8-1-1980

Design and Testing of an Externally Powered Elbow Prosthesis

David E. Weston

DESIGN AND TESTING OF AN EXTERNALLY POWERED ELBOW PROSTHESIS

by

David E. Weston

Bachelor of Science, University of North Dakota, 1977

A Thesis

Submitted to the Graduate Faculty

of the

University of North Dakota

in partial fulfillment of the requirements

for the degree of

Master of Science

Grand Forks, North Dakota

May

1980

This Thesis submitted by David E. Weston in partial fulfillment of the requirements for the Degree of Master of Science from the University of North Dakota is hereby approved by the Faculty Advisory Committee under whom the work has been done.

J. Peter Sadler
(Chairman)

D. P. Naismith

Carl A. Zogg

This Thesis meets the standards for appearance and conforms to the style and format requirements of the Graduate School of the University of North Dakota, and is hereby approved.

A. William Johnson
Dean of the Graduate School

Permission

Title Design and Testing of an Externally Powered Elbow Prosthesis

Department Mechanical Engineering

Degree Master of Science

In presenting this thesis in partial fulfillment of the requirements for a graduate degree from the University of North Dakota, I agree that the Library of this University shall make it freely available for inspection. I further agree that permission for extensive copying for scholarly purposes may be granted by the professor who supervised my thesis work or, in his absence, by the Chairman of the Department or the Dean of the Graduate School. It is understood that any copying or publication or other use of this thesis or part thereof for financial gain shall not be allowed without my written permission. It is also understood that due recognition shall be given to me and to the University of North Dakota in any scholarly use which may be made of any material in my thesis.

Signature David E. Weston

Date APRIL 24 1980

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

ACKNOWLEDGEMENTS

The author wishes to express sincere appreciation to Dr. J. Peter Sadler for his supervision in this study. Thanks is also extended to Dr. D. P. Naismith and Dr. C. A. Zogg for serving as committee members.

A special commendation is extended to Mr. Larry Eickman (Chairman of Orthotics and Prosthetics Department, Grand Forks Medical Center Rehabilitation Hospital) for his contribution and insight added to the project. Additional thanks go to Mr. L. M. Nordling for lending his technical skills to the construction of the model, Mr. Don Mathsen for his help with the project and Linda Refling for her work in typing.

ABSTRACT

There continues to be a need for an externally powered prothesis which can be used by above elbow amputee patients who cannot effectively operate a conventional body powered prosthesis. This device must be reliable, economically constructed, and easily maintained in the field. A device employing a drive mechanism powered by a small DC motor has been designed to meet this need.

The device is based on an inversion of the belt driven pulley system and is a continuation of previous work employing this mechanism. A prototype was designed using this system in a size suitable for patient application. The model was constructed from commercially available parts and some shop fabrication. Once constructed, a laboratory testing program was devised to subject the prototype to typical tasks it would be required to perform in the field. The test results are included in the report. Also included are the kinematic and force analyses of the model and a computer program, based on the design equations written to simulate the motion of the device under load. A comparison between the simulated and experimental results is also presented.

The major intent of this project was to design and test a reliable externally powered above elbow prosthesis from commercially available parts. The design has proven to be a viable concept and should be pursued further based on the recommendations given in this thesis.

CHAPTER 1

INTRODUCTION

A prosthesis is a device used to replace the surgically removed or severed limb of an amputee patient. This device allows the patient to retain some of the mobility lost upon removal of the appendage. Since this paper is only concerned with arm (specifically forearm) prostheses, all subsequent discussion will be in regard to that type of device. Typically, a conventional body-powered prosthesis is raised by shoulder elevation, while the opening and closing of the terminal device (usually a hook) is accomplished by protraction and retraction of the shoulder. The forearm can be locked into position on fifteen degree increments with a nudge switch actuated by the patient's chin. In order to effectively operate this type of prosthesis, a patient should have a total of at least six inches of shoulder excursion. Because of a degree of paralysis or some other physiologic problem, some patients do not have this amount of shoulder mobility. Therefore, they must find other means of extending their existing capabilities or be faced with an existance of little movement. This problem has created an interest in externally powered prosthetics. One solution to this problem employing an externally powered device is the topic of this research paper.

The first attempts to utilize external power sources to actuate limb prostheses were made in Germany at the time of World War I [1]. The power sources were electrical and compressed air and were highly inefficient resulting in only slight benefits to the patients. Later

Alderson [1] designed a number of electrically operated devices as the next attempt at using an external power source. The models were experimental in nature and required a large amount of concentration on the part of the patient in order to effectively use these devices routinely. Still later, groups in Europe and Russia pursued the problem utilizing such external sources as compressed carbon dioxide and electric power with direct muscle bulge or myoelectric control. More recently a hydraulic system with an electrical energy source has been tried as a method of actuation in England [2]. This type of power source has given fair experimental results; however, the weight of the package is high, and, because of the high precision which must go into the component manufacture, the cost is also high. Electromechanical systems have the advantage of using a compact, safe, energy source capable of being conveniently and economically recharged. Therefore, most of the externally powered upper limb prostheses available today in this country employ a mechanical actuation system powered by a rechargable battery pack. One such mechanism is the Veterans Administration Electric Elbow.

The VA Electric Elbow is an experimental device developed through the Veterans Administration [3]. This elbow employs a permanent magnet electric motor, operated from a 25 volt battery pack, which is directly coupled to a planetary roller harmonic drive wave generator housed at the elbow joint. The wave generator forces the flexible spline of the harmonic drive to engage the rigid spline of the elbow housing. The planetary wave generator, in combination with a harmonic drive, achieve an overall speed ratio of 80:1. If the amputation is such that the battery pack can be located in the upper arm, a soft foam endoskeletal forearm is used. If not, the battery pack must be placed in a hard forearm shell which places a greater load on the elbow motor and thus reduces the lifting capability of the mechanism.

The VA Elbow has a fairly high failure rate primarily due to breaking of the flex spline drive component. Also among the complaints were excessive noise, weight, speed of operation, inadvertent operation of the switch, lack of free arm swing, and inadequate lifting force.

Of the powered arms on the market today, there are three which are most prominent. They are the Rancho Los Amigos Hospital elbow, the Army Medical Biomechanical Research Lab (AMBRL) elbow and the Boston elbow [4].

Each of these elbows is powered by a battery operated electrically driven actuator. The Rancho Los Amigos Hospital elbow is controlled by a pull switch in a shoulder harness. The AMBRL elbow is also controlled by a shoulder harness pull switch and has the added advantage of full rotation of the terminal device in the position of full arm extension. The Boston elbow is myoelectrically controlled by the use of surface electrodes on the stump. It employs a feedback system that maintains a constant speed of motion regardless of load within the limits of the design.

The main drive mechanism in all of these elbows is a rotational friction clutch. Because of the nature of this drive, these elbows experience a loss of clutch holding ability shortly after the mechanism is well broken in. Such failure causes excessive maintenance and large amounts of patient disuse time.

Field testing and use of these devices by patients has resulted in a list of desirable and undesirable features associated with these designs [4] [5]. Among the desirable features were a positive elbow lock in any position, the ease with which flexion is achieved, free swing of the elbow and ease of recharging. The undesirable features included such things as noise, weight, slow speed, bulkiness and inadvertent operation of the switch.

For a new design to meet patient acceptability there are several features and design criteria which must be addressed. The power source for the unit should have an adequate current capacity for at least one day's use before recharging, have a long life expectancy, be small and light weight, have a low cost, be readily available and be easily recharged. The actuator should be light and small, rugged, quiet, capable of producing a high torque, and very efficient, and it should be low in cost and readily available. In addition it should have a low static friction so that the arm motion appears smooth and natural. It should also be self-locking so that the arm can be stopped and held in any position against a reasonable load. Finally, the most essential feature is reversibility of the motor and the actuator so that both flexion and extension of the arm can be accomplished.

When the source and actuator have been selected so as to fulfill the above requirements, the overall mechanism should be designed to meet a basic set of operational criteria. These criteria were arrived at through consultation with the Medical Center Rehabilitation Hospital prosthetics staff prior to the start of this project in 1978, and are listed as follows. The speed of the mechanism must be such that the forearm traverses between full extension at 0^0 and full flexion at 120^0 in three seconds. The operational control of the elbow should be independent of the terminal device. The mechanism should be capable of producing three foot-pounds of torque and have a mechanical advantage such that it can resist a static torque loading of twenty-five foot-pounds. The weight of the elbow should be eighteen ounces or less and the mechanism should emit a low noise level during operation. The device should be economical to produce from readily available commercial parts, adjustable to a variety of patients, reliable and easily maintained in the field.

Initial work on a prosthesis design proposed to meet these criteria was performed by Krump in 1978 [6]. This design was a conceptual model and as such did not completely fulfill all the requirements listed above. The subject of this thesis is the development of a working prototype based on the initial design concept. It comes much closer to achieving these requirements but is still to be considered a preliminary design.

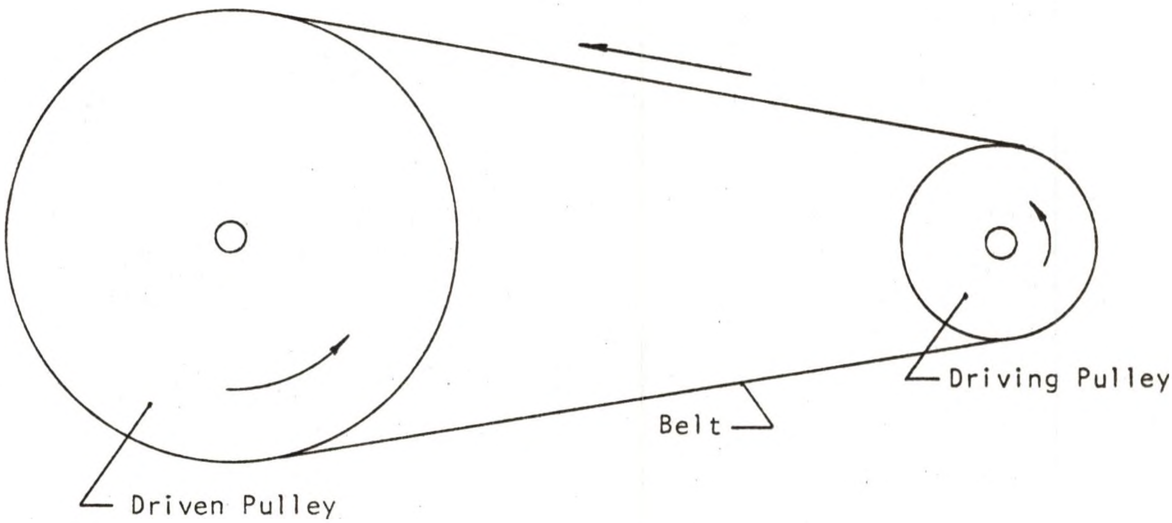
The remainder of this report deals with the design and analysis of the prototype as well as its subsequent testing. The procedure for laboratory testing of the device, as well as the data acquired from these tests, is reviewed in detail. The final chapter deals with the conclusions gained from this research and also recommendations for improvements of subsequent prototype models.

CHAPTER 2

DESIGN AND ANALYSIS OF THE PROTOTYPE

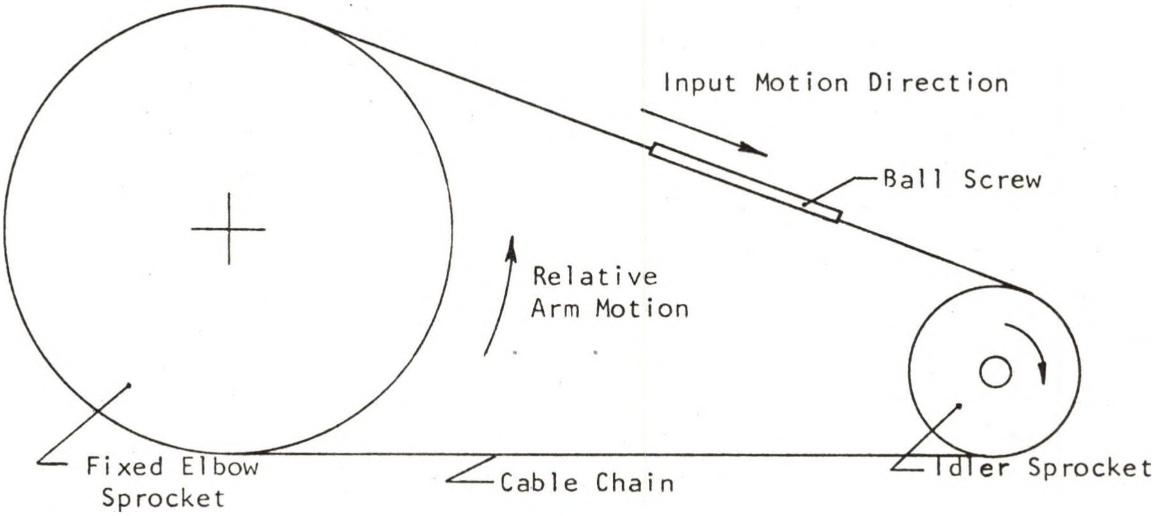
The externally powered elbow prosthesis described here is essentially a working miniaturization of the conceptual model devised by Krump in 1978 [6]. As such the prototype can be used in simulated field tests and is a major step in bridging the gap between the conceptual model and an actual marketable device.

