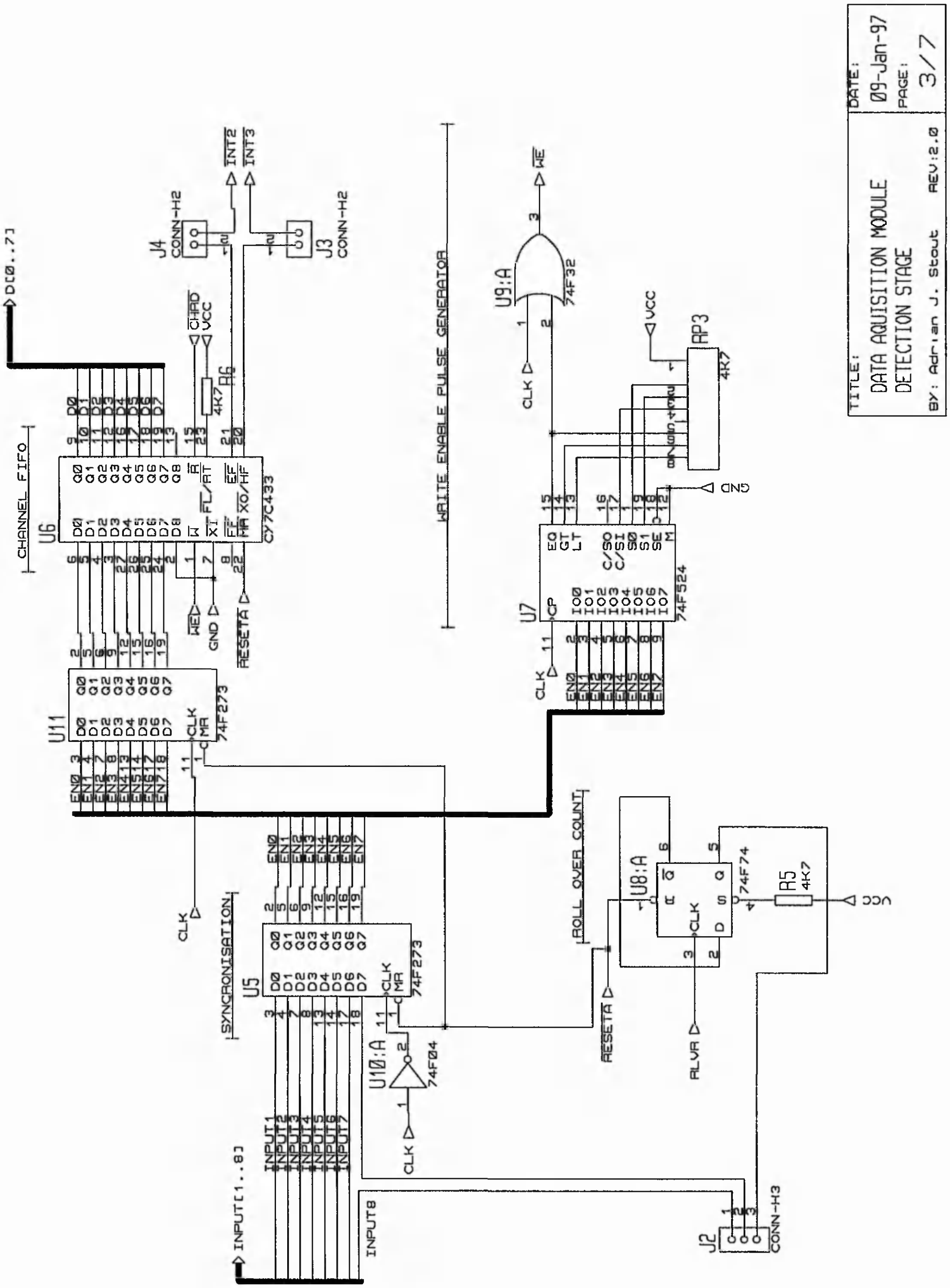
The fundamental principle of the design is as follows :

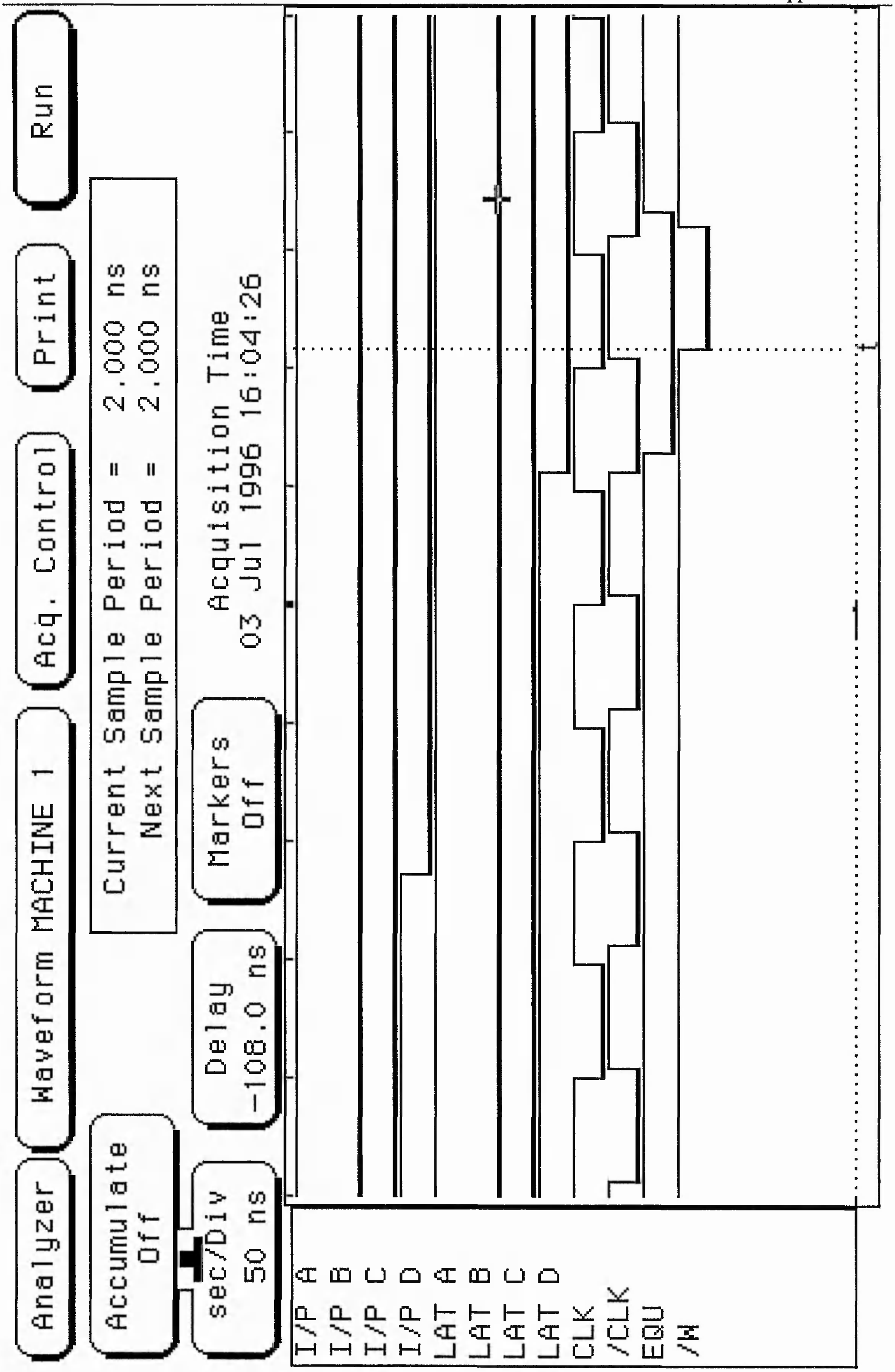
A pair of complimentary digital signals, typically from an encoder, are fed into the detection board. These signals are then passed through a differential line receiver. The output from which is double latched to eliminate Meta-stable conditions. The signals are then fed from the latch into a comparitor which upon detecting a change initiates a write to the FIFO circuits. One set of FIFO circuits stores the input signal so that the user can tell which signal changed. The other set of FIFO circuits are fed from a sixteen bit counter which stores the number of clock pulses that have passed since it was initialized. Thus the digital input signals are time stamped when they initiate a change of state, either from a logic one to zero, or vice versa. This time can then be used to calculate the time period between the changes, hence the frequency of the digital signal. When the FIFO's become Half-full they signal an interrupt to the Host system. The system communicates via a high-speed link to a host D.S.P. system.

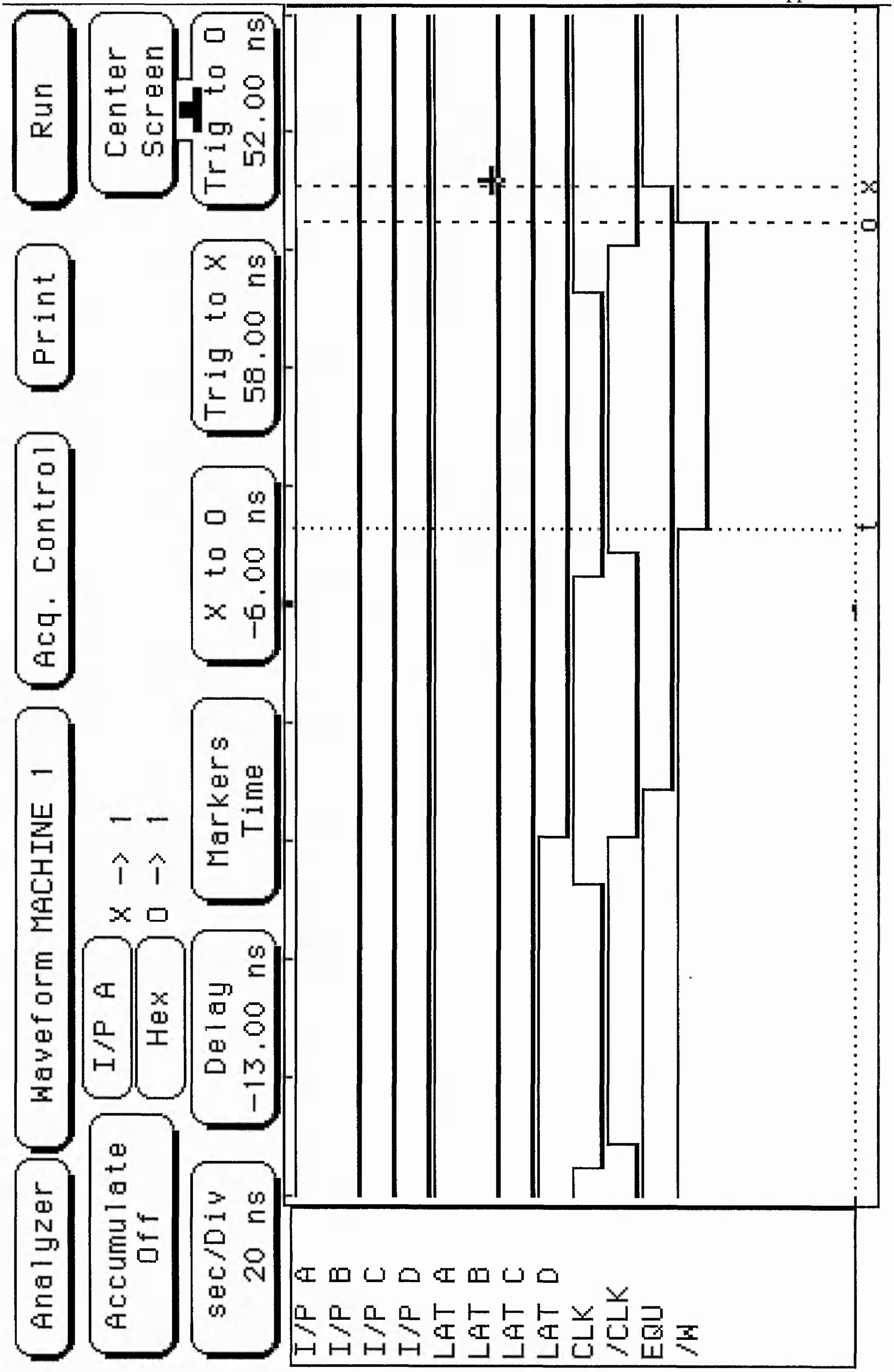

INPUT STAGE
PAGE 2
DETECTION STAGE
PAGE 3
DATA LOGGING
PAGE 4
SIGNAL DECODING
PAGE 7
ANALYSER PORT
PAGE 8
INFRASTRUCTURE
PAGE 9
TITLE:
DATA AQUISITION MODULE
TOP LEVEL CCT
BY: Adrian J. Stout
REV:2.0
DATE:
09-Jan-97
PAGE:
1/7

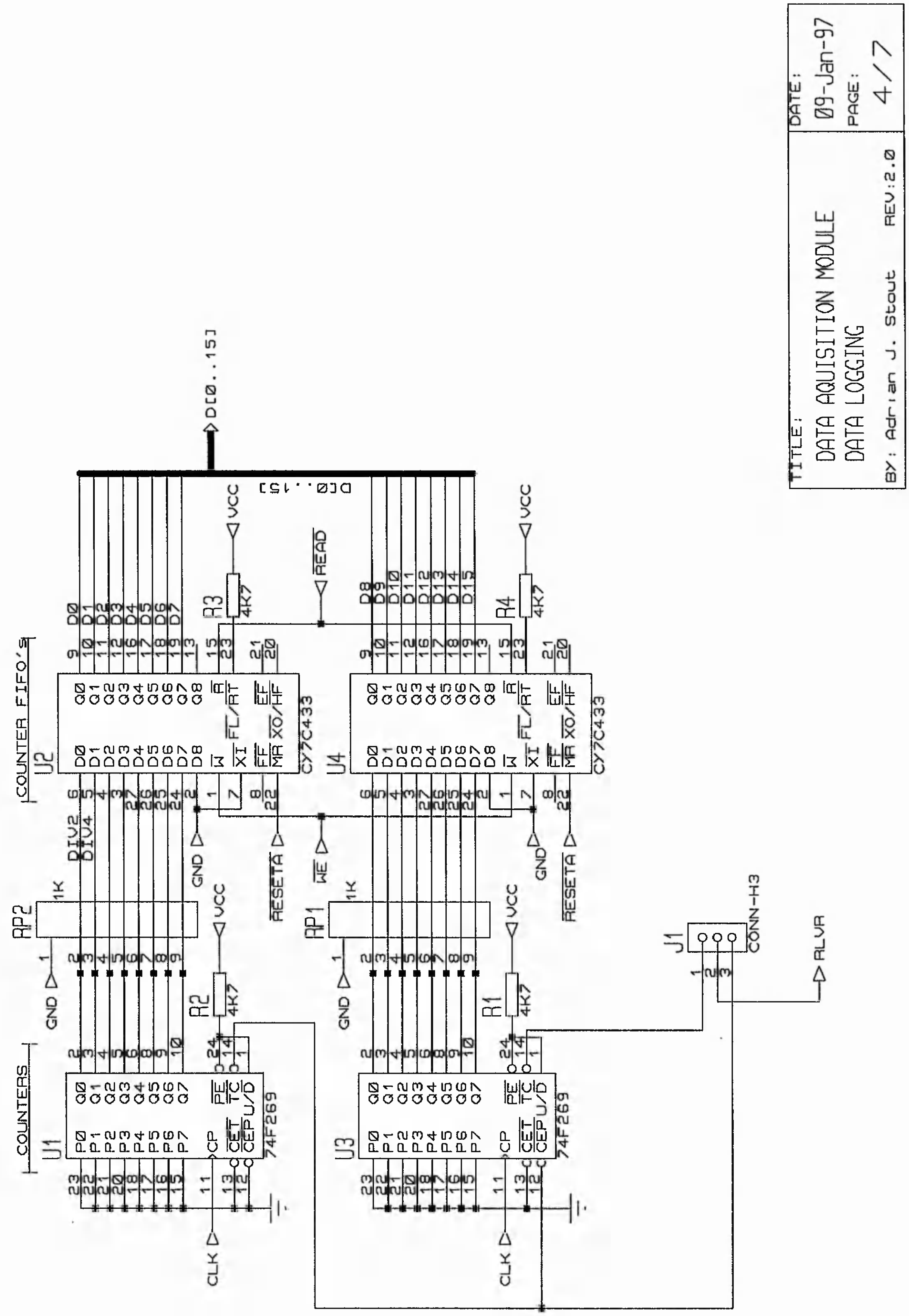

TITLE:
DATA AQUISITION MODULE
INPUT STAGE
BY: Adrian J. Stout
REV:2.0
DATE:
09-Jan-97
PAGE:
2/7
INPUT[1..8]
U12
26LS32
U13
26LS32
INPUT1
INPUT2
INPUT3
INPUT4
INPUT5
INPUT6
INPUT7
INPUT8
GND
VCC
R8
4K7
R9
4K7
R10
R11
R12
R13
R14
R15
R16
R17
R18
100R
ENCODER I/P
CN2
CONN-D37