The prototype model operates on an inversion of the two-pulley belt-drive system. In the conventional application of this system the first pulley drives a second pulley through the actuation of a connecting belt (see Figure 1). All three of these components are present in the prototype design; however, their operations have been altered slightly. The conventional belt-pulley drive relies on friction between the contact surfaces of the belt and the pulleys for the efficient transmission of motion. Therefore, given a fixed belt size, this transmission efficiency can be increased by raising the tension of the belt. At high belt tensions the transmission efficiency under load is good, however, the power requirement for the driving motor, as well as the motor size, increase. This presents no problem in design when size, weight and electrical energy inputs are minor considerations. However, since these are all major considerations in the design of a prosthetic mechanism, a more positive drive was substituted. The pulleys were replaced by sprockets and the belt was replaced by a special cable chain manufactured by W. M. Berg, Inc. The application of this drive system in the prototype locates one sprocket at the elbow and the other in a framework at



Belt Pulley System, Conventional Application

FIGURE 1.



Application of the Belt Pulley System used in the Prototype

FIGURE 2.

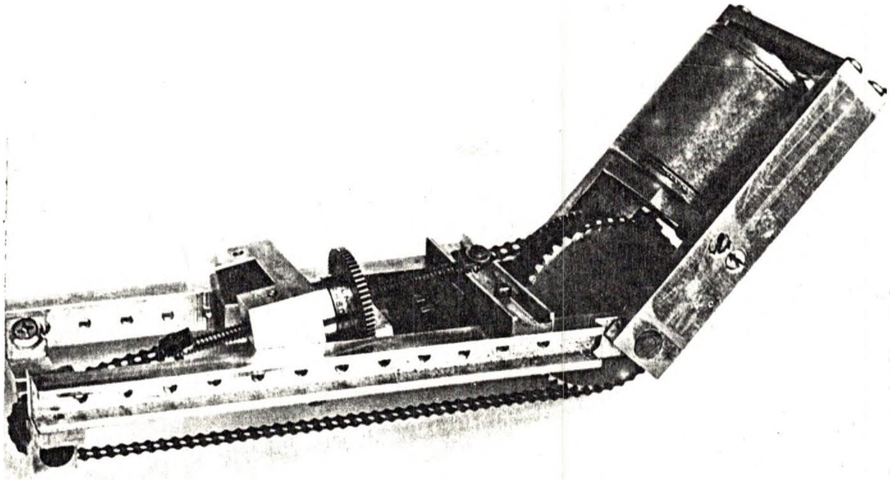
a fixed distance between the elbow and the wrist. The sprocket at the elbow is fixed to a shaft and to the upper arm while the other sprocket is allowed to rotate. Both sprockets are held within a structural frame and are connected to each other by the cable chain. The displacement of this chain is achieved through the use of a ball-bearing, screw-drive, linear actuator which replaces a section of chain. As the chain is displaced, the forearm is forced to rotate about the fixed elbow sprocket (see Figure 2).

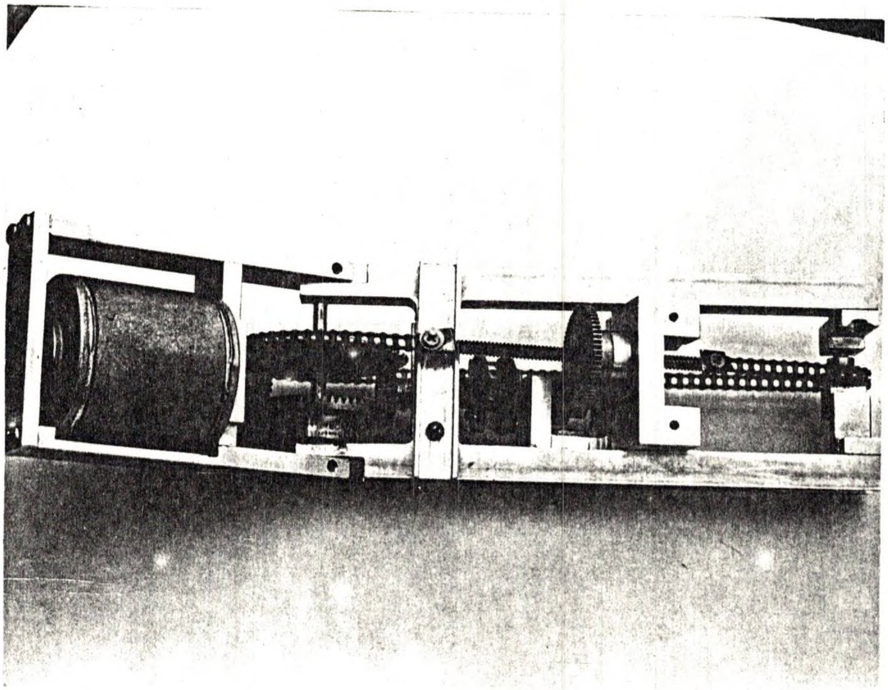
This system has the flexibility of allowing the power source (a motor) to be placed in either the forearm or the upper arm. Since it was desirable to reduce the dead weight of the forearm raised each cycle, the power source was located in the upper arm. The mechanism creates a uniform rotational velocity ratio between input and output during operation. It also offers a large gear reduction between the input and output motions. This reduction can be obtained from an overall summation of the reductions between the motor and the nut, the nut and the screw, and the screw in combination with the relative diameter of the elbow sprocket.

Prototype Construction

The prototype model was constructed based in part on the work by Krump, 1978 [6], and also on the design equations to be presented in later sections of this chapter. The finished model can be seen in the photographs in Figure 3 and also the drawing in Figure 4. The working drawings used for construction of the prototype are presented in Appendix 1.

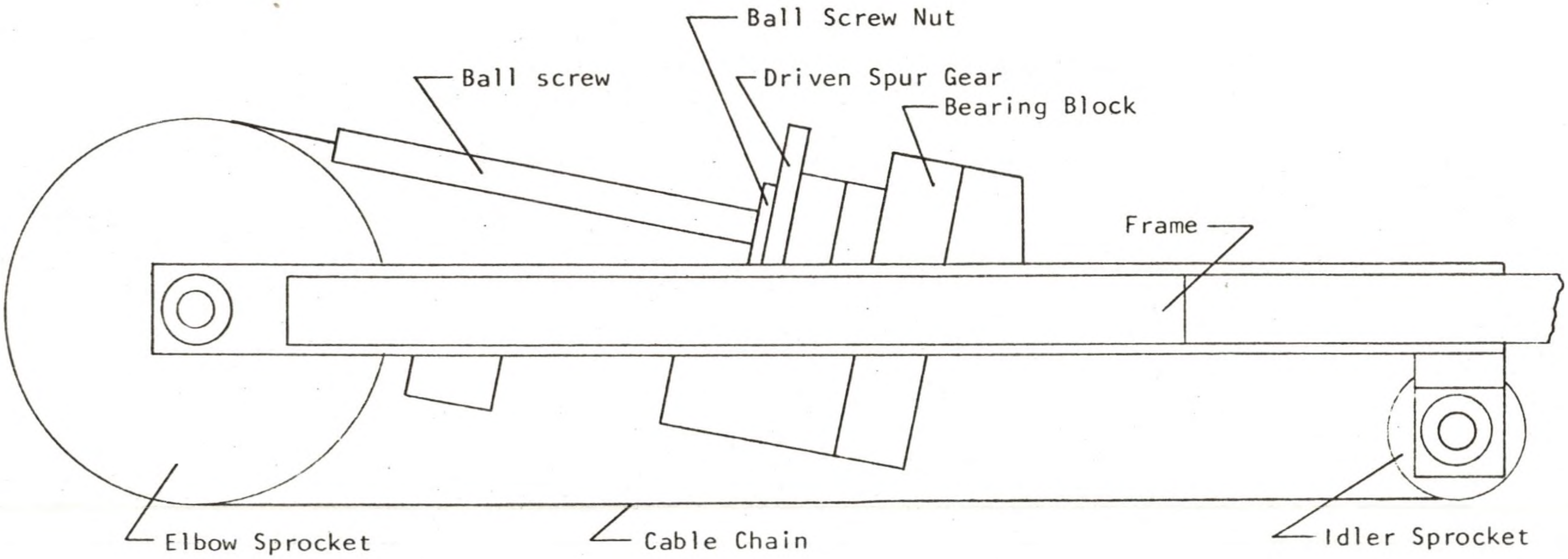
The reasons for component selection and sizing will become clearer in subsequent sections; however, some discussion will be presented here





PHOTOGRAPHS OF THE PROTOTYPE

FIGURE 3



A FULL SIZED PROFILE VIEW OF THE PROTOTYPE MODEL

FIGURE 4

in order to give a physical feel as to where the numerical values used in the design equations originated.

Two of the components, the motor and the ball-bearing screw, had been obtained prior to the start of this project; therefore, all subsequent component selection and design had to be made to accommodate these parts. Since it was desirable to place the motor in the upper arm, the problem became one of transferring the power from the motor to the ball-screw in the lower arm. At first it was thought that a flexible cable drive could be employed to transmit the power, but the minimum bending radius constraint for the cable caused that idea to be abandoned. The approach finally used was to mount a large bevel gear on the elbow pivot shaft and to drive it with a bevel pinion attached to the motor shaft. The bevel gear in turn drove a bevel pinion on the lower arm. This gear train of bevel gears allowed power to be transferred from the motor to the lower arm continuously regardless of the angular position of the lower arm. The lower bevel pinion drove a spur gear pinion through a flexible coupling. A flexible coupling was required since the size constraint and the geometry of the frame made it impossible to use a one-piece solid shaft. The spur pinion drove a spur gear which was concentric with the ball-screw nut and thus provided the motion of the arm.

The length of the frame was chosen after a discussion with the Rehabilitation Hospital prosthetist. It was decided that the prototype should be made adjustable to a variety of patients and that this length could be considered a minium size. The sides of the frame were machined to form tracks for an adjustable wrist extension section to be used to change the length of the arm and also to attach the terminal device.

The following sections present analyses based in part on the component sizes discussed above.

Kinematic Analysis

A kinematic analysis of the gear train from the motor to the ball-screw nut was performed in order to derive a relationship between the angular speeds of the motor and the arm. Using the manufacturer's published data the speed ratios between the gears were determined as follows:

n_1/n_2 = motor bevel pinion to bevel gear = 4:1

n_2/n_3 = bevel gear to lower arm bevel pinion = 1:4

n_3/n_4 = lower bevel pinion to spur gear pinion = 1:1

n_4/n_5 = spur pinion to spur gear = 2:1

n_5/n_6 = spur gear to ball-screw nut = 1:1.

Letting the motor speed equal the variable $\dot{\phi}_m$, the rotational nut speed, N_s, is equal to the motor speed times the combined effect of all of the gear ratios.

$$N_s = \dot{\phi}_m \left[\frac{n_2}{n_1} \; \frac{n_3}{n_2} \; \frac{n_4}{n_3} \; \frac{n_5}{n_4} \; \frac{n_6}{n_5}\right] . \qquad (2\text{-}1)$$

Substituting

$$N_s = \dot{\phi}_m \left[\frac{1}{4} \; \frac{4}{1} \; \frac{1}{1} \; \frac{1}{2} \; \frac{1}{1}\right] = \frac{1}{2}\dot{\phi}_m .$$

A displacement of the cable chain by the amount x causes a relative displacement of the elbow sprocket by an equal amount or:

$$x = \frac{D}{2}\theta \qquad (2\text{-}2)$$

where: D - is the radius of the elbow sprocket

θ - is the angular displacement of the sprocket about its axis relative to the arm.

Since the arm is constrained to rotate about the fixed elbow pulley, the angular displacement of the arm for a given belt movement, x, is

$$\theta = \frac{2}{D} x. \qquad (2\text{-}3)$$

The speed of the belt, $\dot{x}$, is equal to the speed of the nut, N_s, times the lead of the nut, ℓ:

$$\dot{x} = \frac{1}{2} \dot{\phi}_m \ell. \qquad (2\text{-}4)$$

Substituting and simplifying, the arm speed as a function of motor speed becomes:

$$\dot{\theta} = \frac{\ell}{D} \dot{\phi}_m$$

Numerically for the prototype:

$$D = 2.125 \text{ inches} \qquad \ell = .007958 \text{ inch/radian}$$

$$\dot{\phi}_m = 267.03\, \dot{\theta} \qquad (2\text{-}5)$$

Force Analysis

In order to determine the forces expected to act on the prototype during operation, a force analysis of the mechanism was performed. A free body diagram of the arm appears in Figure 5.

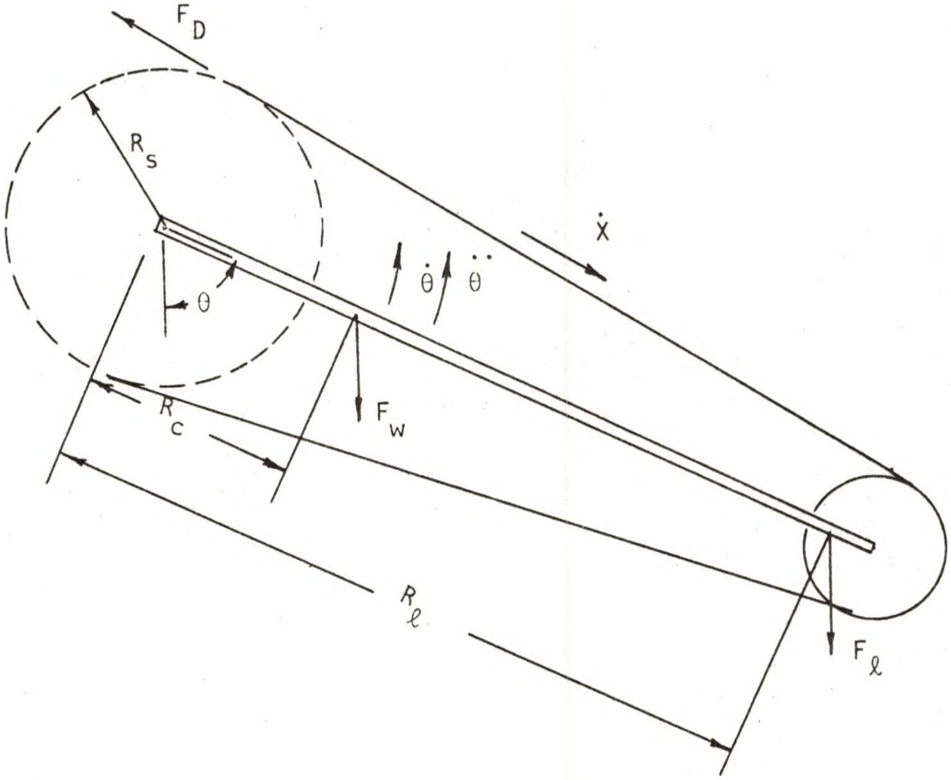
The mathematical relationship which describes the motion of the prototype is given below:

$$\Sigma T - (I_o + I_m)\alpha = 0 \qquad (2\text{-}6)$$

where:

ΣT - is the summation of the torques

$I_o = I_{cg} + mR_c^2$ - is the mass moment of inertia of the arm about the elbow pivot



A FREE BODY DIAGRAM OF THE ARM

FIGURE 5

I_{cg} - is the mass moment of inertia of the arm about its center of gravity

I_m - is the mass moment of inertia of the applied load weight about the elbow

$\alpha=\ddot{\theta}$ - is the angular acceleration

m - is the mass of the forearm

R_c - is the radius to the center of mass of the arm from the elbow pivot.

When the mechanism is just starting to move in either direction the angular velocity of the arm, ω, is zero; however, the angular acceleration, α, is some quantity other than zero. Substituting into the equation yields:

$$\Sigma T = (I_o + I_m)\alpha \tag{2-7}$$

During constant velocity motion the angular acceleration becomes zero and the angular velocity is some constant other than zero. The equation for this special case then becomes:

$$\Sigma T = 0 \tag{2-8}$$

As an initial approximation, it was assumed that the arm motion consisted of a constant acceleration segment, followed by a constant velocity segment, and then a constant deceleration segment, and that the total time for the starting acceleration, α, to take place was approximately 0.10 second. Since it was desired to have the arm traverse 120 degrees or about two radians in three seconds, the constant angular velocity level during the cycle should be 0.7 radians per second. The angular acceleration and velocity are related by the following equation:

$$\omega = \alpha t \tag{2-9}$$

solving for α:

$$\alpha = \frac{\omega}{t} = \frac{0.7}{0.10} = 7.0 \quad \text{rad/sec}^2.$$

In order to calculate the torque, T, the moments of inertia were determined as follows:

$$mR_c^2 = \frac{.955}{386.4}(2.625)^2 = .01703 \quad \text{lb-sec}^2\text{-in}$$

$$I_{cg} = \frac{1}{12} m (a^2 + b^2) = \frac{1}{12}\frac{.955}{386.4}[(7)^2 + (2)^2]$$

$$= .01092 \quad \text{lb-sec}^2\text{-in}$$

where: a,b - are approximate length and height dimensions, assuming the forearm to be a solid rectangular prism.

$$I_o = I_{cg} + m R_c^2 = .02795 \quad (\text{lb-sec}^2\text{-in}) \tag{2-10}$$

The weight of the arm is 0.955 pounds centered at 2.625 inches from the elbow.

$$I_m = m_\ell R_\ell^2 \quad (\text{lb-sec}^2\text{-in}) \tag{2-11}$$

where m_ℓ - is the mass of the applied load weight, and

R_ℓ - is the distance to the load from the elbow pivot.

Combining all inertia terms and summing all torques about the elbow pivot, the force, F_D, required to drive the chain can be determined with reference to Figure 5 and the equation below:

$$F_D(R_s) - F_w(R_c)\sin\theta - F_\ell(R_\ell)\sin\theta - (I_o + I_m)\alpha = 0 \tag{2-12}$$

where: F_w - is the weight of the forearm

F_ℓ - is the applied load weight

R_s - is the elbow sprocket radius

R_c - is the distance to the center of gravity of the arm

R_ℓ - is the distance to the load

where:

θ - is the angular displacement of the arm from a vertical plane through the elbow pivot.

Based on a design load of three pounds at R_ℓ = 14 inches, the worst case situation would be starting upward motion from the $\theta = 90^0$ position.

$$F_D(1.0625) - .955(2.625) - 3(14) - (0.028 + 1.522)7.0 = 0$$

$$F_D = 52.1 \text{ pounds}$$

This analysis presents a worst case driving force equirement, where the load, F_ℓ, is at its maximum expected value and is located in its fully extended position. Also, α, the angular acceleration, is at its maximum value, for the motion was required to start at $\theta = 90^0$. The value of F_D allows calculation of the stresses on all the drive train components in the arm and also the torque requirements for the power source.

Motion Modelling Program

As a second approach, a computer program was written in order to mathematically simulate the motion of the arm under various loading configurations. The equations used in the program were taken from the two previous sections of this chapter.

It was first necessary to derive a relationship between the motor speed and the motor torque. Using the manufacturer's ratings (a copy of which can be found in Appendix 2), it was determined that the motor speed varies linearly with torque. In general the equation for a straight line is:

$$Y = mx + b \qquad (2\text{-}13)$$

where m is the slope of the line and b is the y-axis intercept.

For the motor used in the prototype, the equation specifically becomes:

$$\dot{\phi}_m = mT_m + b. \qquad (2\text{-}14)$$

From the graph in Appendix 2, it was determined that m was 325.5 radians per second per inch-pound and b was 356 radians per second. Substituting into the equation and solving for the motor torque in terms of motor speed yields,

$$T_m = 1.094 - .00307\,\dot{\phi}_m \quad \text{(inch-pounds)}.$$

Since this torque equation is based on an input voltage of 12 volts, not 9 volts as applied by the power source used during laboratory testing, it was desirable to modify this equation to obtain a better correlation with the actual operation of the motor. Operating a direct current motor at other than its rated voltage causes changes in the output characteristics of the motor. It was not known exactly how the characteristics of the motor used in the prototype would change due to an input reduction of approximately three volts. Therefore, some assumptions about these changes were made. First, it was assumed that the relationship between the motor speed and torque would remain linear and that the slope of this curve would not change. Using a tachometer to experimentally measure the no-load speed of the motor at an input voltage of nine volts gave an average speed of 2500 revolutions per minute. Inserting this change into the torque equation results in:

$$T_m = 0.804 - .00307\,\dot{\phi}_m \quad \text{(inch-pounds)}.$$

Further, there are losses in the transmission of power from the motor to the arm which reduce the useful torque which is delivered to the arm. These losses, which are due primarily to friction, are caused by such things as gear misalignment, gear tooth mismatch and bearing friction. The friction losses tend to be independent of speed, and can be accounted for in the torque equation as follows:

$$T_{net} = T_m - T_{losses} = A - B\,\dot{\phi}_m - T_{losses}. \qquad (2\text{-}15)$$

For the prototype model this loss term can be thought of as the torque which must be input by the motor to just overcome the drive train friction and start or maintain motion. A value for this torque loss using the prototype was determined experimentally. The model was mounted horizontally to eliminate any gravitational loading on the motor and a small torque was applied to the motor shaft. This torque was gradually increased until motion occurred. The test was conducted a number of times with reasonable agreement and an average loss torque, T_{losses}, of 0.188 inch-pounds was determined. Substituting into the torque equation gives:

$$T_{net} = 0.804 - .00307\,\dot{\phi}_m - 0.188 \quad \text{(inch-pounds)}.$$

From the kinematic analysis section, the relationship between the motor speed and the chain speed is:

$$\dot{\phi}_m = \frac{2}{\ell}\dot{x} \quad \text{(radians per second)}. \qquad (2\text{-}16)$$

The work done by the driving force, F_D, is equal to the product of this force and its displacement or:

$$\text{Work} = F_D\,x = T_{net}\,\phi_m = T_{net}\frac{2}{\ell}x. \qquad (2\text{-}17)$$

Therefore,

$$F_D = \frac{2}{\ell}T_{net}. \qquad (2\text{-}18)$$

The torque produced by F_D about the elbow is:

$$T = F_D\frac{D}{2} = \frac{D}{\ell}T_{net}. \qquad (2\text{-}19)$$

Again from the kinematic analysis section:

$$\dot{\phi}_m = 267.03\,\dot{\phi}. \qquad (2\text{-}20)$$

Substituting, the torque produced by F_D is:

$$T = \frac{D}{\ell}\,(0.804 - 0.8198\,\dot{\theta} - .188).$$

For the prototype model, D is 2.125 inches and ℓ is .007958 inches per radian. Substitution results in:

$$T = 214.7 - 218.9\,\dot{\theta} - 50.2 = 164.5 - 218.9\,\dot{\theta} \quad \text{(inch-pounds)}$$

Substituting into the torque equation from the force analysis section and solving for the acceleration, α, interms of torque gives:

$$\alpha = \ddot{\theta} = \frac{(164.5 - 218.9\,\dot{\theta} - F_w R_w \sin\theta - F_\ell R_\ell \sin\theta)}{(I_o + I_m)} \qquad (2\text{-}21)$$

This equation can be solved by numerical techniques giving values for $\dot{\theta}$ the angular velocity of the arm and $\dot{\theta}$ the angular position of the arm. A copy of this program can be seen in Appendix 3. From this data one may observe the effects which various loads have on the motion of the arm as it traverses through a complete cycle. The time required to reach essentially constant velocity can also be determined and was used to determine α in the preceding section. Plots were made of the motion versus load data obtained from this program. These graphs are presented in Chapter 3 along with similar plots obtained during laboratory testing of the prototype for comparison purposes.

<u>Range of Motion Analysis and Component Sizing</u>

The original restrictions and requirements for the design of the prototype, mentioned earlier, were used as a basis for the selection of the moving components of the mechanism.

One of the main objectives of this design was to achieve full rotation of the arm through 120 degrees over a period of three seconds. The characteristics which have an effect on this speed are: the size of the elbow sprocket, the lead of the ball bearing screw, the speed of the drive motor under load, and the gear reduction between the motor and the ball-screw nut.

A two and one-eighth inch diameter sprocket was selected for the elbow, since it offered the maximum mechanical advantage for the space available. The other sprocket does not change the mechanical advantage of the system, since it is only an idler. It was desirable to reduce the overall height of the assembly, so this sprocket was chosen to have a three-quarter inch diameter. This allowed the entire mechanism to be tilted downward as can be seen in the photograph (Figure 6).

Once the sprocket sizes had been selected, the screw lead was considered. A greater total gear reduction is achieved as the lead of the screw is reduced. Therefore, it is desirable to select a lead that is as small as possible. A ball bearing screw with a 0.05 inch lead was selected.

The power source available at the start of this project was the D-C motor described earlier. In order to select the proper gear reduction between the motor and the ball-screw nut, the following analysis was performed.