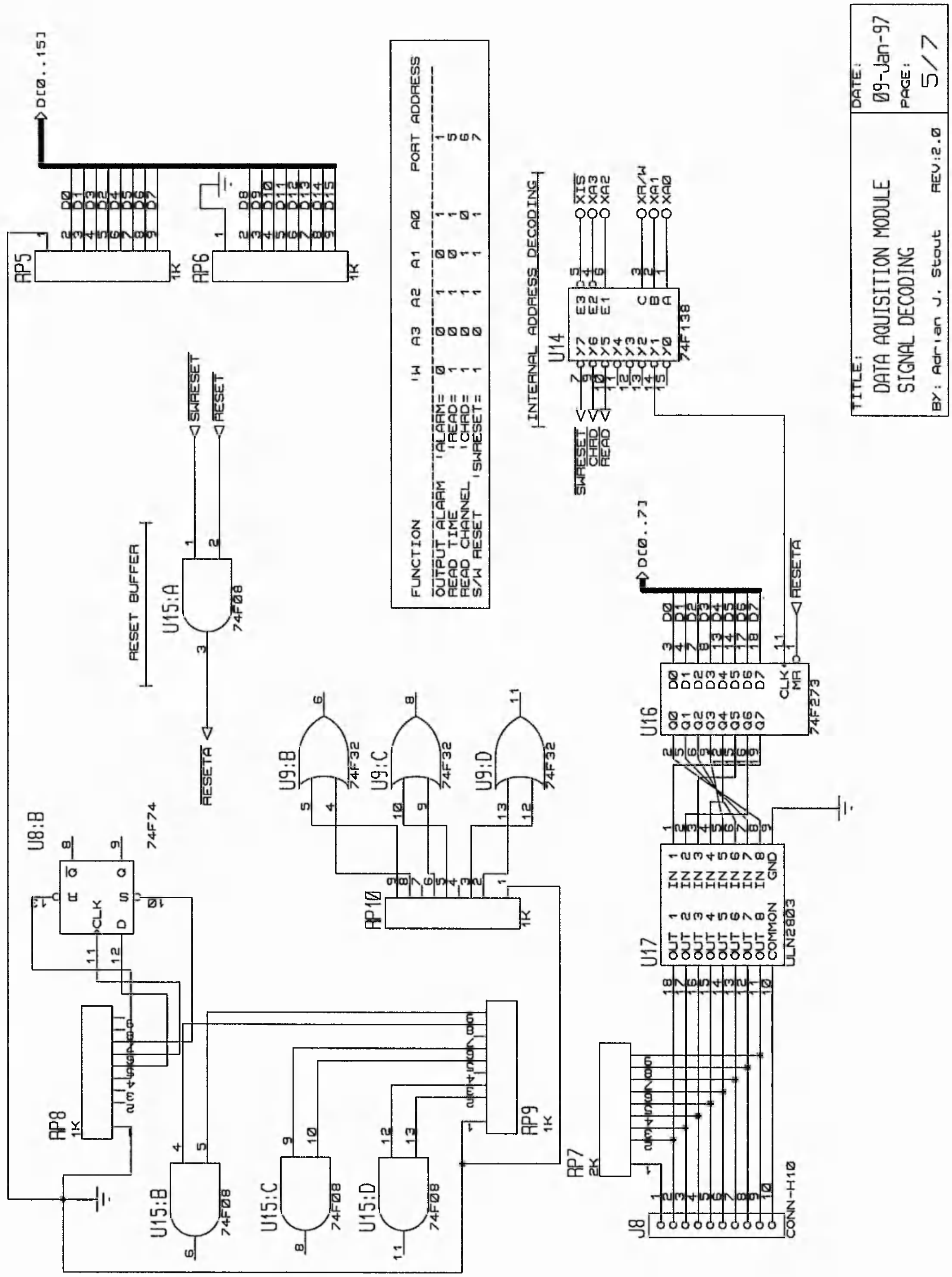

Analyzer
Waveform MACHINE 1
Acq. Control
Print
Run
Accumulate
Off
Current Sample Period = 160.0 ns
Next Sample Period = 160.0 ns
sec/Div
50.0 us
Delay
-84.80 us
Markers
Off
Acquisition Time
03 Jul 1996 14:34:56
I/P A
I/P /A
O/P

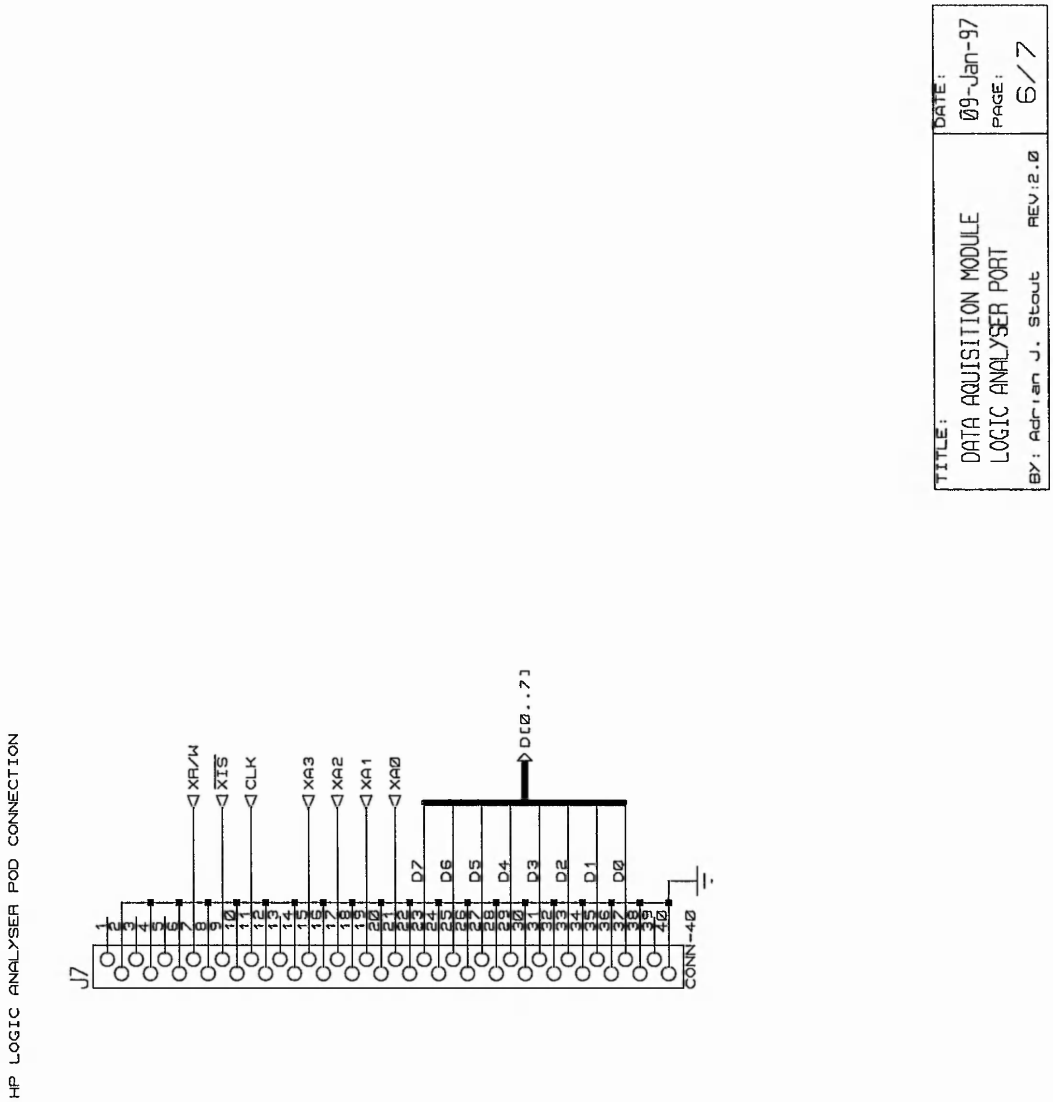

TITLE:
DATA AQUISITION MODULE
DETECTION STAGE
BY: Adrian J. Stout
REV:2.0
DATE:
09-Jan-97
PAGE:
3/7
CHANNEL FIFO
WRITE ENABLE PULSE GENERATOR
ROLL OVER COUNT
SYNCRONISATION

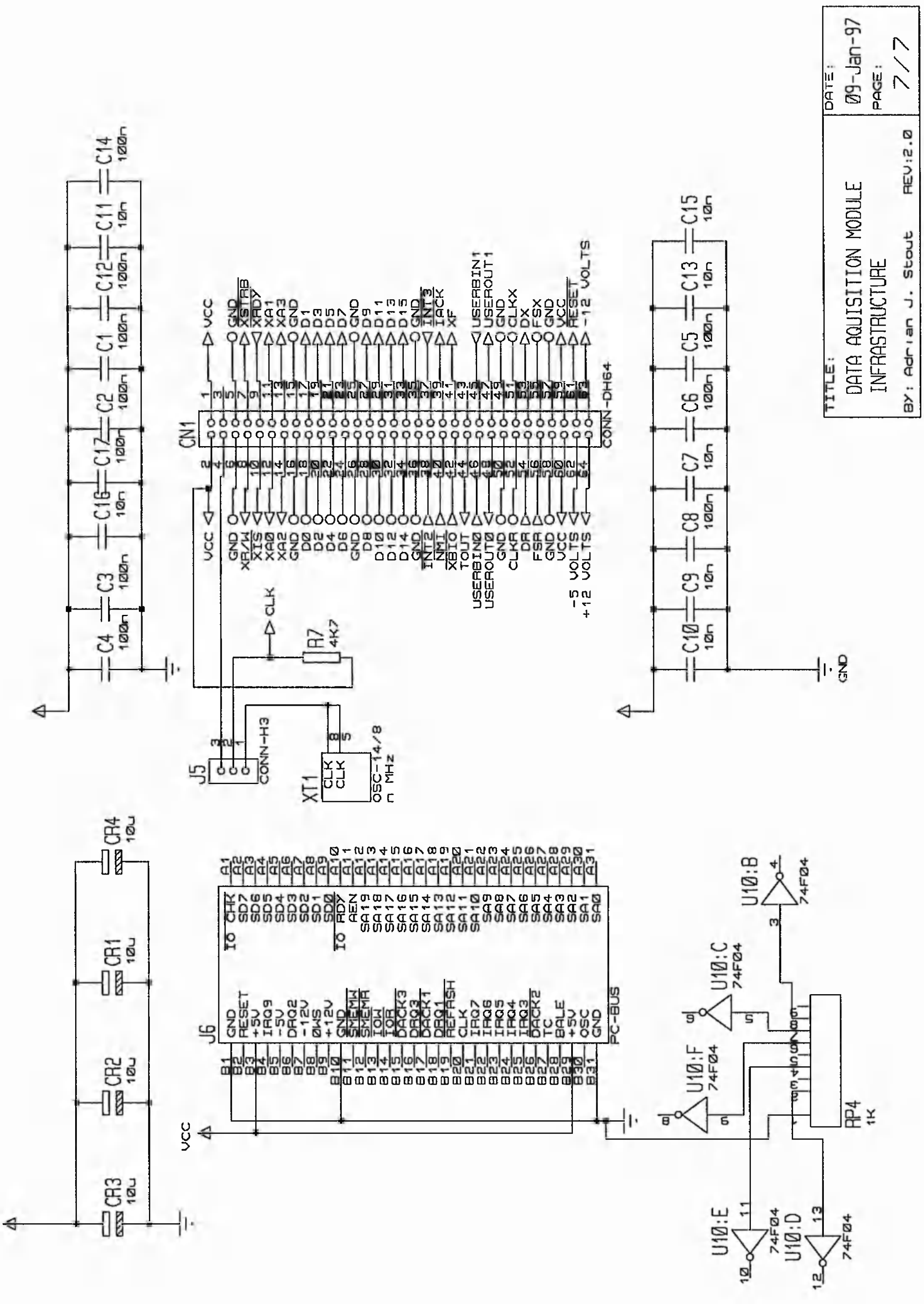

Analyzer
Waveform MACHINE 1
Acq. Control
Print
Run
Accumulate
Off
Current Sample Period = 2.000 ns
Next Sample Period = 2.000 ns
Acquisition Time
03 Jul 1996 16:04:26
sec/Div
50 ns
Delay
-108.0 ns
Markers
Off
I/P A
I/P B
I/P C
I/P D
LAT A
LAT B
LAT C
LAT D
CLK
/CLK
EQU
/W


Analyzer
Waveform MACHINE 1
Acq. Control
Print
Run
Accumulate Off
I/P A
Hex
X -> 1
O -> 1
Center Screen
sec/Div 20 ns
Delay -13.00 ns
Markers Time
X to O -6.00 ns
Trig to X 58.00 ns
Trig to O 52.00 ns
I/P A
I/P B
I/P C
I/P D
LAT A
LAT B
LAT C
LAT D
CLK
/CLK
EQU
/W
X
O
t


TITLE:
DATA AQUISITION MODULE
DATA LOGGING
BY: Adrian J. Stout
REV:2.0
DATE:
09-Jan-97
PAGE:
4/7
COUNTER FIFO's
COUNTERS
U1
U2
U3
U4
CY7C433
74F269
RP1
RP2
1K
R1
R2
R3
R4
4K7
J1
CONN-H3
RLVR
D[0..15]
READ
WE
RESETA
VCC
GND
CLK
DIV2
DIV4


TITLE: DATA AQUISITION MODULE SIGNAL DECODING
BY: Adrian J. Stout REV:2.0
DATE: 09-Jan-97
PAGE: 5/7
FUNCTION 'W A3 A2 A1 A0 PORT ADDRESS
OUTPUT ALARM 'ALARM= 0 0 1 0 1 1
READ TIME 'READ= 1 0 1 0 1 5
READ CHANNEL 'CHRD= 1 0 1 1 0 6
S/W RESET 'SWRESET= 1 0 1 1 1 7
INTERNAL ADDRESS DECODING
RESET BUFFER
D[0..15]
D[0..7]
U14 74F138
U15:A 74F08
U16 74F273
U17 ULN2803
U8:B 74F74
U9:B 74F32
U9:C 74F32
U9:D 74F32
U15:B 74F08
U15:C 74F08
U15:D 74F08
RP5 1K
RP6 1K
RP7 2K
RP8 1K
RP9 1K
RP10 1K
J8 CONN-H10


HP LOGIC ANALYSER POD CONNECTION
J7
CONN-40
XR/W
XIS
CLK
XA3
XA2
XA1
XA0
D[0..7]
D7
D6
D5
D4
D3
D2
D1
D0
TITLE:
DATA AQUISITION MODULE
LOGIC ANALYSER PORT
BY: Adrian J. Stout
REV:2.0
DATE:
09-Jan-97
PAGE:
6/7


TITLE:
DATA AQUISITION MODULE
INFRASTRUCTURE
BY: Adrian J. Stout
REV:2.0
DATE:
09-Jan-97
PAGE:
7/7

DETAILED DESIGN.

Page 1.

This page shows the relevant circuits as Black box entities. The input and output signals for each block are also shown. Each block is described fully in the following entries.

Page 2.

This page shows the connections from the input D-type connector to the relevant differential line receivers. Eight pairs of differential signal channels are fed into the RS 422 receivers. The outputs from the differential line receivers are fed into the detection circuit. The page over-leaf from page 2 shows a trace with a single pair of inputs, A and inverse A, also shown is the respective output as measured by a logic Analyser.

Page 3.

This page shows the components that form the Detection stage of the system. The eight channel inputs are fed into a pair of latches, U5 and U11. The latched signals from U5 are then fed into the second latch and the eight bit comparitor. The comparitor contains the last set of values at time CLK- 1. These are now compared against the present values at time CLK. If there is a change of state on any line the EQU line falls low, and when the clock edge falls a write is initiated to the FIFO, recording the channel upon which the change occurred. When the FIFO becomes half full an interrupt is called to empty the FIF0s. The FIF0s can be written to or read from asynchronously. The next two pages over-leaf from page 3 show traces with all these signals, showing a change of state on input D. The change is passed through the latches and arrives two clock cycles later at the latched output for channel D, (LAT D). This brings the EQU line down, so that when the clock pulse, CLK, falls low a single write pulse is issued.

Page 4.

This page shows the components that form the Data logging stage of the system. The counters are cascaded together in a carry look-ahead fashion to form a sixteen bit counter. The carry from the most significant byte is used as a signal to indicate that the counter has 'rolled over'. This signal is latched and recorded to allow the user to know precisely the absolute time from initialisation. The outputs from the counters will have the facility be either pulled low or high as dictated by the chip classification. For TTL Fast class chips these need to be pulled low. The signals are then fed into a set of counter FIFO's These FIFO's record the time that a change occurred. As above they can be written to or read from asynchronously.

Page 5-7.

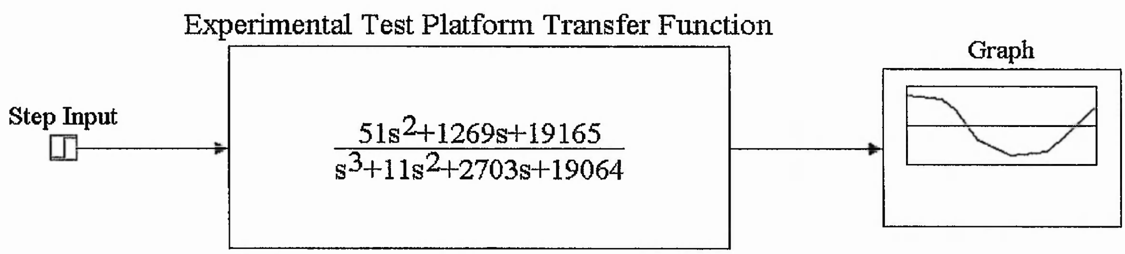
These pages show various infrastructure components. The DSP interface was taken from material contained in the Texas Instrument TMS320C50 DSP manual. The internal decoding and reset circuits are also shown. Page 5 shows additional circuitry that can be used to sound a warning or a signal to a warning LED. Optionally this could be used to fed the drive control of a stepper motor.

Appendix B.

Control System Modeling.

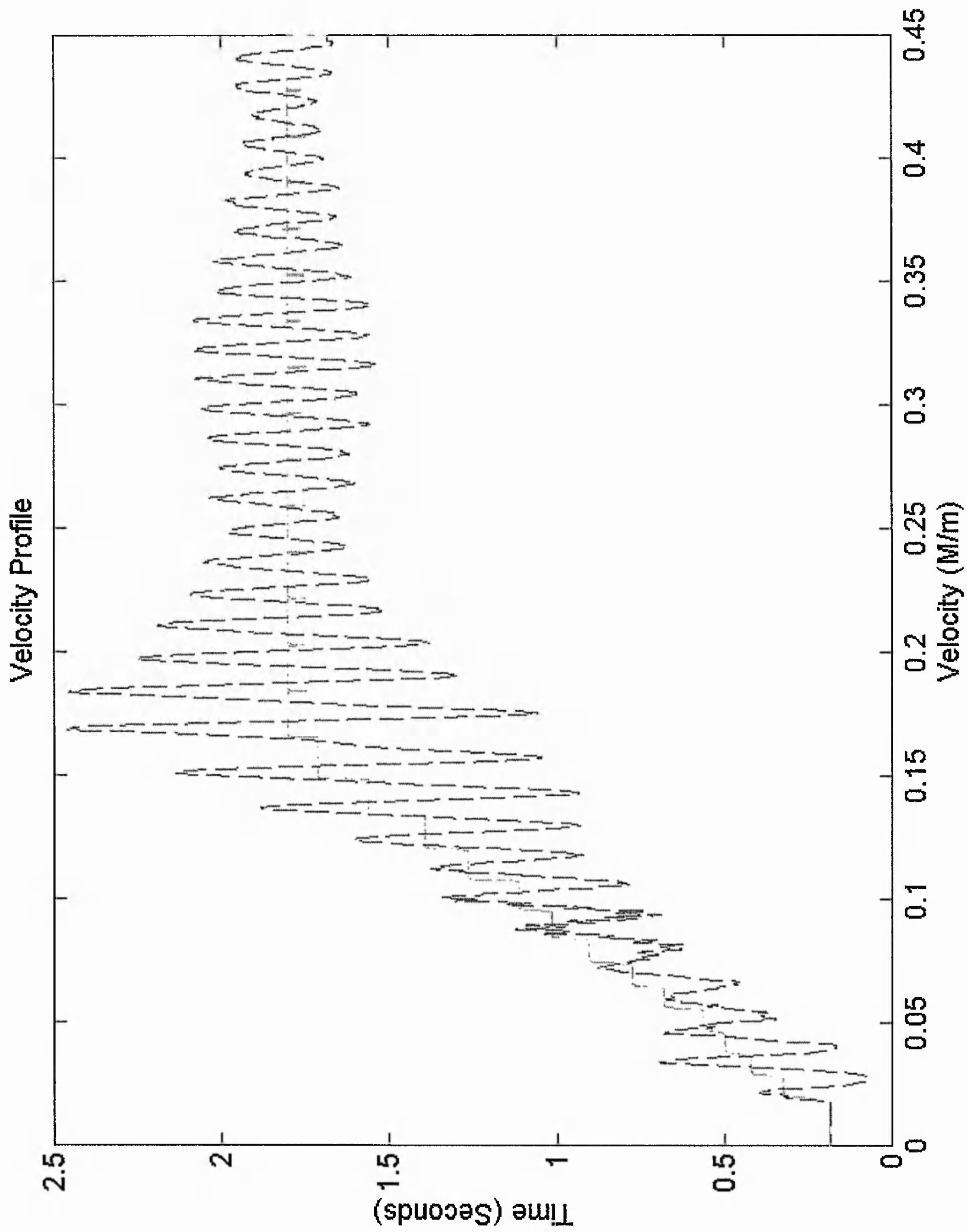
INTRODUCTION.