1) Desired rotation $= 120^0$

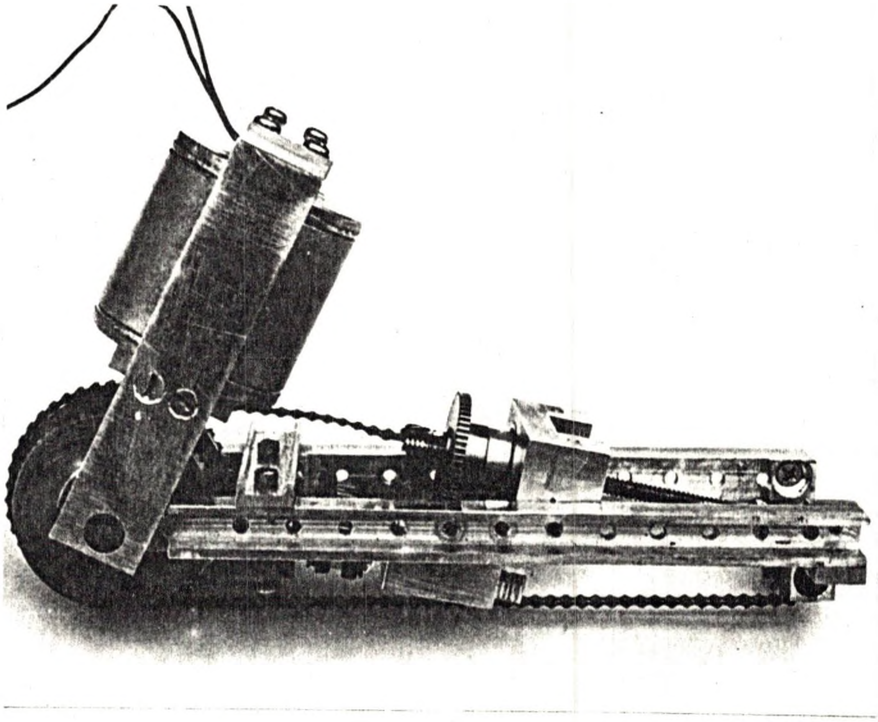
2) Sprocket diameter is $2\frac{1}{8}$ inches, and

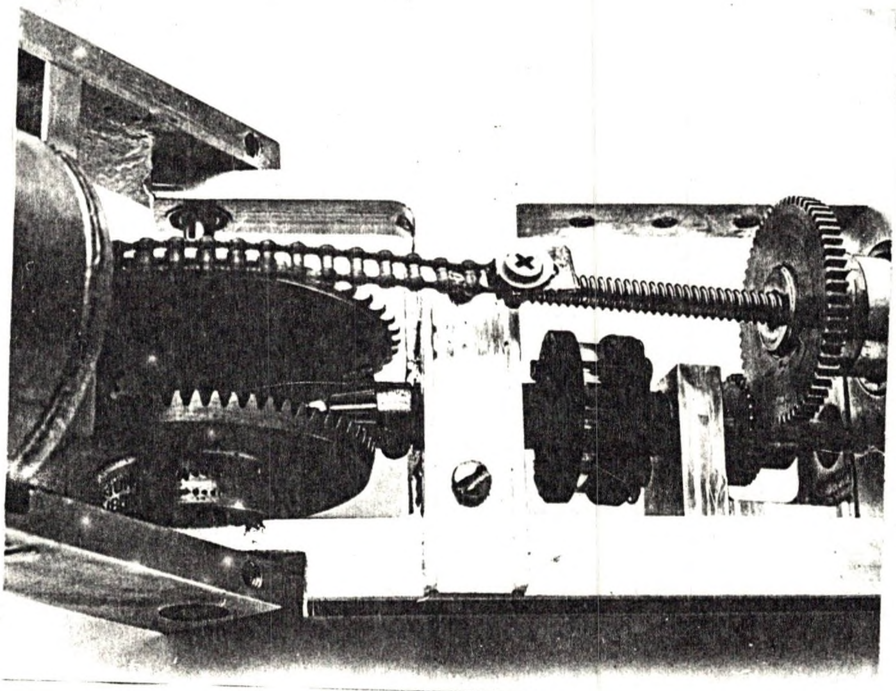
$$\pi(2.125)\ \frac{120}{360} = 2.23 \text{ inches}$$

Therefore, the screw must travel 2.23 inches in order to obtain 120^0 of rotation.

3) Desired time to raise arm $\simeq$ 3 seconds with a 0.05 inch screw lead

$$\frac{(2.23)\ (60)}{(.05)\ (3)} = 892 \text{ rpm} = \text{speed of screw nut}\ .$$





PHOTOGRAPHS OF THE PROTOTYPE

FIGURE 6

4) Motor operates at 2500 rpm

$$\frac{2500}{892} = 2.8 \simeq 3\text{-}1 \text{ gear reduction.}$$

Since it was expected that the motor would operate at a speed even slower than 2500 rpm due to the torque loss term previously discussed, a 2 to 1 gear ratio was used in order to still achieve full flexion or extension in three seconds.

Gear and Drive Train Analysis

Spur gears were used to transfer power from the motor to the ball-screw nut. It was originally thought that a flexible shaft could be used to transmit the power from the motor in the upper arm to the driving spur gear in the forearm. However, this concept presented some problems which could not be remedied. The biggest obstacle to the use of a flexible shaft was its minimum radius requirement, the smallest of which fell outside one and one-half inches. Therefore, it was decided that a set of bevel gears and pinions along with a flexible coupling would be used to transmit the power to the driver spur gear. A large bevel gear was mounted on the elbow shaft with the pinions mounted on the motor shaft and a shaft leading to the driver spur gear. Since there was some misalignment between the bevel pinion and the spur gear shafts, a flexible coupling was used to connect these shafts. Selection of the bevel gear sizes was based on clearance requirements between the two pinions at full arm flexion. For assumed gear sizes and materials, operating speeds and loads, the Lewis formula was used as the method of design [7]. Its application to the spur and the bevel gear drive train follows.

Spur Gear Analysis (Lewis Formula)

From previous assumptions based on the manufacturer's data, the motor torque is 0.193 inch-pounds at 1900 rpm, Gear reduction is 2-1,

Gear on nut - 1.25" diameter, and

Drive gear - .625" diameter.

The horsepower generated by the motor at its rated speed:

$$Hp = \frac{T_m}{63000} = \frac{.193(1900)}{63000} = 0.0058 \text{ hp} \qquad (2\text{-}22)$$

At this motor speed the pitch line velocity of the gears will be:

$$V = \frac{\pi dn}{12} = \frac{\pi(.625)(1900)}{12} = 311 \text{ fpm} \qquad (2\text{-}23)$$

The tangential gear force, W_t, is given by:

$$W_t = \frac{33000 \text{ hp}}{V} = \frac{33000(.0058)}{311} = 0.618 \text{ lb} \qquad (2\text{-}24)$$

The factor for dynamic loading (non-uniform load) is:

$$K_v = \frac{50}{50+\sqrt{v}} = \frac{50}{50+\sqrt{311}} = 0.739 . \qquad (2\text{-}25)$$

The gear pitch, P, is 48, and from $P = \frac{N}{d}$, the tooth numbers are:

$$N_1 = Pd = 48(.625) = 30, \text{ and} \qquad (2\text{-}26)$$

$$N_2 = 48(1.25) = 60.$$

Therefore, the pinion has 30 teeth and the gear has 60 teeth. The geometry factor, J, for these 20^0 full depth gears can be found from a graph which relates the number of gear teeth on the pinion and gear to the factor J [7].

$$J = .39 \text{ from the graph.}$$

The face width of the gears is one-eighth inch.

Given all these factors, the bending stress experienced by the gears is given by the equation:

$$\sigma = \frac{W_t P}{K_v F J} \qquad (2\text{-}27)$$

$$\sigma = \frac{(.618)(48)}{(.739)(.125)(.39)} = 823 \text{ psi}$$

Brass gears were used. Therefore, the ultimate strength is

$$S_{ut} = 72.5 \text{ Kpsi}$$

$$S_e = K_a K_b K_c K_d K_e K_f (.3)(S_{ut})$$

$$S_e = (.8)(1)(.814)(1)(1)(1)(.3)(72.5) = 14200 \text{ psi} \qquad (2\text{-}28)$$

where: K_a is the surface finish modification factor

K_b is the size modification factor

K_c is the reliability factor

K_d is the temperature compensation factor

K_e is the stress concentration factor

K_f is the miscellaneous effects factor.

The factor of safety, n_G, is given by:

$$n_G = \frac{S_e}{\sigma} = \frac{14200}{823} = 17.3 \qquad (2\text{-}29)$$

The total factor of safety can be computed by modifying n_G:

$$n = \frac{n_G}{K_o K_m} \qquad (2\text{-}30)$$

The overload factor, K_o, was found by assuming the power source to transmit a light shock load with the driven machinery offering a moderate shock resistance.

$$K_o = 1.5$$

The load distribution factor K_m was found by assuming that accuracy and mountings were such that less than full-face contact existed.

$$K_m = 2.5$$

Therefore, the total factor of safety is:

$$n = \frac{n_G}{K_o K_m}$$

$$n = \frac{17.3}{(1.5)(2.5)} = 4.6$$

Bevel Gear Analysis (Lewis Formula)

Again, from assumptions based on the manufacturer's data, the motor rating is 0.193 in-lb at 1900 rpm,

3 gear reduction is 1-4-1,

pinion on motor - .5 inch pitch diameter,

gear on elbow shaft - 2 inch pitch diameter,

pinion to spur gear - .5 inch pitch diameter. The horsepower generated by the motor is as before:

$$Hp = 0.0058 \text{ hp}$$

The gear velocity will be:

$$V = \frac{\pi dn}{12} = \frac{\pi(.5)(1900)}{12} = 248.7 \text{ fpm} \tag{2-31}$$

The tangential load, based on pitch radius is given by:

$$W_t = \frac{33000hp}{V} = \frac{33000(.0058)}{248.7} = 0.77 \text{ lb.} \tag{2-32}$$

The dynamic loading factor, K_v, is:

$$K_v = \frac{50}{50+\sqrt{v}} = \frac{50}{50+\sqrt{248.7}} = 0.760 \tag{2-33}$$

The gear pitch, P, is 32, and

$$N_1 = Pd = 32(.5) = 16 \text{ teeth, and} \tag{2-34}$$

$$N_2 = 32(2) = 64 \text{ teeth.}$$

Therefore, the pinions each have 16 teeth and the gear has 64 teeth. The geometry factor, J, can again be found from a published graph [7].

For the pinion $J = .25$.

For the gear $J = .19$.

The face width of these gears is one-quarter inch.

The bending stress is again given by:

$$\sigma = \frac{W_t P}{K_v FJ} . \tag{2-35}$$

For the pinions

$$\sigma = \frac{(.077)(32)}{(.76)(.25)(.25)} = 518 \text{ psi.}$$

For the gear

$$\sigma = \frac{(0.77)(32)}{(.76)(.25)(.19)} = 683 \text{ psi}$$

Brass bevel gears were also used: therefore, the ultimate strength is:

$$S_{ut} = 72.5 \text{ kpsi}$$

$$S_e = (.8)(1)(.8.4)(1)(1)(1)(.3)(72.5) = 14200 \text{ psi}$$

The factor of safety, n_G, is given by:

$$\text{Pinion} \quad n_G = \frac{S_e}{\sigma} = \frac{14200}{518} = 27.4 \qquad (2\text{-}36)$$

$$\text{Gear} \quad n_G = \frac{14200}{683} = 20.8$$

The overload factor, K_o, was assumed to be the same as for the spur gears,

$$K_o = 1.5.$$

The load distribution factor, K_v, was determined assuming both bevel gears to be mounted outboard of the bearings,

$$K_v = 1.3.$$

The total safety factors are:

$$\text{Pinion} \quad n = \frac{27.4}{(1.5)(1.3)} = 14.0,$$

$$\text{Gear} \quad n = \frac{20.8}{(1.5)(1.3)} = 10.7.$$

The factor of safety for all the gears is adequate.

Driving Torque Requirements

From the force analysis section, the value of the driving force, F_D, was:

$$F_D = 52.1 \text{ lbs.}$$

The manufacturer's specification lists a 90% efficiency for a ball-screw with a 0.05 inch lead. Given this information the torque required to drive the screw against the worst case load is given by

$$T_d = \frac{F_D(\ell)}{2\pi e} = \frac{(52.1)(.05)}{2\ \pi\ (.9)} = 0.46 \text{ in-lbs.} \qquad (2\text{-}37)$$

The total torque that the motor can develop on the screw is given by:

$$\frac{T_{screw}}{d_{screw}} = \frac{T_{motor}}{d_{motor}} \rightarrow T_{screw} = 2\ T_{motor} \tag{2-38}$$

$$T_{screw} = 2(.193) = 0.386 \text{ in-lbs.}$$

As can be observed, the torque supplied by the motor is too low to drive the T_d load. However, the relationship between motor speed and torque states that as the speed decreases, the torque increases. Therefore, this load can be raised at the expense of travel speed.

Selection of Components for the Prototype

The selection of components for the prototype model was based primarily on the stock availability of these components since this was a major objective of the design. Most of the components were purchased commercially. Those which could not be purchased were fabricated in the Central Shop at the University of North Dakota. A list of the components follows:

	Components	Source
1.	Gears	Boston Gear Co.
2.	Ball bearings (all rotating Shafts)	
3.	Flexible coupling	
4.	Sprockets and cable chain	W. M. Berg Inc.
5.	Ball Bearing Screw	Warner Electric Co.
6.	Motor	Barber-Colman
7.	Frame and Shafting	(shop fabricated)

A complete listing of all components can be found in Appendix 2.