To control motor driven machinery it is useful to know the systems dynamics and characteristics. The use of these parameters can enable the machine control to be more complex, therefore providing a higher degree of control and contour path accuracy. It is proposed that a new intelligent machine controller be devised to provide open-loop adaptive control to stepper motor drive systems. This controller will improve the performance of continuous path multi-axis stepper motor driven machinery, by overcoming rough motion arising from path generation. This has led the development of an experimental test platform with suitable instrumentation to measure the systems dynamics. The controller will overcome the rough motion problems associated with path generation by dynamically controlling the velocity profile and contour path. This topic has been speculated upon for a few years at the RTMC group based at The Nottingham Trent University and by Axiomatic Ltd. Dr. W. Steiger proposed a new interpolation technique and suggested that the some of the rough motion can be eliminated by improved velocity profile shaping. To validate the idea, data from the system was fed into MATLAB such that the systems dynamics could be modeled.

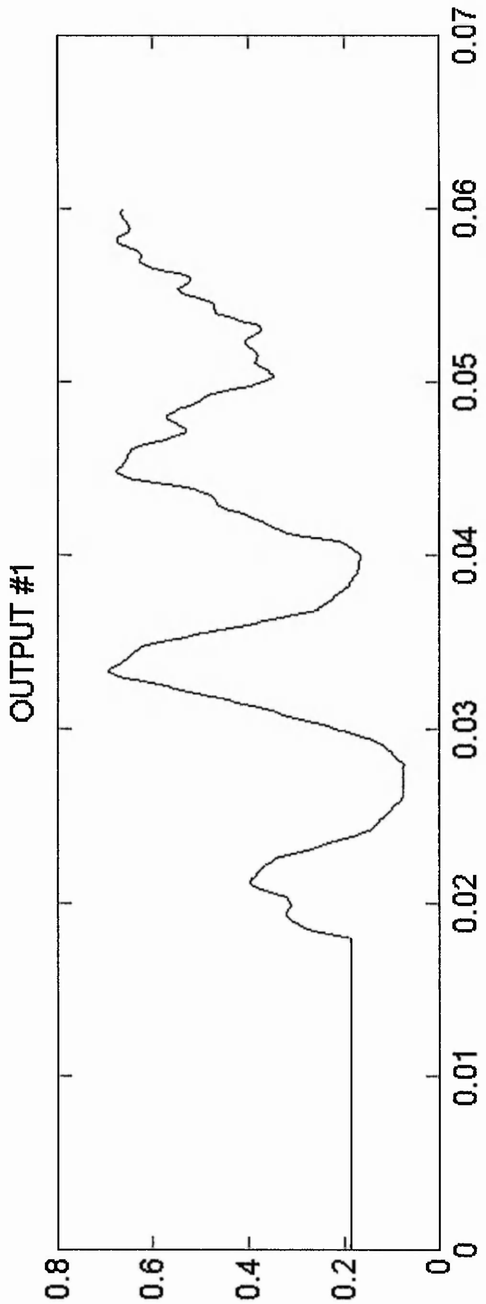


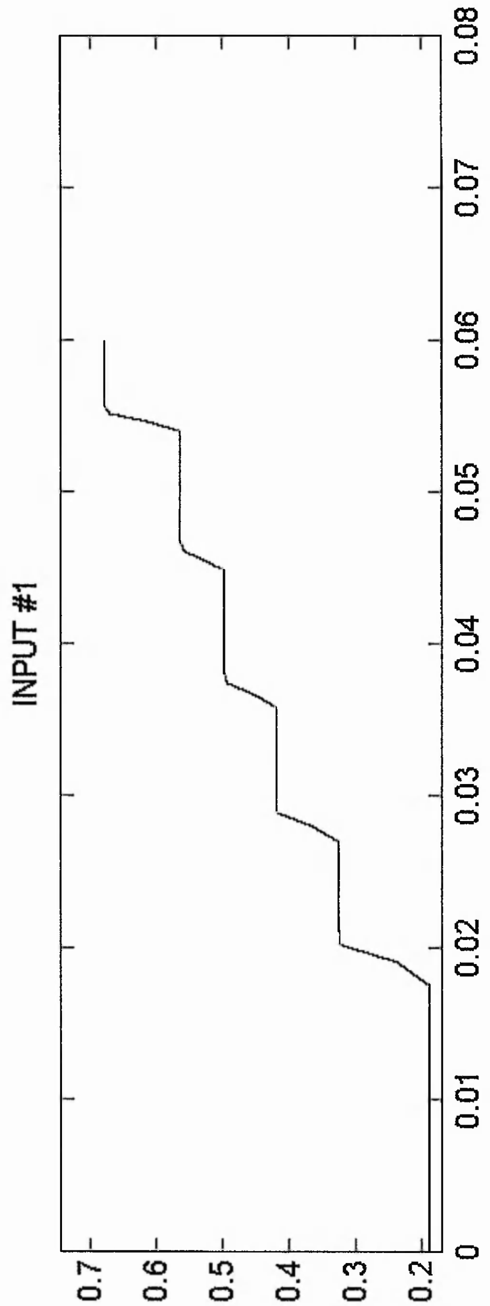
The use of computer modeling & simulation is viewed as an essential part of developing any new system. The model of the system will provide validation to the ideas proposed by Dr. W. Steiger, and the idealized input velocity profile can be determined taking into account of time and efficiency criterion. This idealized input can then be generated on a suitable output stage to validate the theory.

SYSTEM DATA.

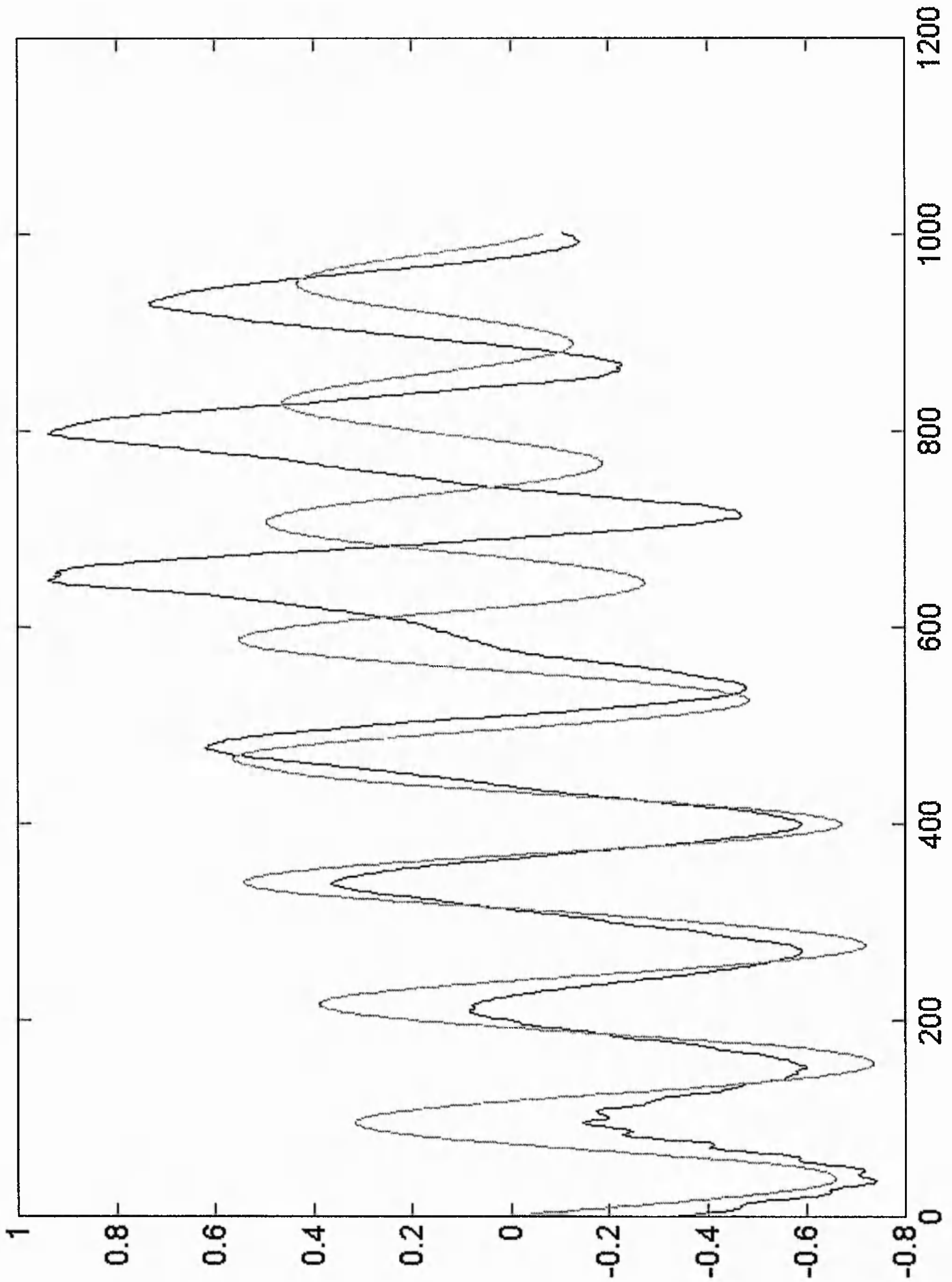

Velocity Profile
Velocity (M/m)
Time (Seconds)
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
0.45
0
0.5
1
1.5
2
2.5

DATA USED FOR MODEL CALCULATION.


OUTPUT #1
0.8
0.6
0.4
0.2
0
0
0.01
0.02
0.03
0.04
0.05
0.06
0.07

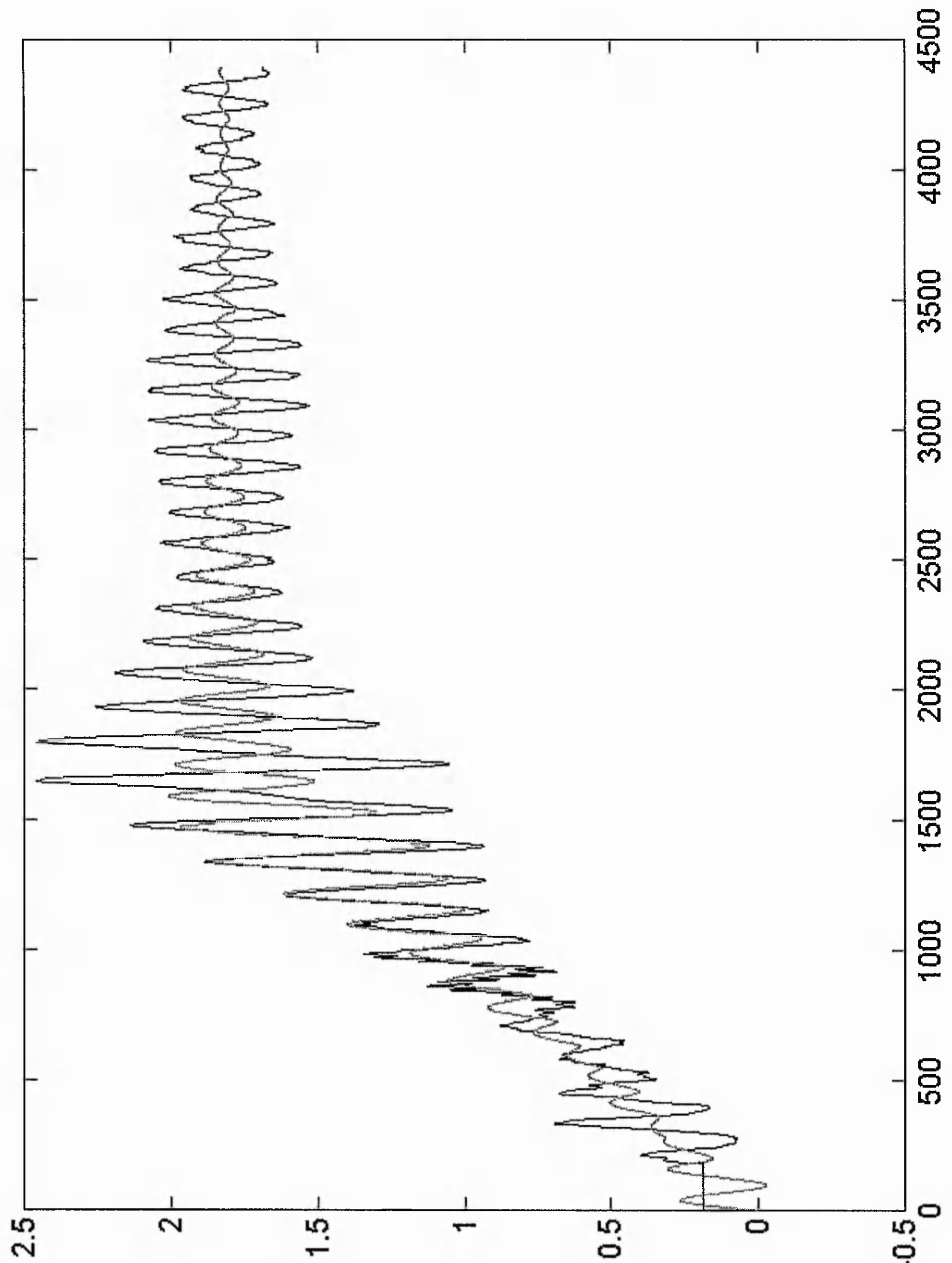

INPUT #1
0.7
0.6
0.5
0.4
0.3
0.2
0
0.01
0.02
0.03
0.04
0.05
0.06
0.07
0.08

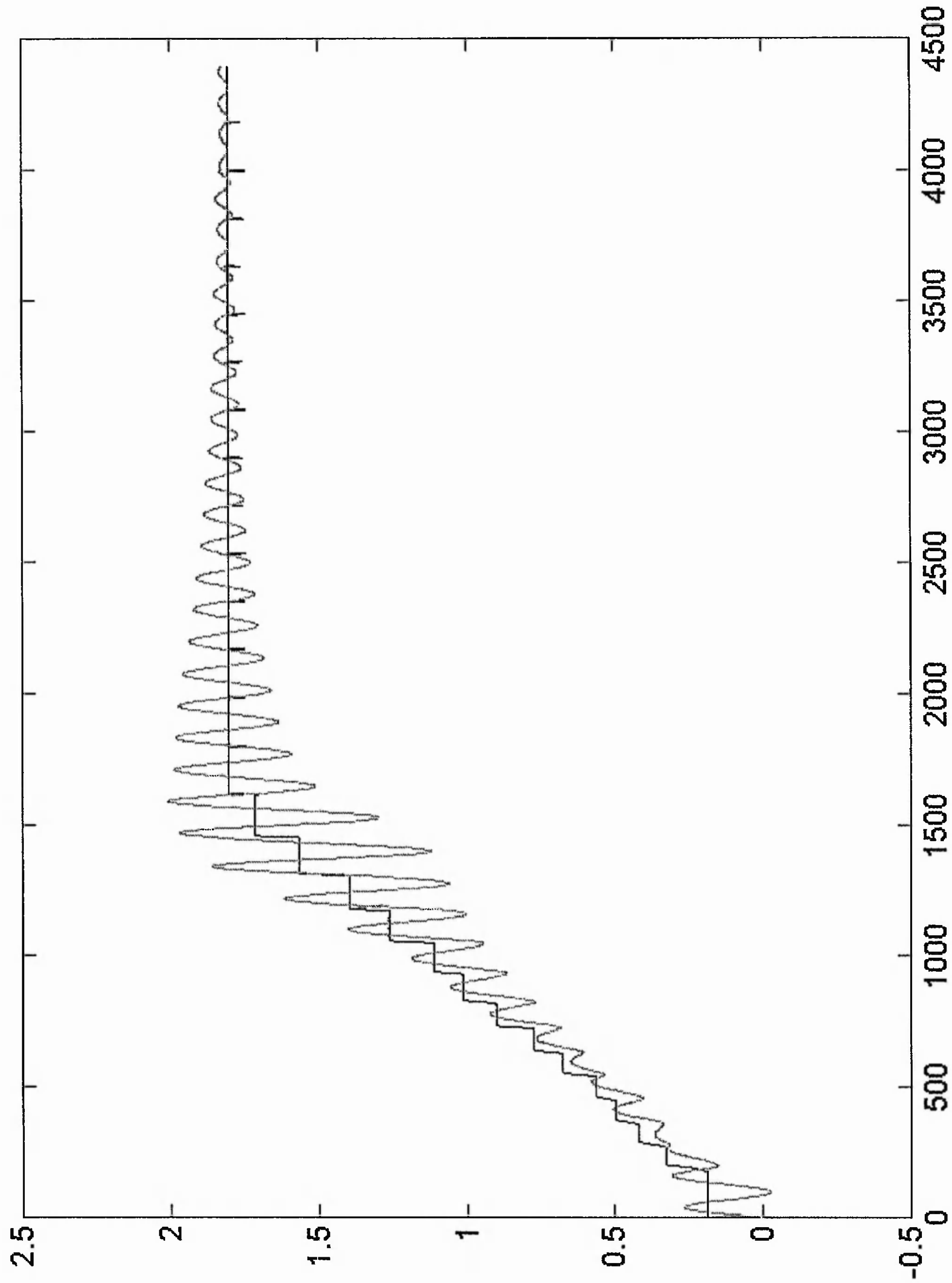
DATA USED FOR MODEL VALIDATION.



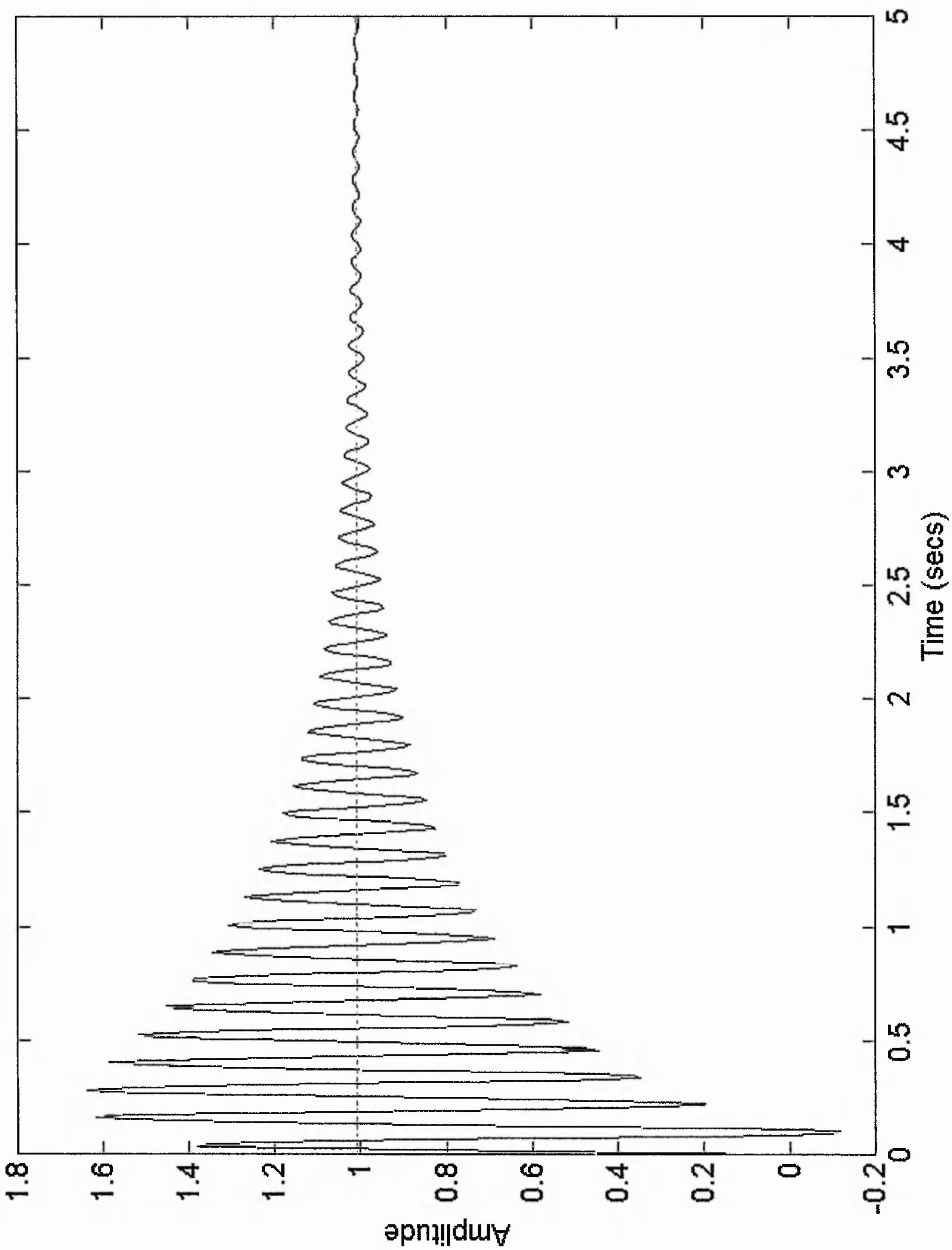
1
0.8
0.6
0.4
0.2
0
-0.2
-0.4
-0.6
-0.8
0
200
400
600
800
1000
1200

ACTUAL & SIMULATED OUTPUT.

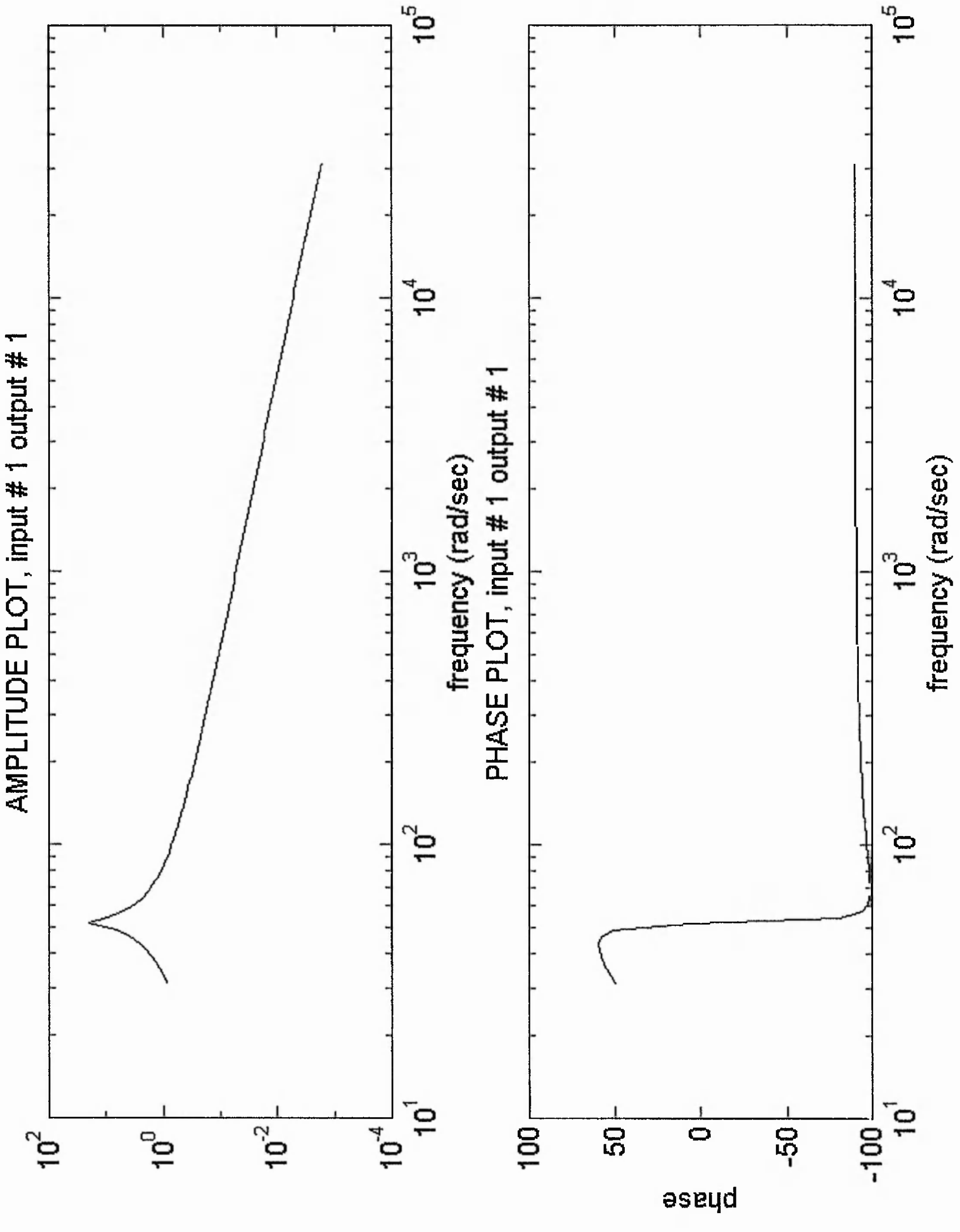

0
500
1000
1500
2000
2500
3000
3500
4000
4500
2.5
2
1.5
1
0.5
0
-0.5


2.5
2
1.5
1
0.5
0
-0.5
0
500
1000
1500
2000
2500
3000
3500
4000
4500

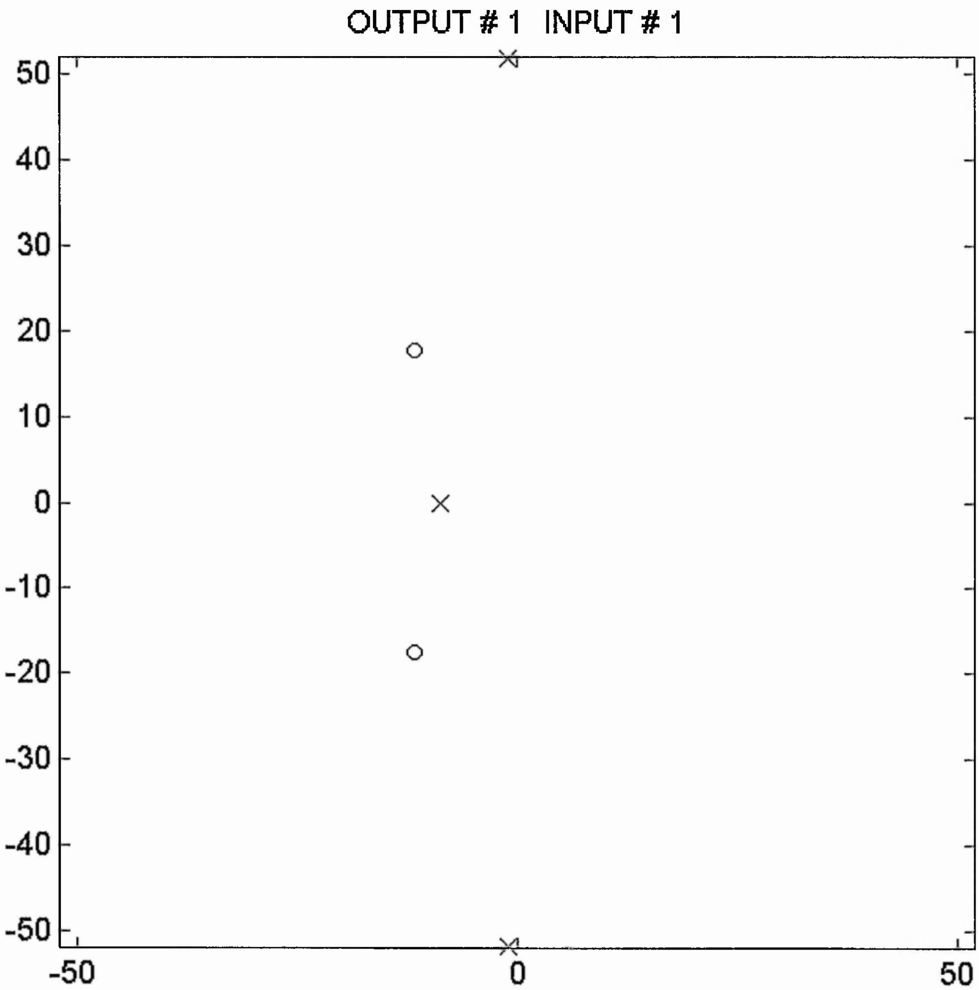
STEP RESPONSE.


Amplitude
Time (secs)
1.8
1.6
1.4
1.2
1
0.8
0.6
0.4
0.2
0
-0.2
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
5

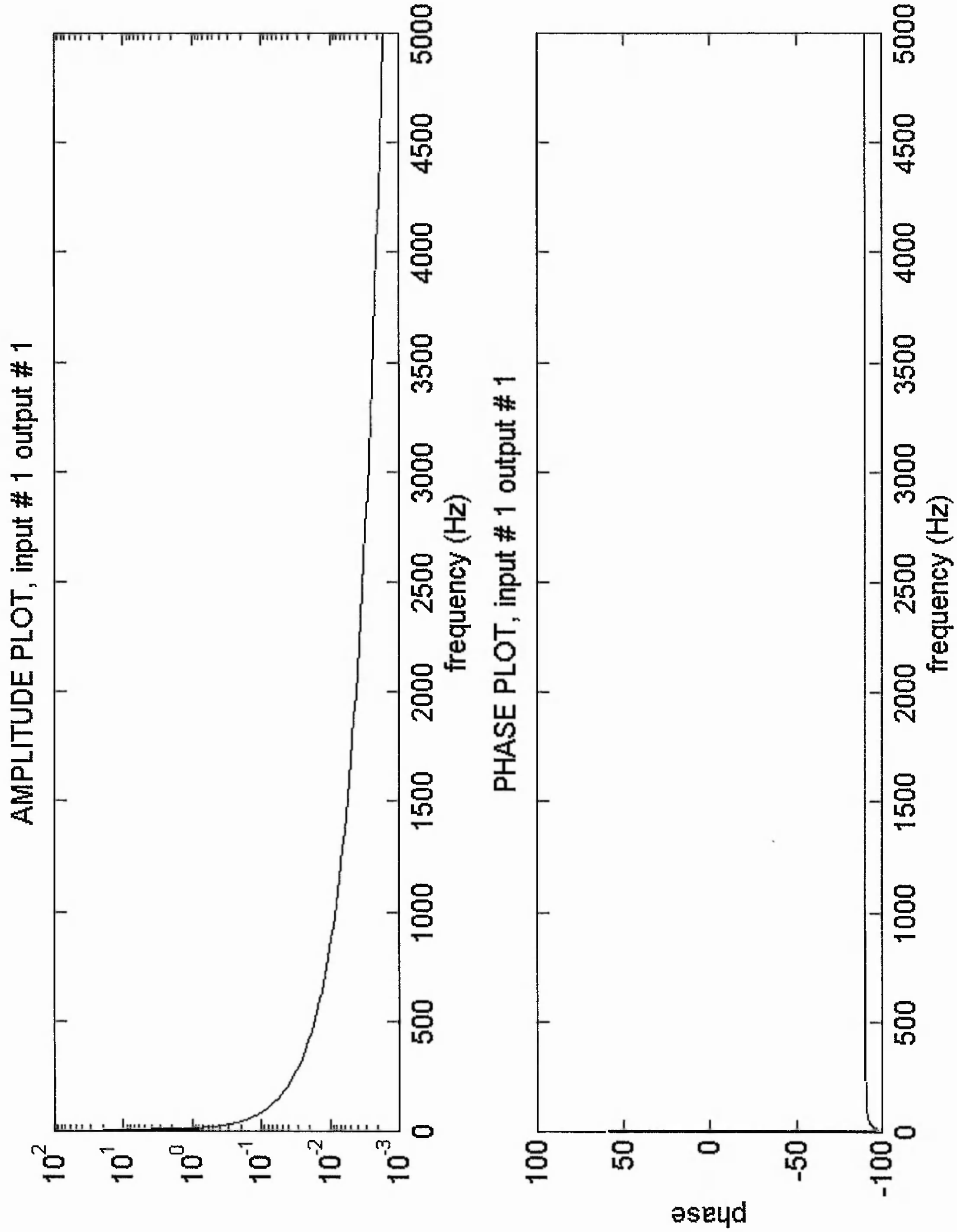
BODE PLOT.


AMPLITUDE PLOT, input # 1 output # 1
frequency (rad/sec)
PHASE PLOT, input # 1 output # 1
phase
frequency (rad/sec)

POLE & ZERO PLOT.


OUTPUT # 1 INPUT # 1
50
40
30
20
10
0
-10
-20
-30
-40
-50
-50
0
50

FREQUENCY PLOT.


AMPLITUDE PLOT, input # 1 output # 1
10^2
10^1
10^0
10^{-1}
10^{-2}
10^{-3}
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
frequency (Hz)
PHASE PLOT, input # 1 output # 1
phase
100
50
0
-50
-100
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
frequency (Hz)