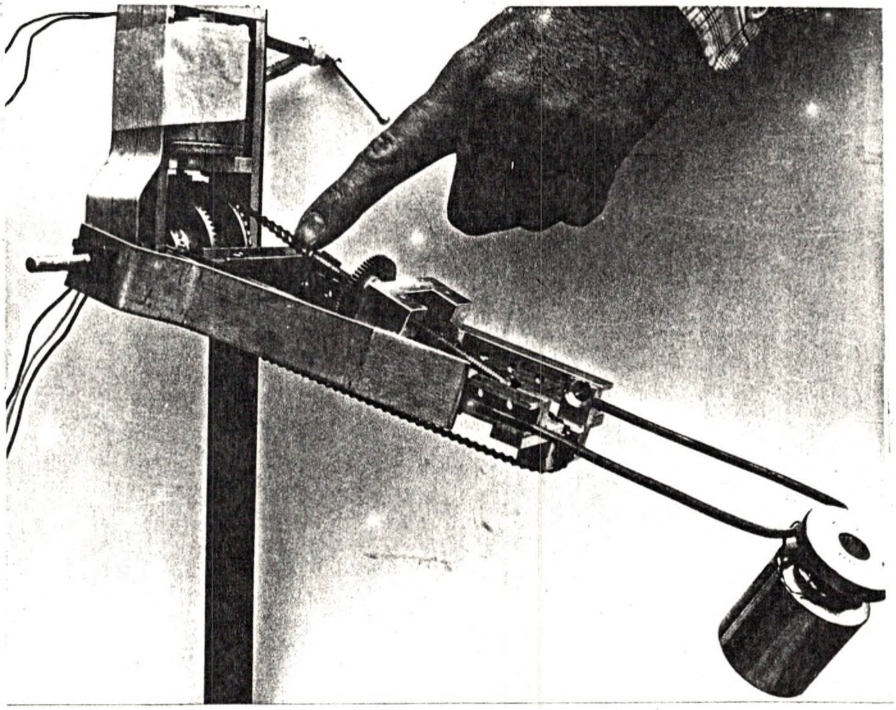
CHAPTER 3

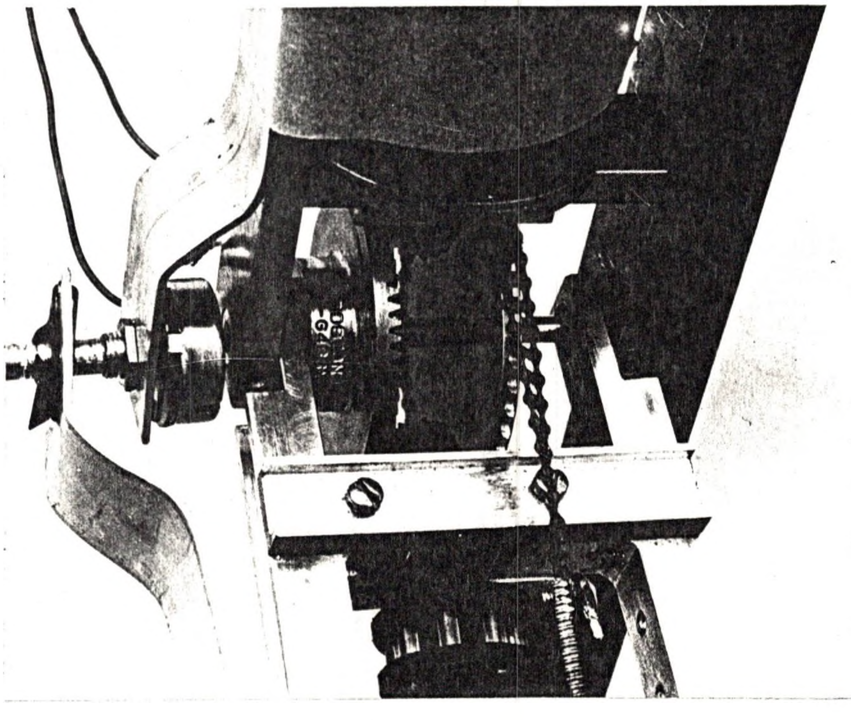
LABORATORY TESTING OF THE PROTOTYPE

A series of laboratory tests were undertaken using the prototype in order to evaluate its performance characteristics and to expose and, if possible, correct any weakness in the design of the device. A testing program was devised to subject the prototype model to various tasks it might be required to perform in the field under normal usage. Included in the program were pull tests which involved raising and lowering weights, and push tests, a field application of which might be as simple as holding a piece of paper in place. The tests were performed with the adjustable wrist extension section in three positions, thus allowing the observation of a range of expected operational characteristics.

Pull Tests

A typical set-up for the pull tests can be seen in the photograph, Figure 7. The driving motor was powered by a rechargable battery with a peak output voltage of approximately nine volts. A variable torque load was applied by attaching a weight to the wrist extension piece. This load was varied in one-half pound increments from zero to a maximum of about three and one-half pounds. For each test, the mechanism was driven from full extension to full flexion through an angle of approximately 116 degrees. Angular displacement was recorded as a function of time on a strip chart recorder through the use of an electrogoniometer, or elgon as it is commonly called to [8] [9]. The elgon





TYPICAL TEST SET-UP

FIGURE 7

consists of a linear potentiometer attached to the prototype at the elbow pivot point. The shaft of the potentiometer was driven by the forearm while the base remained stationary, fixed to the upper arm. During the test, a voltage was applied to the potentiometer and its variable output was monitored with a strip chart recorder. Once the equipment had been calibrated, knowing the chart speed, the time, for each cycle of the prototype under load could be computed. The chart also gave a visual record of the motion linearity through full rotation. Three full flexion cycles were performed at each load in order to obtain readings of the voltage and amperage, and to make a recording of the motion. The average voltage and amperage values were read from separate meters. A detailed listing of the instrumentation and test equipment appears in Appendix 4.

Push Tests

The push tests were conducted in essentially the same manner as the pull tests. In order to apply gravitational loading to the prototype, it was mounted on the test stand upside down with travel going from full flexion to full extension in each load case. Again, three separate trials were run at each load with data recorded as before.

Mid-Range Tests

In the two previous testing modes, motion was started from either the full extension or full flexion positions. Each cycle was completed and motion stopped when the arm had traversed to the opposite travel limitation. Since the arm will probably also be required in field use, to start from positions other than those of full flexion or extension, it was decided to conduct a series of pull tests from the 90 degree flexion position. Starting from this position places a maximum load on all the drive components, thus providing some additional insight into the operational characteristics of the prototype. Only pull tests were conducted from 90 degree flexion to full flexion with strip chart recordings of the motion recorded.

Data and Results

Once the data from the pull and push test modes had been collected, the efficiency of the mechanism under load was determined by calculating the ratio of the average output to input powers. The average input power was calculated, knowing the average input voltage and current, by the formula:

$$P_{in} = \frac{VI}{746} \quad \text{(horsepower).} \tag{3-1}$$

The average output power was computed, knowing the weight of the mechanism, the applied load and the time over which full travel occurred, by applying this formula:

$$P_{out} = \frac{W}{550 \cdot t \cdot 12} \quad \text{(horsepower),} \tag{3-2}$$

where W is the net work performed during time t. The derivation of this formula, in general, follows:

$$\begin{aligned} \text{Work} &= \text{force x translational displacement} \\ &= \text{moment x rotational displacement.} \end{aligned} \tag{3-3}$$

Defining θ as the angle from a vertical axis (below the elbow pivot) counter-clockwise to the axis of the arm, the moment about the elbow required to raise a series of loads at different radii from the elbow is:

$$M_e = \Sigma F \cdot r \cdot \sin\theta. \tag{3-4}$$

An incremental displacement of this moment is equal to the change in angle θ or:

$$\text{displacement} = d\theta. \tag{3-5}$$

The work is then the integral of this moment times displacement product over the entire range of θ.

$$W = \int_0^{\theta} \Sigma F \cdot r \cdot \sin\theta d\theta . \qquad (3\text{-}6)$$

For the specific case of the prototype, the series of load-radii products are of constant value and can be moved outside the integral sign. Integrating what remains and simplifying gives:

$$W = \Sigma F \cdot r \cdot (1-\cos\theta).$$

Further simplification can be realized by using the fact that there are two main loads against which the input power must work once the arm is in motion, namely the weight of the arm itself, F_w, and any additional load F_ℓ:

$$W = (F_w \cdot R_c + F_\ell \cdot R_\ell)(1-\cos\theta) \qquad (3\text{-}7)$$

where: R_c - is the radius from the elbow to the center of mass of the arm,

R_ℓ - is the radius from the elbow to the load, and

θ - is the angle traversed by the arm in time t.

The time, t, for each cycle can be determined from the strip chart. Thus, neglecting friction, the average output power equation is:

$$P_{out} = \frac{(F_w \cdot R_c + F_\ell \cdot R_\ell)(1-\cos\theta)}{550 \cdot t \cdot 12} . \qquad (3\text{-}8)$$

Finally, using the calculated values of P_{out} and P_{in}, the efficiency, N, can be computed by:

$$N = \frac{P_{out}}{P_{in}} \times 100\% . \qquad (3\text{-}9)$$

Tables I and II present the data acquired from the pull and push tests, respectively. Also included are the calculated average input and output powers and efficiencies for each load. Figure 8 is a graph of efficiency versus load for the pull test at each load radius. Figure 9

TABLE I

MEAN PULL TEST DATA AND EFFICIENCIES

Load (Pounds)	R_ℓ (inches)	Voltage	Amperage	Time (sec.)	P_{in}	P_{out}	Efficiency
0.0	8.0	8.9	.6	2.578	.0072	.0002	3.0%
0.4		9.0	.65	3.047	.0078	.0004	5.2%
1.0		8.95	.65	2.812	.0078	.0008	10.4%
1.4		8.9	.65	3.281	.0078	.0009	11.7%
2.0		8.9	.6	3.047	.0072	.0013	18.5%
2.4		9.0	.7	3.047	.0084	.0016	18.4%
3.0		8.9	.75	3.281	.0089	.0018	19.7%
3.4		8.9	.8	3.281	.0095	.0020	20.7%
0.4	11.0	9.15	.55	2.578	.0067	.0006	8.7%
1.0		9.05	.6	2.578	.0073	.0014	15.7%
1.4		8.95	.7	3.047	.0084	.0013	15.3%
2.0		8.85	.8	3.047	.0095	.0018	18.5%
2.4		8.8	.9	3.281	.0106	.0019	18.1%
3.0		8.6	1.05	3.516	.0121	.0022	18.2%
3.4		8.45	1.2	3.750	.0136	.0023	17.1%
0.4	13.5	9.1	.6	2.930	.0073	.0006	8.0%
1.0		8.95	.65	3.047	.0078	.0011	14.7%
1.4		8.9	.75	3.047	.0089	.0015	17.1%
2.0		8.75	.9	3.281	.0106	.0020	18.6%
2.4		8.7	1.0	3.516	.0117	.0022	18.6%
3.0		8.6	1.2	3.750	.0138	.0025	18.1%

TABLE II

MEAN PUSH TEST DATA AND EFFICIENCIES

Load (Pounds)	R_ℓ (inches)	Voltage	Amperage	Time (sec.)	P_{in}	P_{out}	Efficiency
0.0	8.0	9.1	0.6	2.695	.0073	.0002	2.7%
0.4		8.9	0.7	2.695	.0084	.0005	5.0%
1.0		8.8	0.8	2.812	.0094	.0008	8.5%
1.4		8.7	0.9	2.930	.0105	.0010	9.5%
2.0		8.55	1.0	3.164	.0115	.0013	11.3%
2.4		8.45	1.1	3.164	.0125	.0015	12.0%
3.0		8.4	1.2	3.633	.0135	.0016	11.9%
0.4	11.0	8.75	0.75	2.812	.0088	.0005	5.7%
1.0		8.6	0.9	2.930	.0104	.0010	9.6%
1.4		8.45	0.95	3.164	.0108	.0011	10.2%
2.0		8.25	1.2	3.516	.0133	.0015	11.3%
0.4	13.5	8.6	0.8	2.930	.0092	.0006	6.5%
1.0		8.45	0.95	3.047	.0108	.0011	10.2%
1.4		8.2	1.05	3.867	.0015	.0012	10.4%

is a graph of similar data from the push tests. Table III gives the calculated average speeds at each load and radius for each of the three testing modes. Figures 10, 11 and 12 are graphical presentations of the data in Table III. Figures 13, 14 and 15 are comparison plots of the displacement versus time data acquired experimentally and the same data generated by the motion simulation program derived in Chapter 2.

As can be seen from Figures 8 and 9 the overall efficiency of the mechanism is quite low with a maximum calculated value of 20.7%. There are several possible explanations for this low achieved efficiency.

One source of error may have been the method of testing. The voltage and amperage readings taken for each test were fairly constant throughout each test with average values recorded when some fluctuation did occur. The voltage readings may be taken as being reliable since the values read were mid-range on the meter scale. The amperage readings, however, could be in slight error. The ammeter range was 0 to 15 amperes, and readings taken during testing were all less than 2 amperes thus placing all measurements at the low end of the scale. Another source of error could be the method used to record the displacement versus time plot for each trial. The strip chart recorder used had a maximum paper speed of eight inches per minute. This speed was probably too slow in comparison to the angular velocity of the arm. In some instances it was difficult to make accurate time measurements from the readings, thus introducing some error in the P_{out} calculations.

Another source of efficiency loss is within the mechanism itself. In general, this source can be divided into four major areas of concern.