MATLAB PROGRAM.

```
echo on
%
************************************************************************
******
% *                TRANSFER FUNCTION MATLAB PROGRAM                *
%
************************************************************************
******
% * AUTHOR : ADRIAN J. STOUT BEng.          DATE 3/7/96              *
%
************************************************************************
******
%
%
% load & separate the data.
%
%
load transfer.dat;
time = transfer(:,1);
input = transfer(:,2);
output = transfer(:,3);
%
%
% Plot the input & output signal.
%
%
echo off;
plot(time,input,'r-',time,output,'g--');
title('Velocity Profile');
xlabel('Velocity (M/m)');
ylabel('Time (Seconds)');
echo on;
%
```

```
%
%  Shown here is a plot of the input & output signal against time.
%  Press a key to model the data.
%
echo off;
pause;
echo on;
%
%           Select a portion of data to model.
%
in = input(1:600);
out = output(1:600);
z2 = [output(1:600) input(1:600)];
echo off;
idplot(z2,1:600,0.0001);
echo on;
%
%
%           Remove the over-lying trend & model.
%
%


z2 = dtrend(z2);
th =n4sid(z2,3,1,'trace');
th = sett(th,0.001);
u = dtrend(in);
ysim = idsim(u,th);
out = dtrend(out);
%
%
% Shown here is the portion of the signal that is used to model the system.
% Press a key to see how the model responds to the input signal.
%
echo off;
pause;
```

```
echo on;
%
%
% Plot the signal output against the model output.
% Obtain the state space model & transfer function.
%
%
%
echo off;
plot([out ysim]);
echo on;
modthc = thd2thc(th);
[a,b,c,d,k,x0] = th2ss(modthc);
[num,den] = th2tf(modthc);
%
%
% Shown here is the signal output and the output from the model (purple).
% Press a key to see the step response.
%
echo off;
pause;
%
step(num,den);
%
%
echo on;
%  Press a key to see the transfer function.
%
echo off;
pause

% Now shown is the transfer function

num
```

```
den

% Now perform CNCAC. load in another portion of data.
echo on;
%
%
% Now load in another section of data to validate model.
%
%
in = input(1000:2000);
out = output(1000:2000);
u = dtrend(in);
ysim = idsim(u,th);
out = dtrend(out);
%
%
% Press a key to view the actual output and the simulated output.
echo off;
pause;
plot([out ysim]);
%
echo on;
%
%
% Now load the complete data set and simulate the complete response.
%
%

in = input(1:4400);
out = output(1:4400);
u = dtrend(in);
ysim = idsim(in,th);
%out = dtrend(out);
% Press a key to see the output.
echo off;
```

```
pause;


plot([out ysim]);
echo on;
%
%
% Actual output verses Modeled output.
% Press a key to see the input verses the modeled output signal.
%
echo off;
pause;


plot([in ysim]);
echo on;
%
%         Adjust the theta format such that
%         Response reaches 100% of input.
%
msth =[0.0009 -0.0010  1.0000  1.0000  0       3.0000  9.0000  0       0;
     0.0009 19.9600  7.0000  8.0000 10.0000  5.0000 50.0000  2.0000  0;
     0       0       0       0       0       0       0       0       0;
     1.0047  0.0524  0.0282 -0.3052 -0.0102  0      -0.6662  0       1.0000;
    -0.0524  1.0022 -0.0157  0.1597  0.0782  0       0.0857  0       3.0000;
    -0.0126 -0.0158  0.9794  0.1535  0.2455  0       0.6378  0       0;
     0.0009  0       0       0       0       0       0       0       0];


%
% first column, sixth row changed from 0.0126 to 0.0096
% fourth column sixth row changed from 0.1453 to 0.1535
% Fine tune the model such that the simulated output reaches
% 100% of the input signal.
% Now load the complete data set and simulate the complete response.
%
%
in = input(1:4400);
```

```
out = output(1:4400);
u = dtrend(in);
ysim = idsim(in,msth);
%out = dtrend(out);
%
%   Press a key to see the output.
%
echo off;
pause;
plot([out ysim]);
echo on;
%
%
%  Actual output verses Modeled output.
%  Press a key to see the input verses the modeled output signal.
%
echo off;
pause;
plot([in ysim]);
echo on;
%
%
%  This model has a fit of 88%.
%  Shown below is the adjusted Transfer function.
%
[NUM,DEN] = th2tf(msth);

NUM


%
%       NUM =
%               1.0e+004 *
%                   0    0.0053    0.1237    2.3771
```

```
DEN

%      DEN =
%           1.0e+004 *
%            0.0001    0.0011    0.2702    2.3582
%
%
%
% Now obtain the Damping Factors and Natural frequencies
%
%
damp(DEN);
%
%        Eigenvalue       Damping       Freq. (rad/sec)
%
%       -1.1345 +51.7714i   0.0219          51.7839
%       -1.1345 -51.7714i   0.0219          51.7839
%       -8.7941             1.0000          8.7941
%
% Now plot the Bode plot.
%
[g,phiv] = th2ff(msth);
bodeplot(g);
%




%
% Press a key to plot the frequency plot.
echo off;
pause;
ffplot(g);
echo on;
%
```

```
%
% Press a key to see the pole and zero plot.
%
%

[zepo,k] = th2zp(msth);
zpplot(th2zp(msth));
[zeros,poles] = getzp(zepo);

zeros
%      zeros =

%          -11.6703 +17.6766i
%          -11.6703 -17.6766i
poles

%      poles =

%           -1.1345 +51.7714i
%           -1.1345 -51.7714i
%           -8.7941
```

```
%
%       Now feed in a new input signal, and plot.
pause
%
load newinput.m;
u = newinput(1:1900);
ysim = idsim(u,msth);
plot([u ysim]);

%
%  The end.
```

Appendix C.

Papers Published.

[1] STOUT A.J., ORTON P.A., THOMAS P.D.,
"STEPPER MOTOR CONTROL SYSTEM ANALYSIS & DESIGN"
Proceedings of the Circuits, Systems and Computers International Conference
Hellenic Naval Academy
Piraeus, Greece.
1996
ISBN : 960-8485-02-9

[2] STOUT A.J., THOMAS P.D., ORTON P.A.,
"IMPROVED CONTINUOUS PATH MOTION USING STEPPER MOTORS"
Proceedings of the 10th European Simulation Symposium & Exhibition.
The Nottingham Trent University,
Nottingham, United Kingdom.
1998
ISBN : 1-56555-154-0

[3] STOUT A.J., ORTON P.A., THOMAS P.D.,
"IMPROVING THE PERFORMANCE OF STEPPER MOTOR DRIVEN MACHINERY"
Proceedings of the International Conference on Condition Monitoring,
University of Wales Swansea,
Swansea.
1999
ISBN : 901892115

[4] STOUT A.J., ORTON P.A., THOMAS P.D.,
"IMPROVED ENERGY EFFICIENCY OF STEPPER MOTOR SYSTEMS"
Unpublished to date.

STEPPER MOTOR CONTROL SYSTEM ANALYSIS & DESIGN

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ABSTRACT

This paper describes research carried out, within the Department of Computing at the Nottingham Trent University, England into real-time machine control, namely the control of stepper motor driven machinery. Research in the past has led to the development of a new position sensor that may be utilized to provide condition monitoring of bearings. The research has continued into the development of a digital signal acquisition board that can be employed to obtain the results from the novel sensor to be processed by a host personal computer or by a digital signal processor. The novel sensor and digital data acquisition board can provide the user of computer numerical control equipment with invaluable data concerning machine position, path accuracy, and conditional monitoring of bearings.

The research team is also concerned with contour path accuracy of CNC machines when driven by stepper motor drives. The new sensor and acquisition system provides unique information concerning the machines performance, the research has taken the approach of studying more effective ways of providing control for stepper motor driven machinery. This paper documents the development of an experimental test platform with high precision sensors, The platform provides essential information concerning the control of stepper motor driven machinery that can be fed back into the system to provide adaptive control of the machinery. The paper will present the results obtained from the experimental test platform, revealing some of the problems associated with stepper motor control.

INTRODUCTION

Research into machine control normally revolves around the various aspects of motion generation and its instrumentation; of particular interest is the generation of smooth continuous paths. The investigation of such motion requires dynamic measurement of angular position as well as linear motion [1],[2]. Precision measurement at operating speeds requires the use of high speed data acquisition techniques. Data acquisition systems with a high bandwidth and a high resolution can be complex and costly. The Nottingham Trent University have developed a high bandwidth, high resolution digital data acquisition board, which uses a reciprocal timing architecture [3].

Computer Numerical control has traditionally used AC or DC servo motors for the drive system, since they both have a wide range of velocity and Torque ranges. The other kind of drive system in terms of electrical motors is the stepper motor. This motor moves in discrete steps when the relevant coils are energized with power. The stepper motor offers lower cost, and with advances in technology quite high performance. Commonly these motors are used for disk drives and other devices that need precision movement.

Previous investigations into stepper motor control by STEIGER have been concerned with the generation of smooth continuous path motion [4]-[6]. Sherkat & Steiger have demonstrated that problems may arise from the excitation of machine dynamics arising from the interaction of acceleration and interpolation. Steiger in particular has shown theoretically that the velocity profile on a slave axis may be very rough. The consequence of this is likely to be the justification for the fact that stepper motor driven machines rarely achieve their performance potential.

To build on the theoretical work conducted by Steiger an experimental test platform was developed[7]. The aim behind the development of this experimental test platform is to improve the performance of continuous path multi-axis stepper motor driven machinery. The platform will allow rapid analysis of stepper motor control techniques hence facilitate in the development of new techniques that will help overcome the rough motion problems arising from path generation.

STEPPER MOTOR CONTROL ANALYSIS

The stepping motion of the stepper motor drive can lead to machine vibrations, it can be shown that these vibrations propagate through the system to the end effector. These vibrations can result in a loss of accuracy, hence the preference by some users to use AC or DC servo motors.

One of the established ways of improving the accuracy of stepper motor machinery is to ensure that any machine vibrations are damped out by the mechanics of the machine itself.

In some cases torque dampers or frictional driven flywheels may be fitted to the drive. Most CNC drive trains can be adequately modeled as a low order system (a lead screw indirectly coupled to a stepper motor drive can typically be modeled as a third order system). Normally the CNC system will have sufficient mass to damp out any machine vibrations that exist . The system will also have inherent viscous & frictional damping. The end process of this leads to greater machine mass and damping, requiring the use of stepper motors with higher torque ratios and/or reduced velocity/acceleration whilst it is contour path following.

Stepper motors are driven by a digital pulse train. The velocity is dependent upon the frequency of this pulse train, the higher the frequency the greater the velocity. Many controllers give out stepped changes of pulse frequency , these steps forming a staircase effect when viewed as a velocity profile, as seen in figure 1. In effect the controller quantises the input velocity profile.

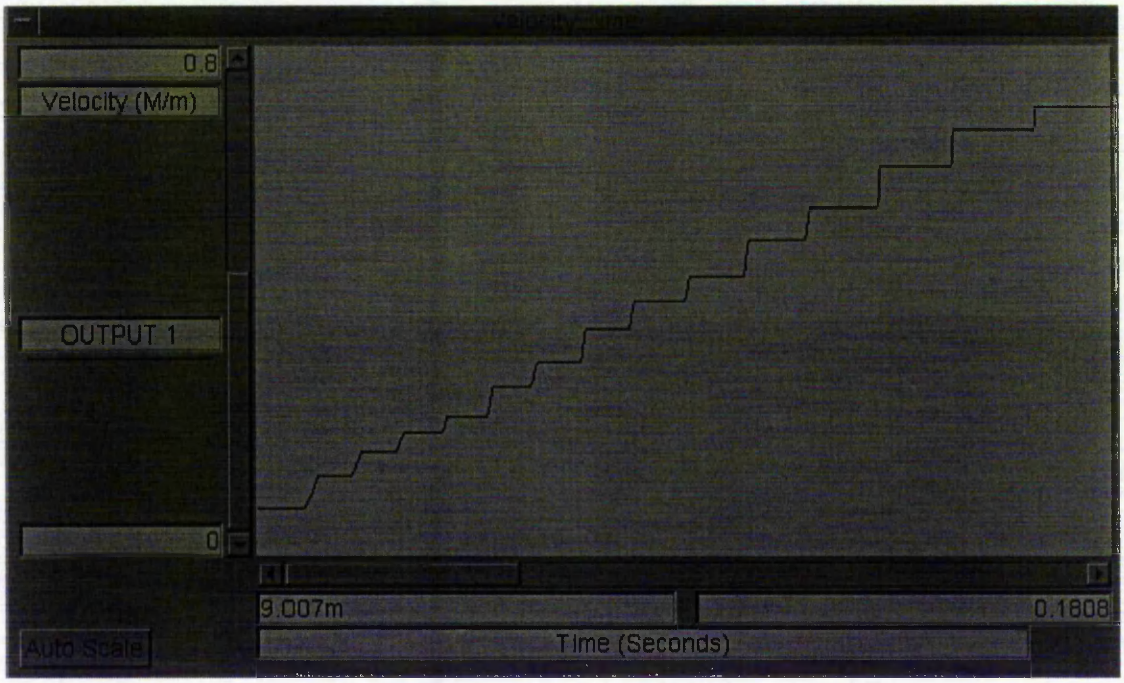


Figure 1. Velocity Profile of Motor Input Pulses.

Therefore in control terms the input to the system can then be viewed as a series of cascading step inputs, with resulting step response. Typically this step response will have a damping ratio less than 1 when the system is lightly loaded.

Research was conducted on various test machines to study their performance. The results showed that the systems would typically be developed such that it will be critically damped when a mass equal to a mid-range value of the maximum load, is placed upon the machining table. The desired mass would then have effect on the system in the form of added mass and added damping. The added damping represents the added frictional & viscous damping resulting from the mass exerting more pressure against the bearings contained in the slide table guideways & lead screw. The effect of this can be such that light work pieces will have inaccuracies in their surface due to an under damped system.

Obtained results have indicated that this overshoot in position would result in a rough surface due to overshoots in the contour path by up to 20-40 microns.

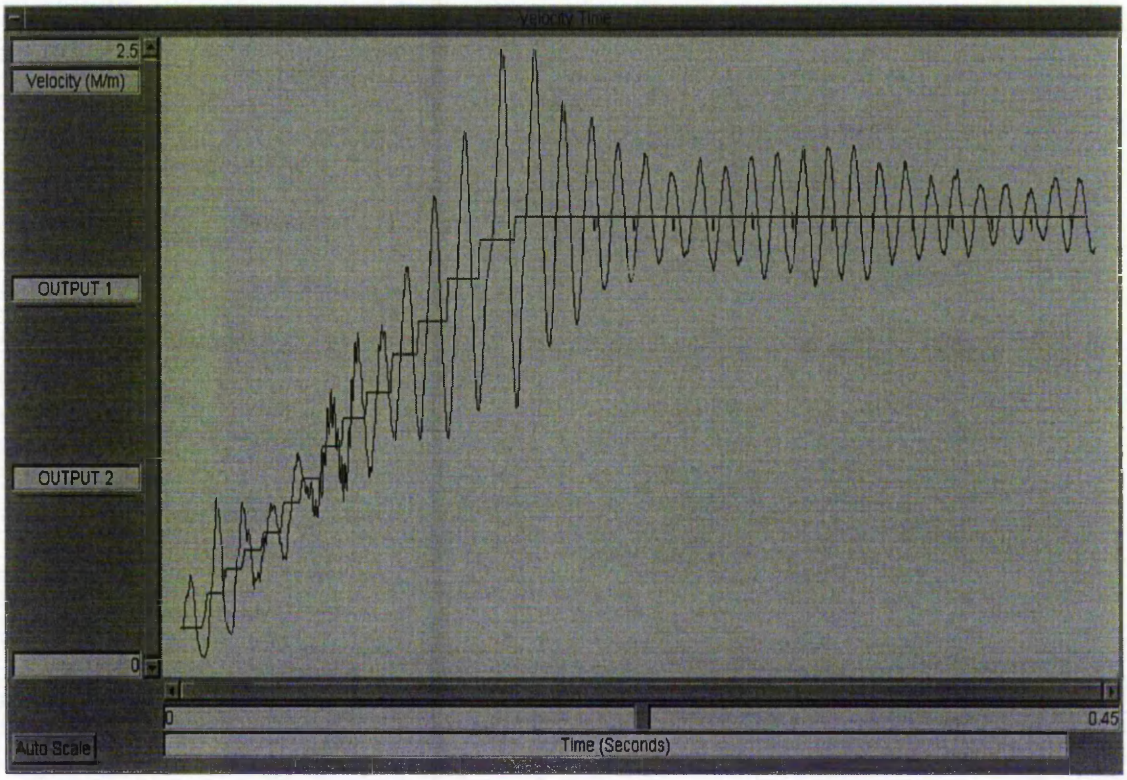


Figure 2. Velocity Profile of Input and Output Motion.

Typically for high precision, pieces of work would be machined on systems appropriate for their mass. This creates problems for users who need precision out of a general CNC machine, accepting a range of small batch work pieces with varying mass.

With intelligent machining techniques, based on our work it is the view of the researchers that these problems can be overcome. This would result in a precision machining system that would consist of an 'off the shelf' machining center and an intelligent adaptive controller.

This controller would measure the dynamics of the system and therefore adjust the input to the system to suit the machine, taking into account the age, condition and work piece.

The experimental Test Platform represents a step towards this goal, in the fact that the motor drive is deliberately poorly mounted to drive train, resulting in a sufficiently under damped response in most circumstances. The Velocity profile for the system showing both the input and resultant output waveform can be seen in figure 2. The step response for this system an under damped response, as expected.

The theory behind this is that by knowing how the system is going to react to the state inputs means that the control algorithm can produce suitable state inputs that will not result in overshoot in either velocity or path motion.

When the theory is validated by developing a novel control algorithm the system will be tested upon the experimental platform, since it would represent the worst case scenario for the controller.

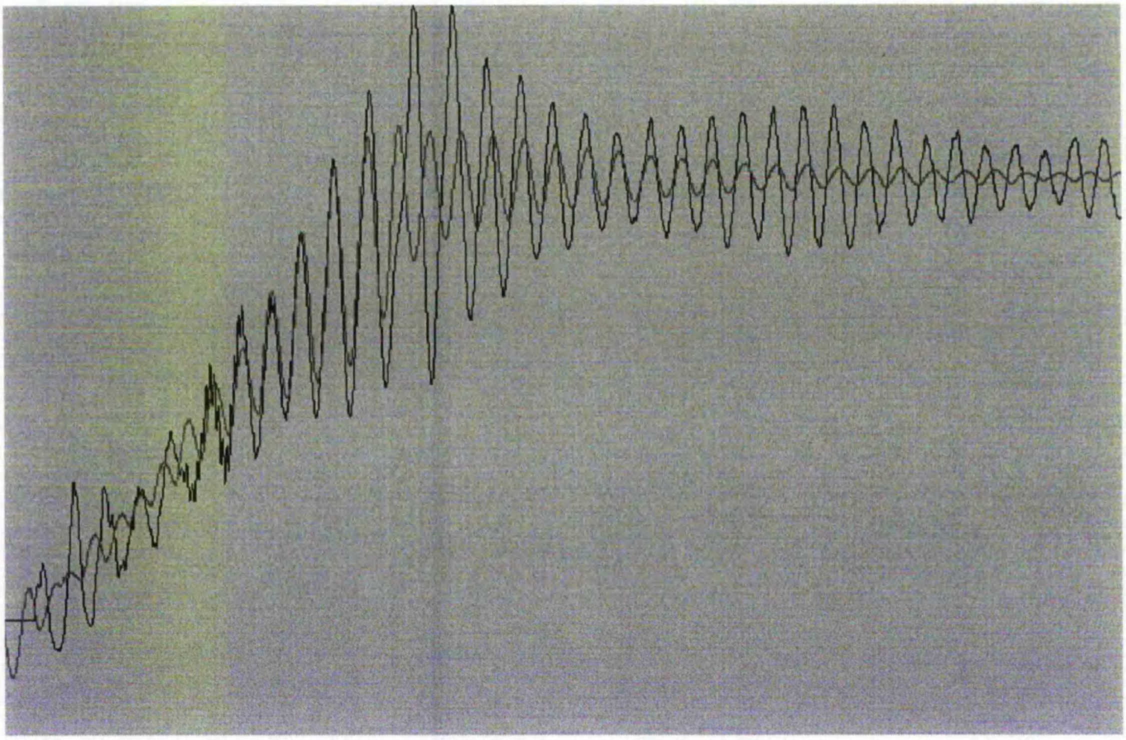
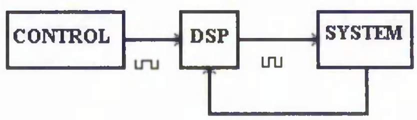


Figure 3. Simulated Velocity Profile Output vs. Actual Velocity Profile Output.

CONCLUSION

This paper briefly mentions a section of the research conducted at The Nottingham Trent University. The research is still in it's early stages whilst already providing good results and good future prospects. The results taken from the system were modeled as a third order system and when compared to the actual system show a good transient correlation The results of this can be seen in figure 3. The theory can now be tested by substituting the real input by a preprocessed input.

The development of the new stepper motor control techniques will take advantage of modern parallel DSP computing technology. In the past many controllers are restricted in performance because of processing limitations. Early DSP devices have successfully been used in machine control to add speed performance. DSP devices with their inherent internal parallelism and fast floating point arithmetic are well suited to tasks such as signal processing, digital filtering and interpolation. Initially a DSP will be used in-line with the motor pulses to condition monitor the input pulses so that the CNC system will receive a modified input signal. This signal will result in smooth contour path motion.



ACKNOWLEDGEMENTS

The Experimental Test Platform was developed with the help of AXIOMATIC Ltd. and HEIDENHAIN Ltd. who provided the drive systems and precision sensors. The authors would like to thank those people who generously gave their free help & to the E.P.S.R.C. who funded the research.

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[1] AYANDOKUN K., ORTON P.A., SHERKAT N., THOMAS P.D., "AN OPTICAL ROTARY MOTION SENSOR FOR REAL TIME CONDITION MONITORING OF ROTATING MACHINERY", International Conference on Condition Monitoring, Swansea, UK, 1994 .

[2] KADHIM A.H., BABU T.K.M., O'KELLY D., "MEASUREMENT OF STEADY-STATE AND TRANSIENT LOAD-ANGLE, ANGULAR VELOCITY AND ACCELERATION USING AN OPTICAL ENCODER", I.E.E.E. Transactions on Instrumentation and Measurement, USA, Vol 41, Issue 4, pp. 486-9, 1992 .

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[4] STEIGER W., "REAL-TIME CONTINUOUS PATH MOTION CONTROL FOR MACHINE TOOLS", PhD Thesis, Computing Department, The Nottingham Trent University, Nottingham England. 1994 .

[5] STEIGER W., SHERKAT N., "SMOOTH CONTINUOUS PATH MOTION THROUGH VARIABLE PULSE CONTROL", ASME Joint European Conference on Engineering Systems Design and Analysis, London, UK, 1994 .

[6] GAN J. G., WOO T. C., "ERROR FREE INTERPOLATION OF PARAMETRIC SURFACES" , Transactions of the A.S.M.E. Journal of Engineering for Industry, Vol. 114, pp. 271-276, August 1992.

[7] SZABO O., "INSTALLATION OF AN ULTRA PRECISION CNC LATHE LABORATORY, IT'S ACCURACY AND APPLICATION", Mechatronics, Vol. 3, No 2, pp. 215-219, 1993

IMPROVING CONTINUOUS PATH MOTION USING STEPPER MOTORS

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ABSTRACT

Stepping motors have become increasingly attractive in motion control applications, mainly due to the simplicity of their interface requirements and low costs. Unfortunately, the small but discrete steps of movement inherent in the motor design, combined with the inevitable quantisation of time leads to roughness of motion which may result in real limitations of system performance. Previous work has demonstrated the value of carefully chosen acceleration profiles for single axis motion. However work conducted within the real-time machine control group, Nottingham Trent University has shown such profiles to be of limited value in the case of multiple axis continuous path systems, where interaction of acceleration and interpolation results in further roughness of the path.

The work reported in this paper shows the results of precision measurements made on interpolating systems. Modelling and analysis of the data, within the HPVEE software package, demonstrates the validity of theoretical conclusions regarding rough motion. The work further shows the dynamic response of typical drive systems leading to velocity fluctuations and corresponding dynamic position errors. Such errors can be sufficient as to adversely affect cut quality on machine tools and to limit the performance of a machine with given motors and drive electronics.

It is proposed that a novel filter algorithm be employed such that path accuracy and cut quality can both be improved. Such a filter should also facilitate higher performance from a given motion system. Simulation results are presented showing the potential improvements to be made and the validity of the simulation model is demonstrated.

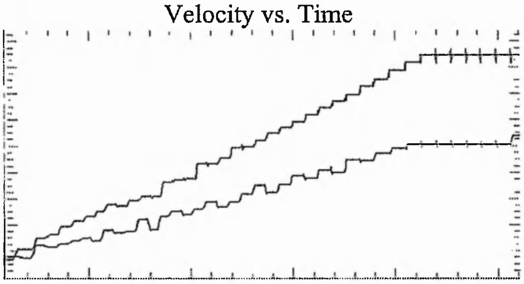
INTRODUCTION

Previous work in the field of stepper motor control by authors such as Palmin [1] and Acarnley[2] has shown the importance of acceleration profile shaping if satisfactory performance is to be achieved from stepping motor systems. Motors themselves and transmission systems have natural resonances and the significance of these can be reduced if the profile is suitably shaped. Unfortunately, this is of little benefit in multiple axes interpolating systems, such as machine tools, where simultaneous synchronised motion in several axes may be required. Steiger, et al.[3-4] have shown from a theoretical point of view how the interaction of any acceleration profile and interpolation may lead to rough motion. In particular it was found that the least dominant axis suffered from coarse velocity changes, these being developed by the action of synchronising the two axes. See Figure 1. In this paper, we show how these conclusions are substantiated in practice, how the motion may excite the drive system dynamics, resulting in unacceptably large velocity fluctuations and position errors.

Stepper motor systems are generally controlled by open loop electronics, where clock & direction signals are applied to appropriate counter and power electronic circuits in order for a sequence of energisation to be applied to the motor windings. It is unusual for position encoders to be used with stepper motors; they increase cost and do little to address the problems discussed in this paper.

In order to analyse the motion it is necessary to instrument a typical transmission system. An experimental system has been constructed and this is fitted with both rotary and linear encoders. A special purpose data logger has been developed within the group [5], allowing precision logging of the drive pulse trains and those gathered from the encoders. In order to allow maximum flexibility for subsequent analysis, the precise timing of each pulse edge transition is recorded. This minimises the significance of measurement quantisation, since the clock speed can be made significantly high in relation to the pulse rate.

Velocity vs. Time



Y axis 0 – 2 m/min. X axis 0 – 0.26 Seconds

Figure 1. Input velocity of X & Y-axis

STEPPER MOTOR CONTROL ANALYSIS

One of the established ways of improving the accuracy of stepper motor machinery is to ensure that any machine vibrations are damped out by the mechanics of the machine itself. The system will have inherent viscous & frictional damping. Normally the CNC system will have sufficient mass to damp out any machine vibrations that exist. In some cases torque dampers or frictional driven flywheels may be fitted to the drive to provide extra damping.

The disadvantage with this type of vibration control is that it generally leads to greater machine mass to damp out resonant frequencies. This in turn increases the viscous friction due to the added pressure on the guide rails. To compensate either higher torque stepper motors are needed or the overall performance of the machine needs to be down graded. An alternative method currently being researched and implemented in industry is to electronically damp out these vibrations.

Stepper motors are driven by a digital pulse train. The rotational velocity of the spindle is dependent upon the frequency of this pulse train, the higher the frequency the greater the velocity, see figure 2. This figure shows the pulse trains captured using a logic analyser.

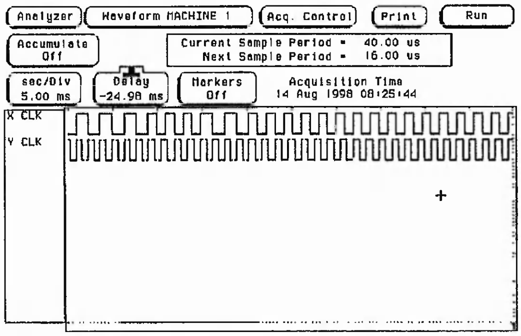
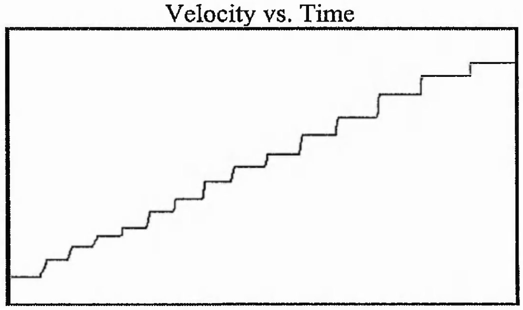


Figure 2. Stepper motor pulse trains for X & Y axis

Many controllers give out stepped changes of pulse frequency, these steps forming a staircase effect when viewed as a velocity profile, as seen in figure 3. The steps shown are due to the large interpolation step size used by this example. The graph in figure 3 shows the acceleration phase of a velocity profile, accelerating to a velocity of 0.8 m/min.

In effect the controller quantises the demand velocity profile. Therefore in control terms the input to the system can then be viewed as a series of cascading step inputs, with resulting step response. Typically this step response will be under-damped in nature, when the system is lightly loaded.

Velocity vs. Time



Y axis 0 – 1 m/min. X axis 0 – 0.2 Seconds

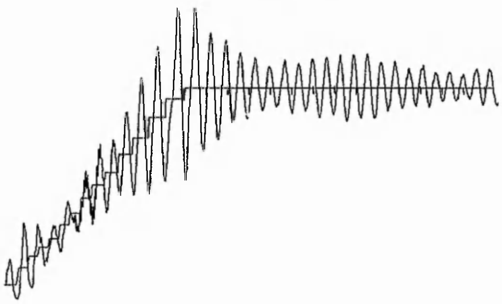
Figure 3. Velocity Profile of Controller Output Pulses.

In the path-planning world the path can be interpolated to within one unit corresponding to a single stepper motor pulse, or a unit representing many pulses. In the physical world many factors contribute to the actual path. Errors can be introduced into the system from several sources, including the following;

- Rounding errors in the path planning algorithm, or Interpolation algorithm.
- Inadequacies with the Interpolator algorithm, causing misalignment, or speed changes as the tool machines around arcs, or circles.
- Machine vibrations due to resonance effects, chatter, bearing clearances.
- Inadequacies with motion control system or drive system, due to component tolerances or control/drive algorithm.

If the velocity profile of figure 3 is implemented on the lightly damped test rig the velocity of the machine can be seen to fluctuate as seen in figure 4. The graph in figure 4 shows the demand and output response of a velocity profile, accelerating to a velocity of 2 M/min. The measured velocity is shown to fluctuate , these fluctuations forming the dynamic response to the roughness of motion. These fluctuations in velocity correspond to a path error in the order of 30 microns.

Velocity vs. Time

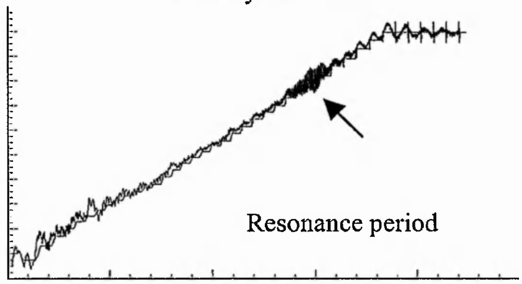


Y axis 0 – 2.5 M/min. X axis 0 – 0.45 Seconds

Figure 4. Velocity Profile of Input and Output Motion

This stepped input is modelled to be a representation of the disturbances that the stepper motor may experience whilst following a contour path, as shown in figure 1. A representation of a normal velocity profile is shown in figure 5. The graph in figure 4 shows a velocity profile, accelerating to a velocity of 2 m/min.

Velocity vs. Time

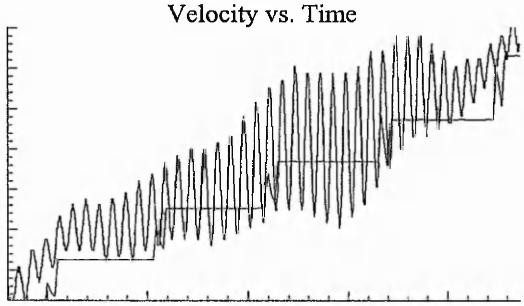


Y axis 0 – 2.2 m/min. X axis 0 – 0.25 Seconds

Figure 5. Measured Normal Slew Velocity Profile

Whilst the amplitude of the velocity fluctuations is small, and being of no serious detriment to the machines performance, closer inspection shows there to be a resonance period. Upon analysis this resonance period corresponds to the natural frequency of the pulley belts used. An expanded view is shown in figure 6. The range of the x-axis of the graph shown in figure 6 is 25 milliseconds with each grating representing 1 millisecond. The grating on the Y-axis represents 0.01 m/min within a velocity range of 1.4 – 1.7 m/min.

Velocity vs. Time



Y axis 1.4 – 1.7 m/min.
X axis 0.135 – 0.16 Seconds

Figure 6 Exploded view of resonance period from figure 5.

MODELLING DATA

The use of data modelling and representation tools such as Hewlett Packard's Visual Engineering Environment (HP VEE) and Mathworks MATLAB allow the design engineer to analyse the system in question in real-time. HP VEE is a graphical programming language for creating test systems and for solving engineering problems. The user can select from a variety of standard mathematical formulae, and special user supplied functions can easily be added. Shown in figure 7 is the block diagram for generating a FFT from the signal shown in figure 6.

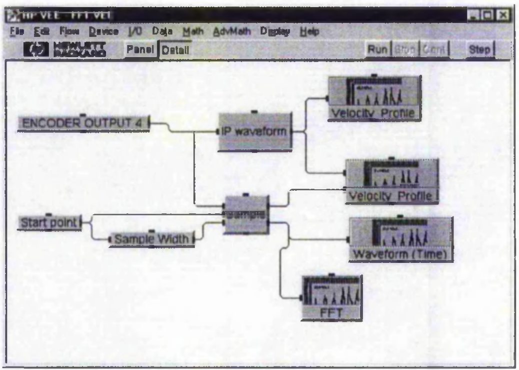


Figure 7 HP VEE Interface.

Previous research in the area of velocity profile generation has indicated that a parabolic velocity profile allows maximum utilisation of stepper motor torque [1, 2]. To confirm the findings of Palmin et al, and to investigate the use of a parabolic velocity profile for a continuous path multi-axis machine, a model of the machine was developed.

Analysis of any control system can be achieved by one of two ways. The First method is to build a mathematical model of the system in the form of differential equations, then apply Laplace transforms to the model to determine information such as the pole placement etc. This typically involves prior knowledge of the systems parameters, often involving considerable experimentation. The alternative method is to use systems identification. Systems identification is the process of approximating the systems parameters for a given computer model. In the Matlab environment the computer can be used to assist in the prediction of model order. The parameters are then determined by applying a least squares approach.

Researchers such as Kenjo [6] have shown how the electrical characteristics of a stepper motor can be modelled. For the simulation of the machine dynamics in our case a low order model was chosen. Most CNC drive trains can be adequately modelled as a low order system (a lead screw indirectly coupled to a stepper motor drive can typically be modelled as a third order system).

In order to validate the Matlab approach, the mechanics of the test rig were analysed. From the construction of the test rig it was determined that a model of approximately third order would be sufficient to model the system. The model can be seen in figure 8.

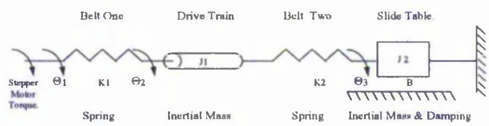
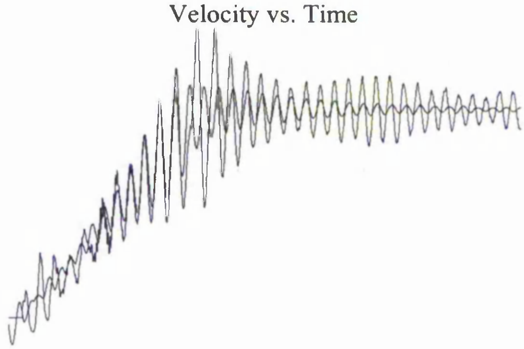


Figure 8 Analytical Model of Test Rig.

The captured data represented in figure 4, was then used as the model reference input for the Matlab environment. Systems identification was used to identify the unknown parameters. The mathematical model of the system, figure 8, in this case was used to provide confidence in the model determination. The input was then fed into the model, which produced excitations similar to those experienced on the real machine. Shown in figure 9 is a comparison between the actual machine output, taken from the results in figure 4 and the simulation output.



Y axis 0 – 2.5 M/min. X axis 0 – 0.45 Seconds

Figure 9. Simulated Velocity Profile Output vs. Actual Velocity Profile Output.

The reader can see from figure 9 that the two outputs do differ slightly, but on the whole the simulation output models the general behaviour of the machine adequately. This is the main point; the model was developed to be a generalised model of the behaviour of the machine, a task that can be completed in real-time. An exact model would take a longer time to compute and would require additional parameters to be determined.

SIMULATION

Once a model has been established it is possible to produce a simulator of the modelled system. This is conventionally achieved by taking the Laplace transform and converting it to a Z transform by the means of a Bi-linear transformation. From the Z transform it is a small step to convert the transform into a difference equation that can directly produce a numerical simulation. This standard method is embodied within the simulink toolbox for Matlab. The author used this toolbox to build a simulator.

In the same way the simulator can be built to simulate a production CNC machine such that otherwise unobtainable information, could be determined. The simulation of the system was used to confirm Palmin's earlier results involving analogue-based ramp generation of a parabolic velocity profile. The author confirmed the findings of Palmin et al, concerning the maximum utilisation of torque by the use of a parabolic velocity profile.

The results obtained from the simulation were used to prototype a digital based velocity ramp generator involving a digital signal processor. This generator was used to confirm the findings developed from the simulator. When results where collected from the test rig they showed that care must be taken with the generation of a parabolic velocity profile. The parabolic velocity profile advances very slowly through the range of frequencies from the starting velocity through to the maximum velocity. This means that if there are any natural frequencies within this range then they will resonate with a duration dependant upon the acceleration slope of the velocity profile. Therefore the acceleration slope should be made as short as the machine would allow. This again is likely to cause problems with multi-axis machinery.

Many authors recommend using a two-stage interpolator, the reasoning behind this is to obtain better interpolation for the given time constraint. This is a good option if we are considering the path-planning environment but it fails to correct the velocity fluctuations in the physical world. One answer is to employ a novel filter algorithm such that path accuracy and cut quality can both be improved. Such a filter should also facilitate higher performance from a given motion system. This additional stage would smooth the output velocity in a parabolic manner. A conceptual diagram of the proposed system is shown in figure 10. The system would measure the input pulse stream and then after passing through a software algorithm, the modified pulse stream would be regenerated. The regenerated pulse length will change to reflect the smoothed velocity profile.

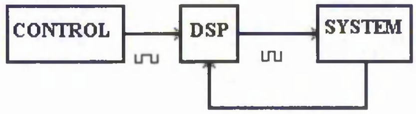


Figure 10. Proposed System

Through the use of computer simulation the filtering technique has been found to be effective in eliminating the velocity fluctuations. The output from the computer simulation with the modified DSP input can be seen in figure 11. The use of the simulation model provides evidence towards the potential performance benefits of such a system. The next step is to implement the algorithm on the DSP and test the algorithm on the test rig. The algorithm will be evaluated in terms of performance improvements, this being reported in a future paper.