The first area was the motor itself. The manufacturer's data places the maximum motor efficiency at 74% (see Appendix 2). This

TABLE III

AVERAGE SPEED FOR EACH TEST MODE (RADIANS PER SECOND)

Load (Pounds)	R_ℓ (inches)	Pull Test	Push Test	Mid-Range Pull Test
0.0	8.0	0.785	0.751	0.975
0.4		0.664	0.751	0.968
1.0		0.720	0.720	0.774
1.4		0.617	0.691	0.645
2.0		0.664	0.640	0.553
2.4		0.664	0.640	0.484
3.0		0.617	0.557	0.430
3.4		0.617	---	0.430
0.4	11.0	0.785	0.720	0.553
1.0		0.785	0.691	0.553
1.4		0.664	0.640	0.484
2.0		0.664	0.576	0.484
2.4		0.617	---	0.430
3.0		0.576	---	---
3.4		0.540	---	---
0.4	13.5	0.691	0.691	0.645
1.0		0.664	0.664	0.645
1.4		0.664	0.524	0.553
2.0		0.617	---	0.553
2.4		0.576	---	0.484
3.0		0.540	---	0.387

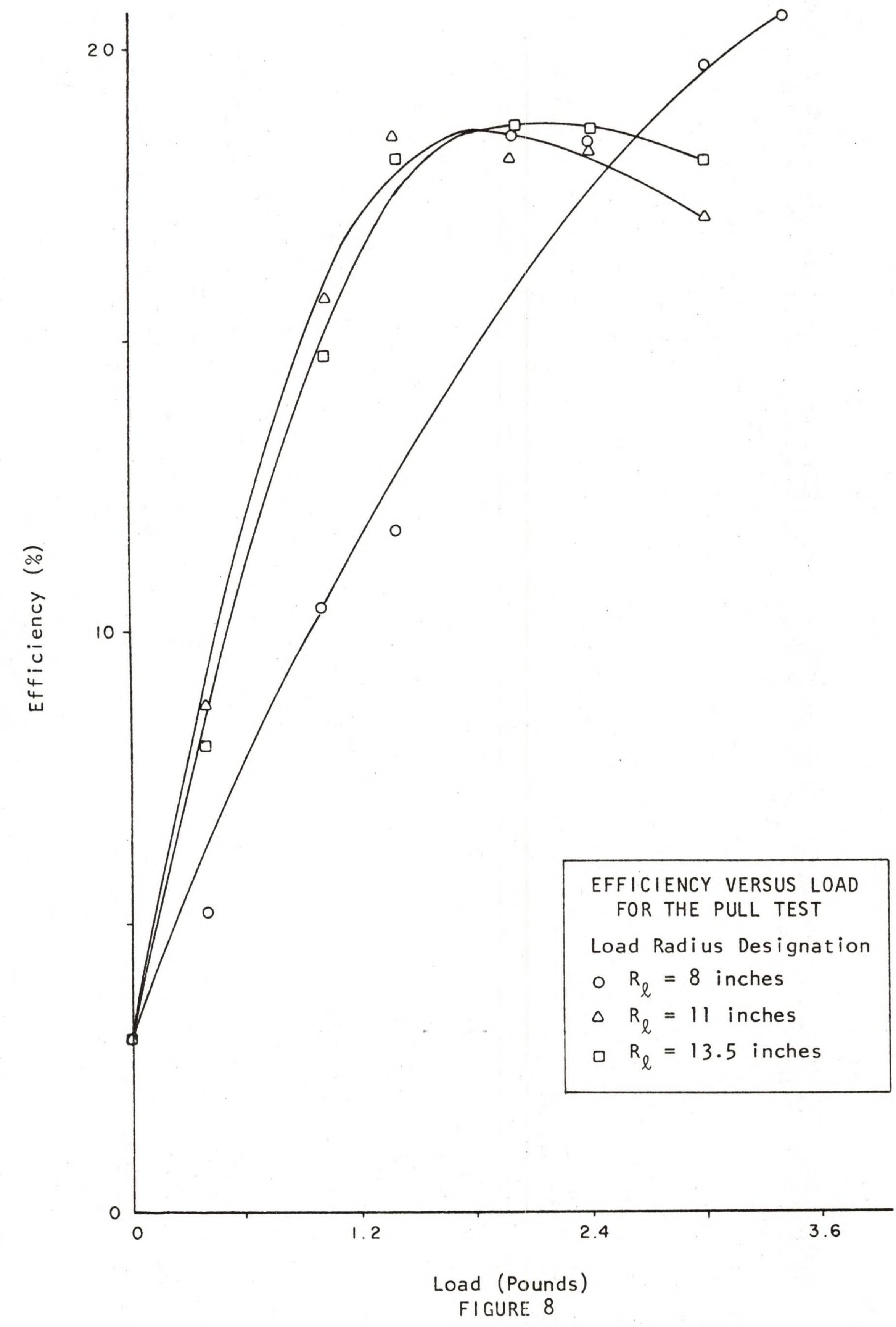

20
10
0
Efficiency (%)
0
1.2
2.4
3.6
Load (Pounds)
EFFICIENCY VERSUS LOAD
FOR THE PULL TEST
Load Radius Designation
○ R_ℓ = 8 inches
△ R_ℓ = 11 inches
□ R_ℓ = 13.5 inches

FIGURE 8

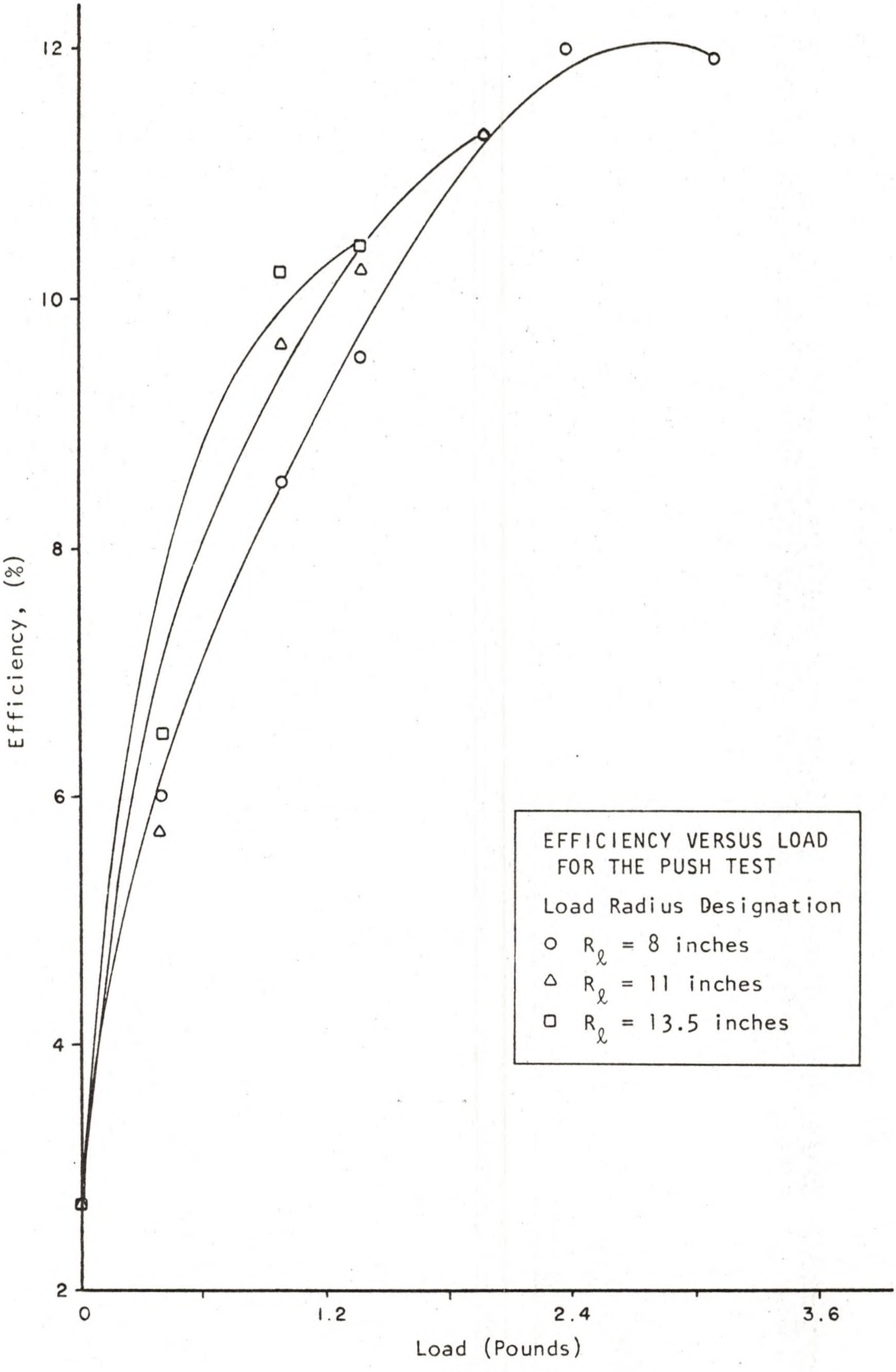

12
10
8
6
4
2
Efficiency, (%)
0
1.2
2.4
3.6
Load (Pounds)
EFFICIENCY VERSUS LOAD FOR THE PUSH TEST
Load Radius Designation
○ R_ℓ = 8 inches
△ R_ℓ = 11 inches
□ R_ℓ = 13.5 inches