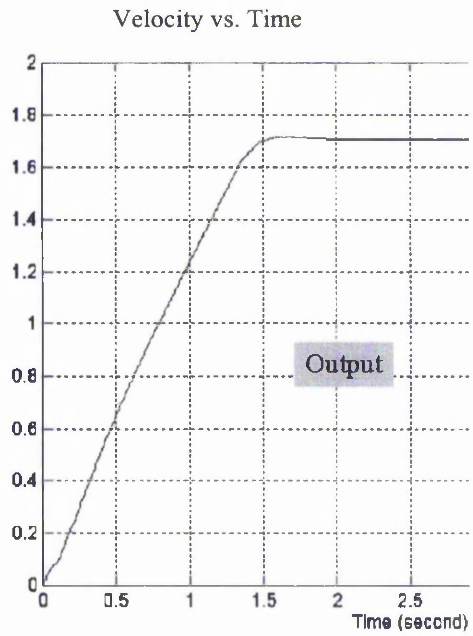


Figure 11. Output from model with proposed system

CONCLUSION

From the results presented it can be seen that the previous assumptions of rough motion in interpolating systems were valid. Further because the damping of mechanics may be light, excitation of dynamics may occur giving rise to significant position errors. The simulation shows that the performance of an experimental test machine can be improved through the addition of a filter stage to the interpolator. The improvements offered by the filter technique are better cut quality and higher performance, in effect high quality, faster production for a given cost. Further work is planned to confirm the validity of the filter technique on the experimental test rig and also on a production machine.

ACKNOWLEDGEMENTS

The Experimental Test Platform was developed with the help of AXIOMATIC TECHNOLOGY Ltd. and HEIDENHAIN Ltd. who provided the drive systems and precision sensors. The authors would like to thank those people who generously gave their free help & to the E.P.S.R.C. who funded the research.

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Improving the Performance of StepperMotor Driven Machinery

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Abstract

Stepping motors have become increasingly attractive in motion control applications, mainly due to the simplicity of their interface requirements and low costs. Unfortunately, the small but discrete steps of movement inherent in the motor design, combined with the inevitable quantisation of time leads to roughness of motion which may result in real limitations of system performance. Previous work has demonstrated the value of carefully chosen acceleration profiles for single axis motion. However work conducted within the real-time machine control group, Nottingham Trent University has shown such profiles to be of limited value in the case of multiple axis continuous path systems, where interaction of acceleration and interpolation results in further roughness of the path.

The work reported in this paper shows the results of precision measurements made on interpolating systems. The work further shows the dynamic response of typical drive systems leading to velocity fluctuations and corresponding dynamic position errors. Such errors can be sufficient as to adversely affect cut quality on machine tools and to limit the performance of a machine with given motors and drive electronics.

It is proposed that a novel filter algorithm be employed such that path accuracy and cut quality can both be improved. Such a filter should also facilitate higher performance from a given motion system. Experimental results are presented showing the potential improvements to be made, and the validity of the proposed solution is demonstrated. This is the third paper in a series that explores the idea of improving stepper motor driven machinery whilst controlled in an open loop continuous path manner.

Key Words: Stepper motors, improved performance, continuous path, contouring, performance monitoring, C.N.C. machinery.

Introduction

Previous work in the field of stepper motor control by authors such as Palmin [1] and Acarnley[2] has shown the importance of acceleration profile shaping if satisfactory performance is to be achieved from stepping motor systems. Motors themselves and transmission systems have natural resonances and the significance of these can be reduced if the profile is suitably shaped. Unfortunately, this is of little benefit in multiple axes interpolating systems, such as machine tools, where simultaneous synchronised motion in several axes may be required. In contrast to point to point systems, the objective for continuous path systems is not only to move from point A to B but also the precise path taken to reach point B.

Steiger, et al.[3-4] have shown from a theoretical point of view how the interaction of any acceleration profile and interpolation may lead to rough motion. In particular it was found that the least dominant axis suffered from coarse velocity changes, these being developed by the action of synchronising the two axes. See Figure 1. In this paper, we show how these conclusions are substantiated in practice, how the motion may excite the drive system dynamics, resulting in unacceptably large velocity fluctuations and position errors.

Stepper motor systems are generally controlled by open loop electronics, where clock & direction signals are applied to appropriate counter and power electronic circuits in order for a sequence of energisation to be applied to the motor windings. It is unusual for position encoders to be used with stepper motors since they increase cost and complexity to the system.

In order to analyse the motion it is necessary to instrument a typical transmission system. An experimental system has been constructed and this is fitted with both rotary and linear encoders. A special purpose data logger has been developed within the group [5], allowing precision logging of the drive pulse trains and those gathered from the encoders. In order to allow maximum flexibility for subsequent analysis, the precise timing of each pulse edge transition is recorded. This minimises the significance of measurement quantisation, since the clock speed can be made significantly high in relation to the pulse rate.

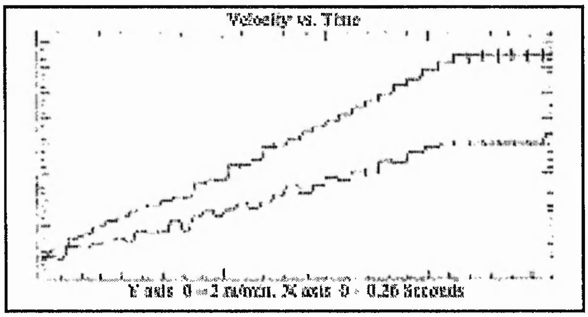

Figure 1. Input velocity of X & Y axis

Stepper Motor Control Aanalysis

One of the established ways of improving the accuracy of stepper motor machinery is to ensure that any machine vibrations are damped out by the mechanics of the machine itself. The system will have inherent viscous & frictional damping. Normally the CNC system will have sufficient mass to damp out any machine vibrations that exist. In the past torque dampers or frictional driven flywheels may have been fitted to the drive to provide extra damping. These devices are commonly found in the American market.

The disadvantage with this type of vibration control is that it generally leads to greater machine mass to damp out resonant frequencies. This in turn increases the viscous friction due to the added pressure on the guide rails. To compensate either higher torque stepper motors are needed or the overall performance of the machine needs to be down graded. An alternative method currently being researched and implemented in industry is to electronically damp out these vibrations. This technique is increasingly being used by design engineers in the UK & Europe. Electronic damping comes in many shapes and forms; the most common is in the form of an advanced drive amplifier which condition monitors the current pulses, another sort is a stepper motor which is filled by a viscous liquid, thereby simulating the action of a viscous flywheel.

Stepper motors are driven by a digital pulse train. The rotational velocity of the spindle is dependent upon the frequency of this pulse train, the higher the frequency the greater the velocity, see figure 2. This figure shows the pulse trains captured using a logic analyser.

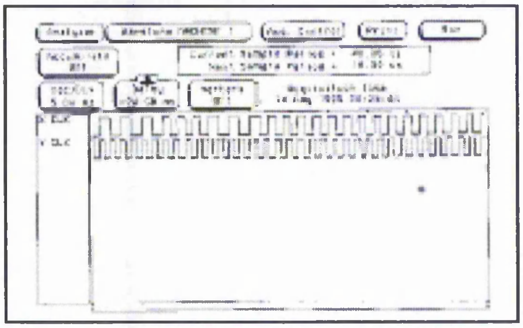

Figure 2. Stepper motor pulse trains for X & Y axis

For continuous path motion the movement upon the axes needs to be synchronised. The simplest way of achieving this is to move a set interpolated distance at a set velocity, and then synchronise all the axes together. This means that the controller gives out stepped changes of pulse frequency, these steps forming a staircase effect when viewed as a velocity profile, as seen in figure 3. The steps shown are due to the large interpolation step size used by this example. The graph in figure 3 shows the acceleration phase of a velocity profile, accelerating to a velocity of 0.8 m/min.

In effect the controller quantises the demand velocity profile. Therefore in control terms the input to the system can then be viewed as a series of cascading step inputs, with resulting step response. Typically this step response will be under-damped in nature, when the system is lightly loaded.

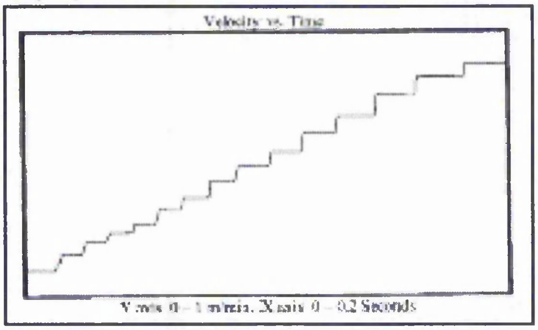

Figure 3. Velocity Profile of Controller Output Pulses.

In the path-planning world the path can be interpolated to within one unit corresponding to a single stepper motor pulse, or a unit representing many pulses. In the physical world many factors contribute to the actual path. Errors can be introduced into the system from several sources, including the following;

- Rounding errors in the path planning algorithm, or Interpolation algorithm.
- Inadequacies with the Interpolator algorithm, causing misalignment, or speed changes as the tool machines around arcs, or circles.
- Machine vibrations due to resonance effects, chatter, bearing clearances.
- Inadequacies with motion control system or drive system, due to component tolerances or control/drive algorithm.

Taken from the earlier papers in the series if the velocity profile of figure 3 is implemented on the lightly damped test rig the velocity of the machine can be seen to fluctuate as seen in figure 4. The graph in figure 4 shows the demand and output response of a

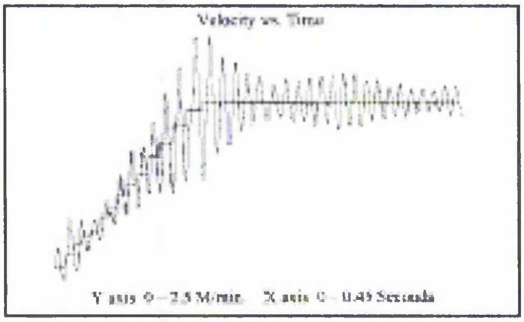

Figure 4. Velocity Profile of Input and Output Motion

velocity profile, accelerating to a velocity of 2 M/min. The measured velocity is shown to fluctuate , these fluctuations forming the dynamic response to the roughness of motion. These fluctuations in velocity correspond to a path error in the order of 30 microns.

This stepped input is modelled to be a representation of the disturbances that the stepper motor may experience whilst following a contour path, as shown in figure 1. A representation of a normal velocity profile is shown in figure 5. The graph in figure 4 shows a velocity profile, accelerating to a velocity of 2 m/min.

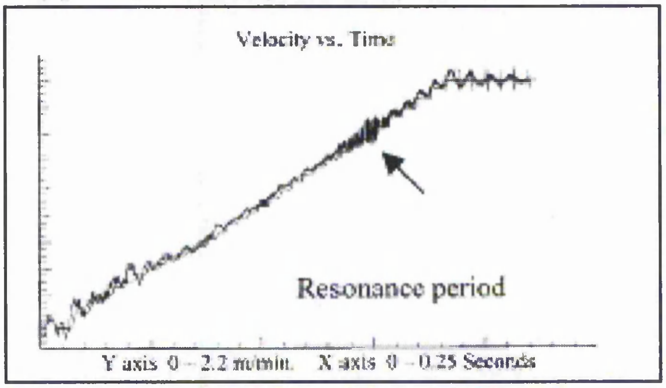

Figure 5. Measured Normal Slew Velocity Profile

Whilst the amplitude of the velocity fluctuations is small, and being of no serious detriment to the machines performance, closer inspection shows there to be a resonance period. Upon analysis this resonance period corresponds to the natural frequency of the pulley belts used. An expanded view is shown in figure 6. The range of the x-axis of the graph shown in figure 6 is 25 milliseconds with each grating representing 1 millisecond. The grating on the Y-axis represents 0.01 m/min within a velocity range of 1.4 – 1.7 m/min.

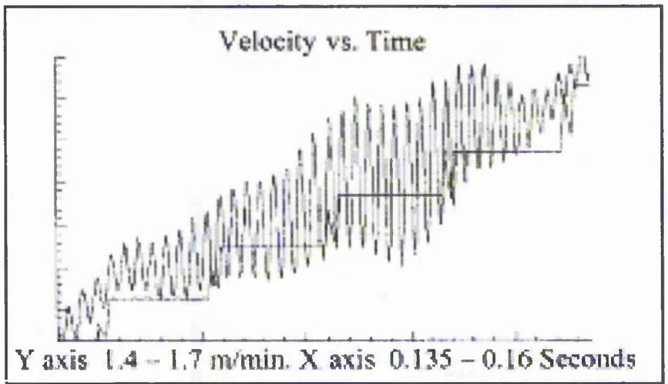

Figure 6 Exploded view of resonance period from figure 5.

Modelling Data

Previous research in the area of velocity profile generation has indicated that a parabolic velocity profile allows maximum utilisation of stepper motor torque [1, 2]. To confirm the findings of Palmin et al, and to investigate the use of a parabolic velocity profile for a continuous path multi-axis machine, a model of the machine was developed.

Researchers such as Kenjo [6] have shown how the internal electrical characteristics of a stepper motor can be modelled by a fourth order model. For the simulation of the machine dynamics in our case a low order model was chosen. Most CNC drive trains can be adequately modelled as a low order system (a lead screw indirectly coupled to a stepper motor drive can typically be modelled as a third order system).

Once a model has been established it is possible to produce a simulation of the modelled system. This is conventionally achieved by taking the Laplace transform and converting it to a Z transform by the means of a Bi-linear transformation. From the Z transform it is a small step to convert the transform into a difference equation that can directly produce a numerical simulation. The simulation of the system was used to confirm Palmin's earlier results involving analogue-based ramp generation of a parabolic velocity profile. The author confirmed the findings of Palmin et al, concerning the maximum utilisation of torque by the use of a parabolic velocity profile.

The results obtained from the simulation were used to prototype a digital based velocity ramp generator involving a digital signal processor. This generator was used to practically confirm the findings developed from the simulator. When results where collected from the test rig they showed that care must be taken with the generation of a parabolic velocity profile. The parabolic velocity profile advances very slowly through the range of frequencies from the starting velocity through to the maximum velocity. This means that if there are any natural frequencies within this range then they will resonate with a duration dependant upon the acceleration slope of the velocity profile. Therefore the acceleration slope should be made as short as the machine would allow. This again is likely to cause problems with multi-axis machinery.

Proposed Solution

Many authors recommend using a two-stage interpolator, the reasoning behind this is to obtain better interpolation for the given time constraint. This is a good option if we are considering the path-planning environment but it fails to correct the velocity fluctuations in the physical world.

One answer is to employ a novel filter algorithm such that path accuracy and cut quality can both be improved. Such a filter should also facilitate higher performance from a given motion system. This additional stage would smooth the output velocity in a parabolic manner. A conceptual diagram of the system is shown in figure 7. The system would measure the input pulse stream and then after passing through a software algorithm, the modified pulse stream would be regenerated. The regenerated pulse length will change to reflect the smoothed velocity profile. This stage is can be viewed as a filter only in the broadest sense of the word. Conventionally digital filtering works by taking a measurement of the quantity in question at discrete time intervals. The proposed method is radically different since it inverts this whole concept, rather than count units per time period it counts clock pulses per unit measured and the smoothing algorithm is implemented.

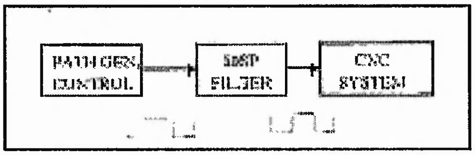

Figure 7. Proposed System

Through the use of first computer simulation and then practical implementation the filtering technique has been found to be effective in eliminating the velocity fluctuations. The normal interpolated output from the controller used can be seen in figure 8, showing the standard velocity fluctuation experienced on an under-damped CNC machine. Figure 9 shows the input and output response when the novel digital filter is used. The peak velocity deviation of the normal output response is 0.158 m/min. with a nominal velocity of 0.5 M/min. The peak deviation of the output using the filter is 0.046m/min. this level representing the background noise level.

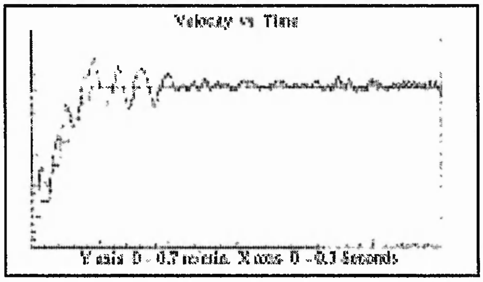

Figure 8 Output Before DSP filter.

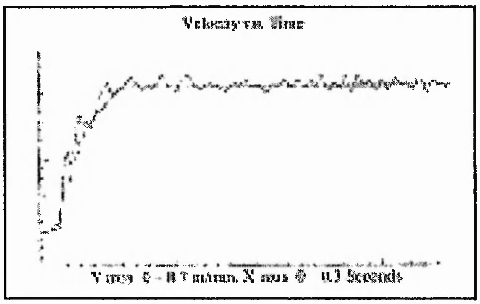

Figure 9 Output After DSP filter.

The use of the simulation model provides evidence towards the potential performance benefits of such a system. The next step is to measure the effectiveness of the filtering solution in terms of the improvement to machine performance. The performance of a given stepper motor is normally assessed by analysing it's speed/torque characteristic in the form of a curve on a velocity/torque graph. To utilise the graph the design engineer determines the total load on the system for a given acceleration and reads off the maximum velocity, taking into account a safe region of about 20%. This total load is the combination of frictional and inertial loads. Frictional load predominates in determining the necessary power for constant speed operation. Inertial load defines the amount of additional torque required to start and reach full speed within a specified time (in the process compensating for frictional load), and also to stop (conversely, aided by frictional load). In simple speed control applications, frictional load may be the most important. In position-control applications that require numerous start/stop operations and high system resolution, inertial load becomes the dominant factor. Therefore two curves are normally plotted on this graph, a stop/start curve and a slew curve.

Raising the friction from zero to a particular value and then applying a uniform pulse train for one revolution generates Start/stop curves. If the motor follows it successfully, a higher speed uniform pulse train is tried until one is reached which the motor does not follow correctly. The last successful speed is then plotted as a point on the curve. Load inertia is normally equated to rotor inertia for matched start/stop curves.

Accelerating the motor from zero speed to a specific speed shown on the curve generates slew curves. This is ideally done with no frictional load and with as much acceleration time as required. When at speed, friction is increased from zero to the value at which the motor stalls. These points are plotted to form the slew curve. It represents the outer limit of performance capability.