FIGURE 9

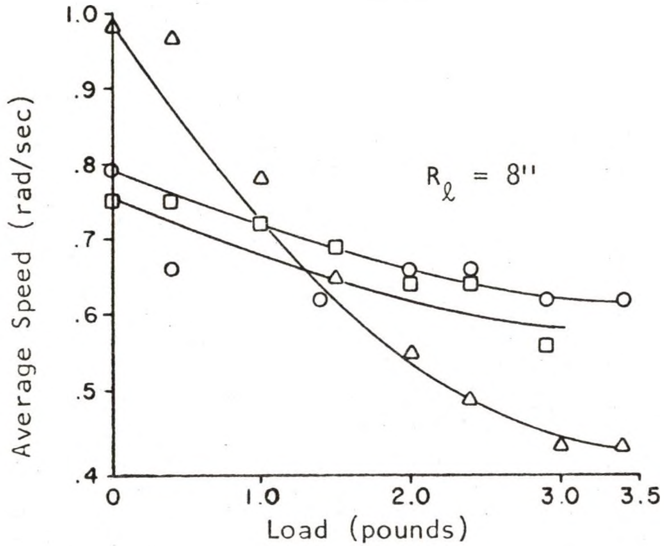


FIGURE 10

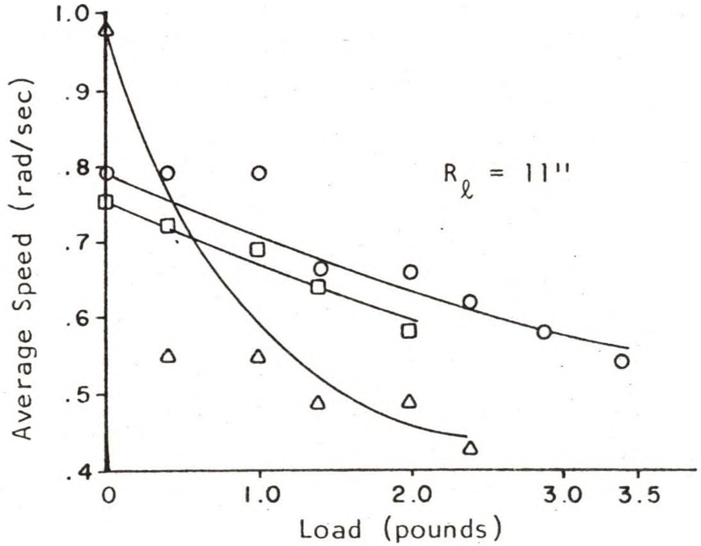


FIGURE 11

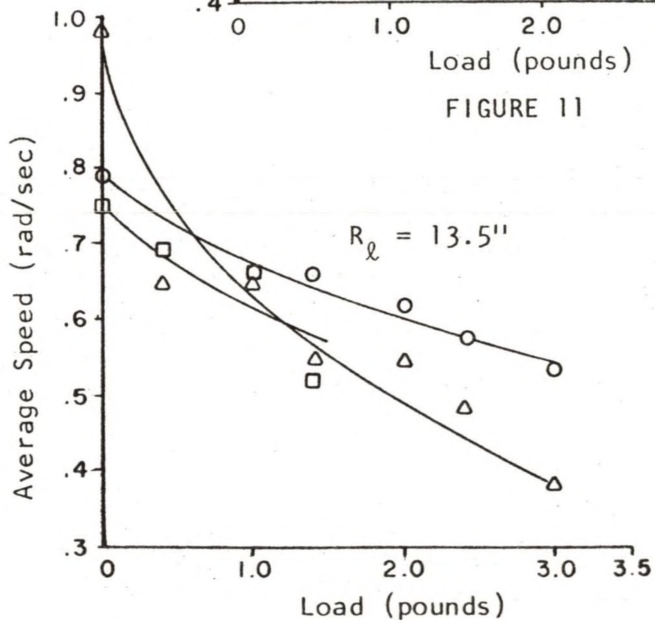


FIGURE 12

AVERAGE SPEED VERSUS LOAD FOR PULL, PUSH AND MID-RANGE TESTS

Test Designation

△ Mid-Range Test

○ Pull Test

□ Push Test

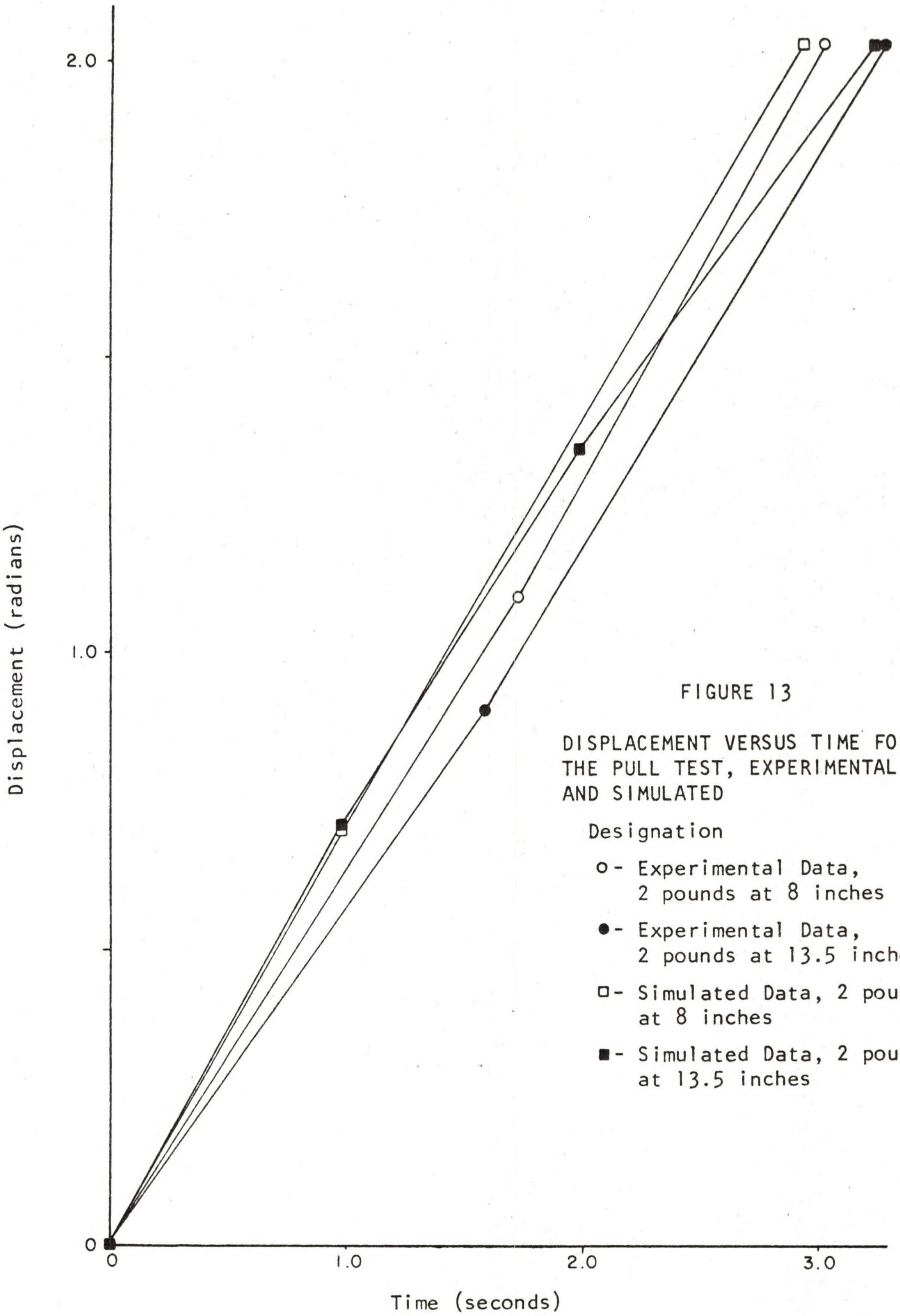


FIGURE 13

DISPLACEMENT VERSUS TIME FOR THE PULL TEST, EXPERIMENTAL AND SIMULATED

Designation

○ - Experimental Data, 2 pounds at 8 inches

● - Experimental Data, 2 pounds at 13.5 inches

□ - Simulated Data, 2 pounds at 8 inches

■ - Simulated Data, 2 pounds at 13.5 inches

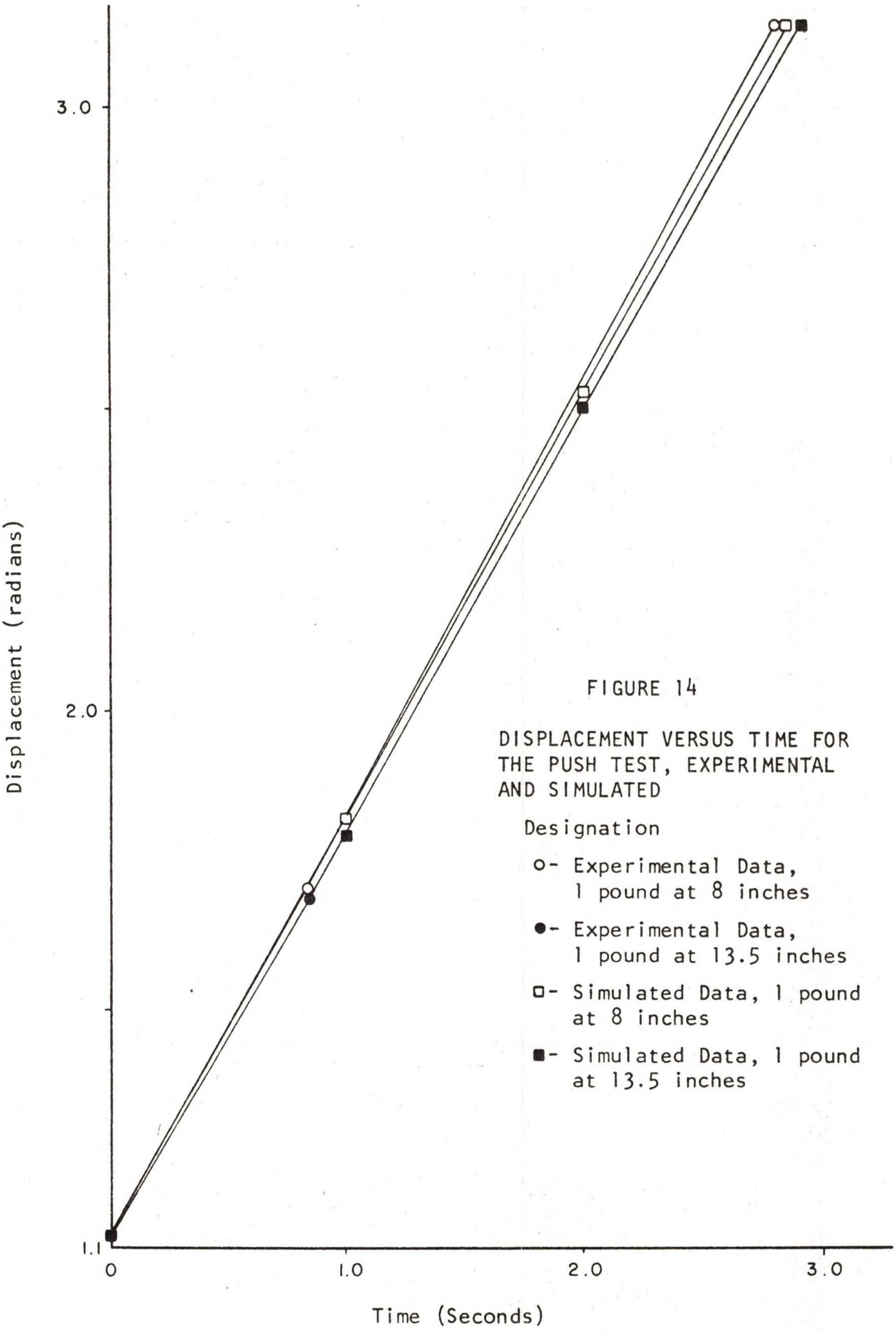


FIGURE 14

DISPLACEMENT VERSUS TIME FOR THE PUSH TEST, EXPERIMENTAL AND SIMULATED

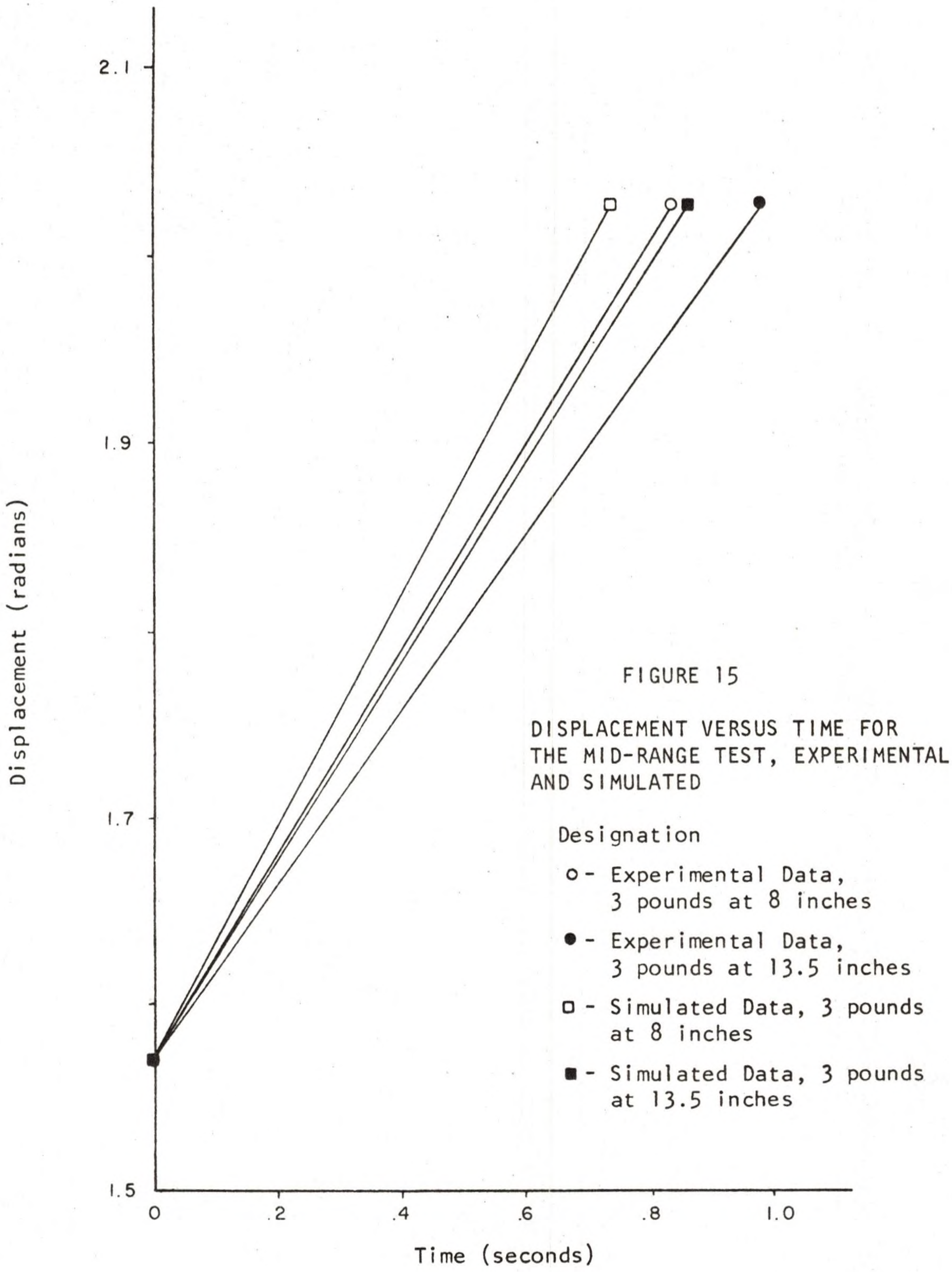


FIGURE 15

DISPLACEMENT VERSUS TIME FOR THE MID-RANGE TEST, EXPERIMENTAL AND SIMULATED efficiency is at rated load assuming the correct voltage input is used. Moving to either side of this rated load causes the efficiency to fall off sharply. Since the motor was operated at less than its rated voltage the efficiency most assuredly is lowered and contributes to a lower overall mechanism efficiency.

The next area of consideration is the drive train gearing. Visual examination of the mechanism after assembly revealed misalignment of the spur gears and a large amount of backlash in the bevel gear train caused by tolerances and mounting clearances. These mismatches cause the friction load on the motor to be greater than it should be, therefore decreasing the available output of the motor and subsequently reducing the efficiency.

An attempt was made during the design of the prototype to eliminate as much of the sliding fricition as possible. However, some efficiency loss could occur here also. All gears and rotating shafts were supported on ball bearings to minimize fricition in this area. Some sliding friction did occur between the upper and lower arm frames at the elbow pivot point and also between the back side of the bevel gear and the forearm frame.