For practical purposes, true system performance limits fall between these two curves. Instantaneous torque output approaching that of holding torque (inherent in permanent-magnet & hybrid stepping motors) serves to boost the start/stop curve far closer to the slew curve than would initially be expected. A Typical theoretical Torque/speed graph is shown in figure 10.

Figure 10. Theoretical Torque vs. Velocity performance graph

The improved performance arising from the use of the DSP filter stage will be assessed by plotting the torque vs. velocity graph for normal operation against the corresponding graph for filtered operation. The algorithm will be evaluated in terms of performance improvements, this being reported in a future paper.

Conclusion

From the results presented it can be seen that the previous assumptions of rough motion in interpolating systems were valid. Further because the damping of mechanics may be light, excitation of dynamics may occur giving rise to significant position errors. The results show that the performance of an experimental test machine can be improved through the addition of a filter stage to the interpolator. The improvements offered by the filter technique are better cut quality and higher performance, in effect high quality, faster production for a given cost. Further work is planned to confirm the validity of the filter technique on the experimental test rig and also on a production machine.

Acknowledgements

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IMPROVED ENERGY EFFICIENCY OF STEPPER MOTOR SYSTEMS

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KEYWORDS

Energy Efficiency, Stepper Motors, Continuous Path Motion, Multiple Axis Control.

ABSTRACT

Stepping motors have become increasingly attractive in motion control applications, mainly due to the simplicity of their interface requirements and low costs. Unfortunately, the small but discrete steps of movement inherent in the motor design, combined with the inevitable quantisation of time leads to roughness of motion which may result in real limitations of system performance. Previous work has demonstrated the value of carefully chosen acceleration profiles for single axis motion. However work conducted within the real-time machine control group, Nottingham Trent University has shown such profiles to be of limited value in the case of multiple axis continuous path systems, where interaction of acceleration and interpolation results in further roughness of the path.

The 'art of stepper motor system design' suggests that motors typically having a torque capability of double the load requirement should be sufficient to ensure reliable operation. However when full account of interpolation and ramping are considered, yet larger motors may be required. Steiger has shown theoretically that the resulting rough motion of the machine, causes the limitation in performance through an increase in load being reflected back to the motor.

The work reported in this paper presents a technique/algorithm that can be used to reduce the required power of continuous path multi-axis stepper motor driven machinery, whilst still maintaining the same level of performance.

The algorithm is implemented in software on a digital signal processor and evaluated to the control signals for a four-axis CNC machine. The results were taken from both an experimental rig and a CNC machine.

The performance of the algorithm was assessed by measuring the dynamic displacement of the rotor position compared to it's nominal position. Acarnley and others discuss how to calculate the nominal rotor position for a given load. From these results it can be seen that the performance of stepper motor driven machinery can be improved.

Using this technique with more precise control, smaller motors can be employed giving sufficient energy efficiency savings.

INTRODUCTION

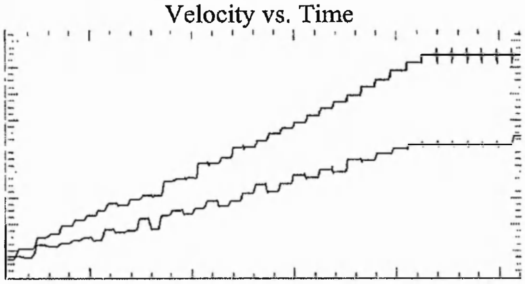
Previous work in the field of stepper motor control by authors such as Palmin [1] and Acarnley[2] has shown the importance of acceleration profile shaping if satisfactory performance is to be achieved from stepping motor systems. Motors themselves and transmission systems have natural resonances and the significance of these can be reduced if the profile is suitably shaped. Unfortunately, this is of little benefit in multiple axes interpolating systems, such as machine tools, where simultaneous synchronised motion in several axes may be required. Steiger, et al.[3-4] have shown from a theoretical point of view how the interaction of any acceleration profile and interpolation may lead to rough motion. In particular it was found that the least dominant axis suffered from coarse velocity changes, these being developed by the action of synchronising the two axes. See Figure 1. In this paper, we show how these conclusions are

substantiated in practice, how the motion may excite the drive system dynamics, resulting in unacceptably large velocity fluctuations and position errors.

Stepper motor systems are generally controlled by open loop electronics, where clock & direction signals are applied to appropriate counter and power electronic circuits in order for a sequence of energisation to be applied to the motor windings. It is unusual for position encoders to be used with stepper motors; they increase cost and do little to address the problems discussed in this paper.

In order to analyse the motion it is necessary to instrument a typical transmission system. An experimental system has been constructed and this is fitted with both rotary and linear encoders. A special purpose data logger has been developed within the group [5], allowing precision logging of the drive pulse trains and those gathered from the encoders. In order to allow maximum flexibility for subsequent analysis, the precise timing of each pulse edge transition is recorded. This minimises the significance of measurement quantisation, since the clock speed can be made significantly high in relation to the pulse rate.

Velocity vs. Time



Y axis 0 – 2 m/min. X axis 0 – 0.26 Seconds

Figure 1. Input velocity of X & Y-axis

STEPPER MOTOR CONTROL ANALYSIS

One of the established ways of improving the accuracy of stepper motor machinery is to ensure that any machine vibrations are damped out by the mechanics of the machine itself. The system will have inherent viscous & frictional damping. Normally the CNC system will have sufficient mass to damp out any machine vibrations that exist. In some cases torque dampers or frictional driven flywheels may be fitted to the drive to provide extra damping.

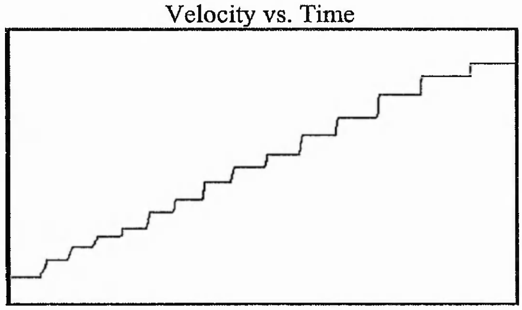
The disadvantage with this type of vibration control is that it generally leads to greater machine mass to damp out resonant frequencies. This in turn increases the viscous friction due to the added pressure on the guide rails. To compensate either higher torque stepper motors are needed or the overall performance of the machine needs to be down graded. An alternative method currently being researched and implemented in industry is to electronically damp out these vibrations.

Stepper motors are driven by a digital pulse train. The rotational velocity of the spindle is dependent upon the frequency of this pulse train, the higher the frequency the greater the speed.

Many controllers give out stepped changes of pulse frequency, these steps forming a staircase effect when viewed as a velocity profile, as seen in figure 2. The steps shown are due to the large interpolation step size used by this example. The graph in figure 2 shows the acceleration phase of a velocity profile, accelerating to a velocity of 0.8 m/min.

In effect the controller quantises the demand velocity profile. Therefore in control terms the input to the system can then be viewed as a series of cascading step inputs, with resulting step response. Typically this step response will be under-damped in nature, when the system is lightly loaded.

Velocity vs. Time



Y axis 0 – 1 m/min. X axis 0 – 0.2 Seconds

Figure 2. Velocity Profile of Controller Output Pulses.

In the path-planning world the path can be interpolated to within one unit corresponding to a single stepper motor pulse, or a unit representing many pulses. In the physical world many factors contribute to the actual path. Errors can be introduced into the system from several sources, including the following;

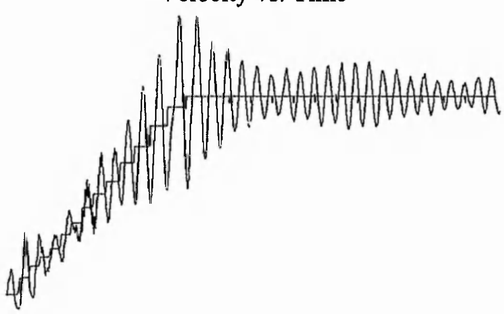
- Rounding errors in the path planning algorithm, or
Interpolation algorithm.

- Inadequacies with the Interpolator algorithm,
causing misalignment, or speed changes as the tool
machines around arcs, or circles.

- Machine vibrations due to resonance effects,
chatter, bearing clearances.

- Inadequacies with motion control system or drive
system, due to component tolerances or
control/drive algorithm.

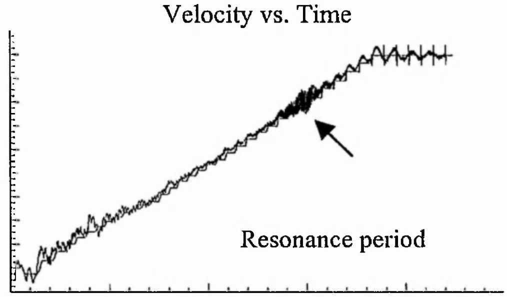
If the velocity profile of figure 2 is implemented on the lightly damped test rig the velocity of the machine can be seen to fluctuate as seen in figure 3. The graph in figure 3 shows the demand and output response of a velocity profile, accelerating to a velocity of 2 M/min. The measured velocity is shown to fluctuate , these fluctuations forming the dynamic response to the roughness of motion. These fluctuations in velocity correspond to a path error in the order of 30 microns.



Y axis 0 – 2.5 M/min. X axis 0 – 0.45 Seconds

Figure 3. Velocity Profile of Input and Output Motion

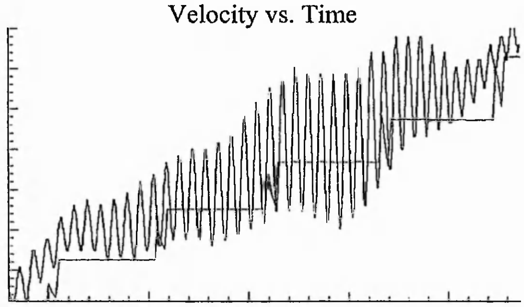
This stepped input is modelled to be a representation of the disturbances that the stepper motor may experience whilst following a contour path, as shown in figure 1. A representation of a normal velocity profile is shown in figure 4. The graph in figure 4 shows a velocity profile, accelerating to a velocity of 2 m/min.



Y axis 0 – 2.2 m/min. X axis 0 – 0.25 Seconds

Figure 4. Measured Normal Velocity Profile

Whilst the amplitude of the velocity fluctuations is small, causing a minimal amount of displacement error, closer inspection shows there to be a resonance period. Upon analysis this resonance period corresponds to the natural frequency of the pulley belts used. An expanded view is shown in figure 5. The range of the x-axis of the graph shown in figure 5 is 25 milliseconds with each grating representing 1 millisecond. The grating on the Y-axis represents 0.01 m/min within a velocity range of 1.4 – 1.7 m/min.



Y axis 1.4 – 1.7 m/min.
X axis 0.135 – 0.16 Seconds

Figure 5 Exploded view of resonance period from figure 4.

The resonance period may only cause a minimal displacement error, but the sharp changes in acceleration and deacceleration limit the performance of the machine in other ways. When the maximum acceleration is of these flucatuations measured it corresponds to the maximum theoretical value, whilst from the control aspect the operating speed/accleration is often one tenth of this value.

Acarnley describes how the maximum torque can be obtained from a stepper motor if the exact rotor position is fed back to the drive controls and used to control the conduction angle or switching angle, as referred to by Acarnley, for each phase of the stepper motor. Ideally the position pulses should be generated at the crossover points of the phase torque characteristic, see figure 6

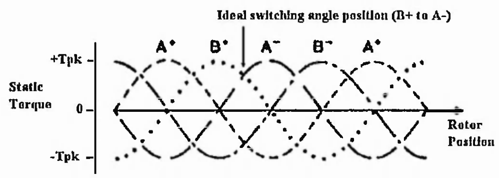


Figure 6. Ideal Switching Angle. (Courtesy of Acarnley).

Many authors have studied the method presented by Acarnley and as a consequence, suitable drive solutions have been developed. However the problem arises that the solutions tackle the problem at the low level and do not take account of any path-planning or interpolation disturbances to the system.

Palmin presents the case that if a parabolic velocity profile is implemented for a single axis transmission system then maximum torque utilisation can be achieved.

The generation of the parabolic velocity profile for multi-axis machinery would mean that the control system would have to interpolate on a single step basis, which in turn may also lead to dynamic displacement errors caused by the secondary axis stopping and starting constantly due to the coarse interpolation angle (i.e 0 , 45 or 90 degrees). The coarse interpolation angle being required to fulfil the requirement of synchronising the distance, & velocity to time, essential for continuous path motion.

SIMULATION

To develop a solution to the presented performance limitation problem the CNC machine was mathematically modelled. Once a model has been established it is possible to produce a simulator of the modelled system. This is conventionally achieved by taking the Laplace transform and converting it to a Z transform by the means of a Bi-linear transformation. From the Z transform it is a small step to convert the transform into a difference equation that can directly produce a numerical simulation. This standard method is embodied within the simulink toolbox for Matlab. The author used this toolbox to build a simulator.

The simulation of the system was used to confirm Palmin's earlier results involving analogue-based ramp generation of a parabolic velocity profile. The author confirmed the findings of Palmin et al, concerning the maximum utilisation of torque by the use of a parabolic velocity profile.

The results obtained from the simulation were used to prototype a digital based velocity ramp generator involving a digital signal processor. This generator was used to confirm the findings developed from the simulator. When results where collected from the test rig they showed that care must be taken with the generation of a parabolic velocity profile. The parabolic velocity profile advances very slowly through the range of frequencies from the starting velocity through to the maximum velocity. This means that if there are any natural frequencies within this range then they will resonate with a duration dependant upon the acceleration slope of the velocity profile. Therefore the acceleration slope should be made as short as the machine would allow. This again is likely to cause problems with multi-axis machinery.

Many authors recommend using a two-stage interpolator, the reasoning behind this is to obtain better interpolation for the given time constraint. This is a good option if we are considering the path-planning environment but it fails to correct the velocity fluctuations in the physical world. The answer put forward by the author is to employ a novel filter algorithm such that path accuracy and cut quality can both be improved. Such a filter should also facilitate higher performance from a given motion system. This additional stage would smooth the output velocity in a parabolic manner. A conceptual diagram of the proposed system is shown in figure 7. The system would measure the input pulse stream and then after passing through a software algorithm, the modified pulse stream would be regenerated. The regenerated pulse length will change to reflect the smoothed velocity profile.

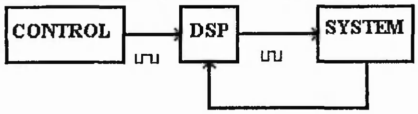
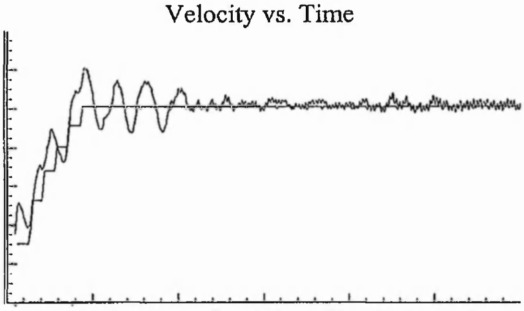


Figure 7. Proposed System

Through the use of computer simulation the filtering technique has been found to be effective in eliminating the velocity fluctuations. The use of the simulation model provides evidence towards the potential performance benefits of such a system. The next step was to implement the algorithm on the DSP and test the algorithm on the test rig. The reader can see from figure 8 the resultant velocity under normal conditions, and from figure 9 the resultant velocity with the filter applied.

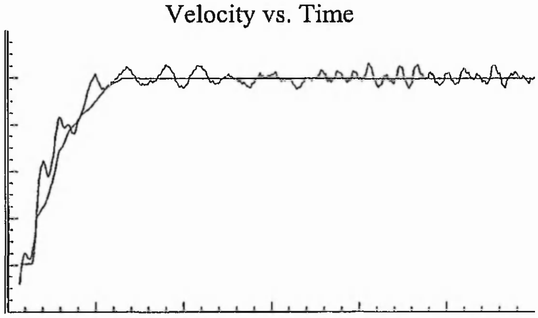
Velocity vs. Time



Y axis 0.1 – 0.7 m/min. X axis 0 – 0.3 Seconds

Figure 8. Measured Normal Velocity Profile

Velocity vs. Time



Y axis 0.1 – 0.7 m/min. X axis 0 – 0.3 Seconds

Figure 9. Filtered Velocity Profile

The reader can see that the filtered velocity profile has been smoothed, creating less velocity fluctuations. Since this process is an additional stage to the controller it presents no constraints upon the performance of such controller, in terms of interpolation etc.

The performance of the algorithm was quantitatively assessed by measuring the dynamic displacement of the rotor position compared to it's nominal position. Acarnley and others discuss how to calculate the nominal rotor position for a given load. If the rotor position deviates by two full steps then the stepper will gain or lose a step and stalling will occur in most situations. By measuring the change in rotor position we can estimate the torque capabilities, hence performance of the stepper motor when driven in any control/drive configuration. The technique is derived from Newton's force equation.

Force(N) = Mass x Acceleration + Friction

Figure 10 shows the resultant dynamic displacement error vs distance travelled. The reader can see that initially the filtered profile (b) causes more displacement error. The reason for this is due to parabolic nature of waveform, during the linear part of acceleration phase the stepper is accelerating at a much greater rate then the unfiltered profile. Once this stage is finished the acceleration decreases and consequently the motor has more torque reserve to decrease the displacement error back to a nominal level. This level represents the frictional components interacting with the velocity, not shown is that both waveforms settle down to this nominal value.

Dynamic Displacement Error vs. Distance Traversed.

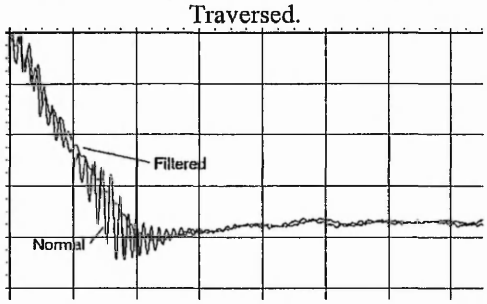


Figure 10. Displacement Error :-
(A) Unfiltered velocity profile
(B) Unfiltered velocity profile

CONCLUSION

From the results presented it can be seen that the previous assumptions of rough motion in interpolating systems were valid. Further because the damping of mechanics may be light, excitation of dynamics may occur giving rise to significant position errors. The results show that the performance of an CNC machine can be improved through the addition of a filter stage to the interpolator. Using this technique with more precise control, it can be seen that the motor has a larger reserve of torque to utilise. The benefit is such that smaller motors can be employed giving sufficient energy efficiency savings.

ACKNOWLEDGEMENTS

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