Finally, the ball-bearing screw itself may have achieved less than the manufacturers rated efficiency due to factors such as misalignment, wear, etc.

Again, in reference to the motor manufacturers data recorded in Appendix 2, it can be seen that as the load is increased, the efficiency rises to a maximum and then falls off. This fact is in evidence in Figures 8 and 9. For a given weight load, an increase in load radius results in a higher efficiency until the peak value is reached after which the efficiency can be seen to fall off. The 11 inch radius curve should fall below the 13.5 inch radius curve in Figure 8 but does not. Causes for this error have been discussed previously with the most probable cause for error being the cycle time determination.

The trends in Figures 10-12 for average speed are as expected, with the pull or flexion tests consistently achieving the highest average speed. This behavior can be accounted for by the fact that it is much easier for the mechanism to start from full extension rather than the other two test starting positions. Also, the greatest load is placed on the model near the end of its travel thus allowing it more time to build up speed resulting in a higher average speed. Since 90 degrees of flexion is the worst possible starting position it would be expected that under increasing load the average cycle speed would fall off much more quickly. As can be seen from the figures, this does occur.

Finally, Figures 13-15 show a good correlation between the predicted displacement versus time plot and that measured experimentally. This model is desirable from a design standpoint since it allows the designer to change any number of parameters within the prototype model and then mathematically predict how the mechanism should behave. This process saves much time by easily eliminating unworkable designs.

As an example, the program was used to theoretically predict the behavior of the prototype under a load of 10 pounds located at 12 inches. A plot of displacement and velocity versus time can be seen in Figures 16 and 17. Examination of these figures shows that this loading approaches the maximum capacity for the prototype. Indeed this load can be rejected as too large already since the total time for flexion is more than seven

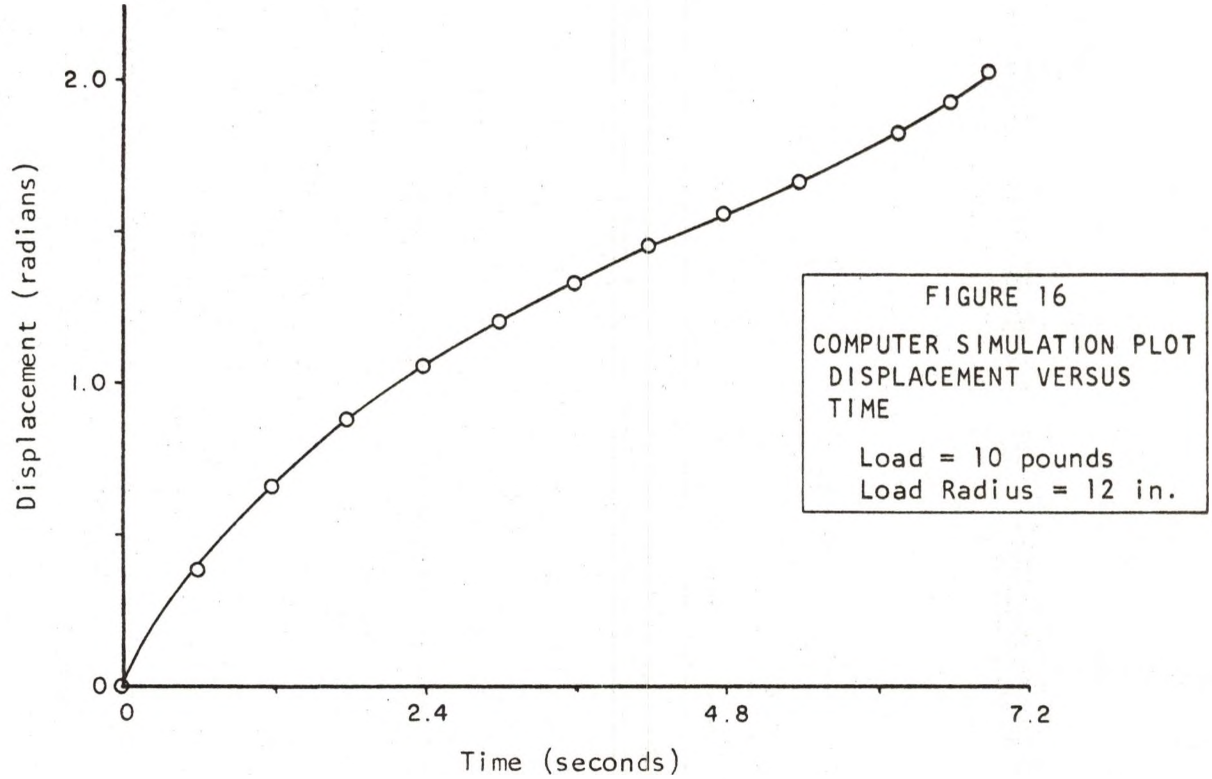


FIGURE 16

COMPUTER SIMULATION PLOT
DISPLACEMENT VERSUS
TIME

Load = 10 pounds
Load Radius = 12 in.

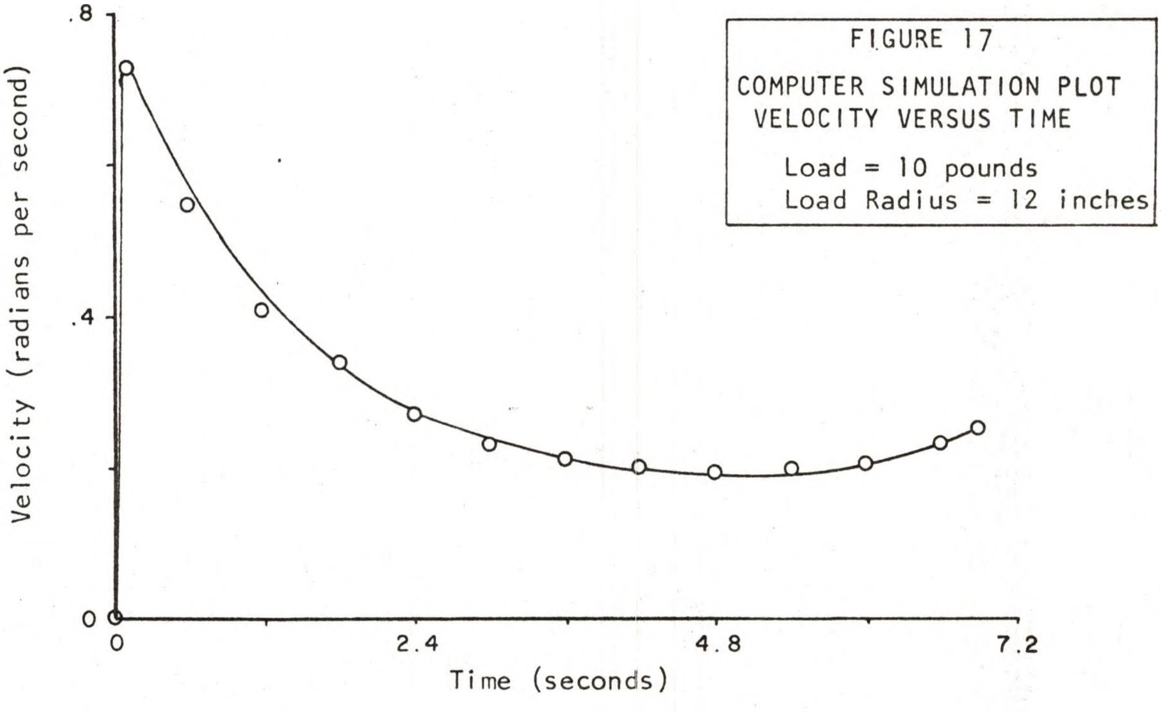


FIGURE 17

COMPUTER SIMULATION PLOT
VELOCITY VERSUS TIME

Load = 10 pounds
Load Radius = 12 inches

seconds, far exceeding the design target of full flexion in three seconds. It should also be noted that while this is theoretically a load which could be lifted, failure of other components could possibly occur terminating the lift.

The program was also used to predict a maximum torque load which could be raised by the prototype. The selection of this load was based on the examination of the change of velocity with time over the complete cycle. The maximum torque load and therefore the slowest speed occurs at θ = 90 degrees. If a great enough load is placed on the model, this speed will drop to zero and the motor will theoretically stall. Again note that this is a theoretical load. Based on the program the maximum torque load is approximately 14 foot-pounds.

CHAPTER 4

CONCLUSIONS AND RECOMMENDATIONS

It is believed that the design upon which this prototype is based has the capabilities of becoming a marketable externally powered prosthesis for above elbow amputees. The actuation system is electromechanical and, as such, should be much easier to maintain and more reliable in the field than the gas or hydraulically actuated systems found in other designs. The use of the sprocket belt-ball screw combination in the prototype is believed to be the first application of this type of drive in a powered prosthetic. The friction clutch and flex spline drives used within other known available devices have been eliminated from this design and, with them, many of their associated problems. Therefore, it is believed that this design has a distinct advantage over previous designs.

The prototype, in its present form, has some disadvantages and weaknesses which must be addressed with any subsequent development. The discussion which follows will first cover specific problems associated with the present prototype and then suggest other areas of design which should also be pursued.

The first problems encountered were with the flexible cable chain. The cable itself was strong enough to withstand all tensile loads placed on the prototype; however, some difficulties occurred in the area of attachment to the ball-screw shaft. The belt forces are great enough that simply anchoring the chain by placing a pin through the links into the screw shaft did not provide a strong enough connection. The final solution was to weld washers to the ends of the cable and anchor to the screw shaft through them. Both the proximal and distal attachment points had to have this modification since the tension end of the belt alternates between these connections depending upon whether pulling or pushing is undertaken. Another problem associated with the chain was that, because it was flexible, it did not offer a high degree of torsional stability. As the load increased the cable tended to twist with the screw motion to some extent. One way to partially eliminate this problem would have been to add an adjustable feature to the frame in order to increase the belt tension. This would have also been beneficial in the field since the cable tends to stretch somewhat during use. Both of these problems could probably have been eliminated had a small pitched metal roller chain been used in place of the cable chain. Attachment to the ball-screw would have been simpler and twisting would probably not have occurred; however, some additional weight load would have been added to the forearm. The adjustable tension feature should be added in any case.

The next area was the limitation to the range of motion. It would have been desirable to obtain a greater range of motion than 116 degrees, probably 135 degrees or more; however, size constraints limited the travel to this amount. With the given frame and elbow sprocket sizes, the ball screw had moved its total allowable length to achieve this rotation. Kinematically, for a given belt displacement, the angular displacement of the arm varies inversely with the elbow sprocket diameter. In other words, as the sprocket diameter decreases the arm displacement increases for a specific belt movement. On the other hand, the force analysis leads to the conclusion that the smaller the elbow sprocket, the less the mechanical advantage. Therefore, as the elbow sprocket diameter decreases the maximum load which can be raised also decreases proportionally. From this standpoint, a large diameter sprocket would be most desirable. Since the frame size is the more fixed of the two variables, a closer study would have to be made to determine an optimal sprocket size.

Another problem encountered was the method by which the elbow sprocket was fixed to the pivot shaft. The hollow pin used to make this attachment, sheared off several times during testing necessitating replacement. This characteristic is not entirely undesirable since it acts as an overload safety feature. In the future, it is suggested that the mechanism be designed around this weakness, possibly changing the pin diameter such that a predictable maximum safe load is not exceeded. This failure occurred at approximately seven foot-pounds.

The gear drive train, especially the bevel gears, emitted an undesirable amount of noise during operation. The gear material should not be changed from metal, since the gear stresses are believed too high for a non-metalic gear and would cause failure of the train. Gear tolerances and alignments were probably not as optimal as they could have been, and this could account for some of the noise. Also, straight bevel gears are inherently noisy at high speed operation. Therefore, it is suggested that if this type of power transmission is to be used, spiral bevel gears be substituted for the straight bevel gears since they are noted to be more silent in high speed applications. A sound absorbing cover for the arm should also be used to further reduce this gear noise.

The motor, although it gave adequate performance, is too heavy and too large to be used in a commercial application of this device. A motor of smaller size and weight producing an equal or greater torque would be beneficial in the future. If a small enough motor were found, it could be located in the forearm, thus eliminating much of the drive train, however, a higher torque motor would be needed since the motor weight would then become a portion of the load torque.

Finally, it would be desirable to make the frame narrower while still maintaining the length adjustability feature.

Near the conclusion of this project, a meeting was held with a bilateral above elbow amputee in order to get his opinions on the design. He had used other externally powered prosthetics with limited success, so any insight he could give was deemed desirable from a design standpoint. He viewed the overall design as good; but since he had become quite independent with his conventional body powered prosthesis, he suggested that the means of control of the arm and the type and control of the terminal device would also be very important features to him. Neither of these areas were specifically addressed in this design. For future work, it is suggested that a commercially available terminal device be incorporated into the design and that various means of control be investigated. Also, it would be helpful to obtain periodic input from a suitable amputee patient from the start of the project.

The two most popular methods of controlling powered prosthetics are with a mechanical switch and myoelectrically. The switch type of control generally employs two microswitches encased within the body harness of the patient. Actuation of these microswitches requires a physical movement such as hunching the shoulders resulting in subsequent prosthesis movement.

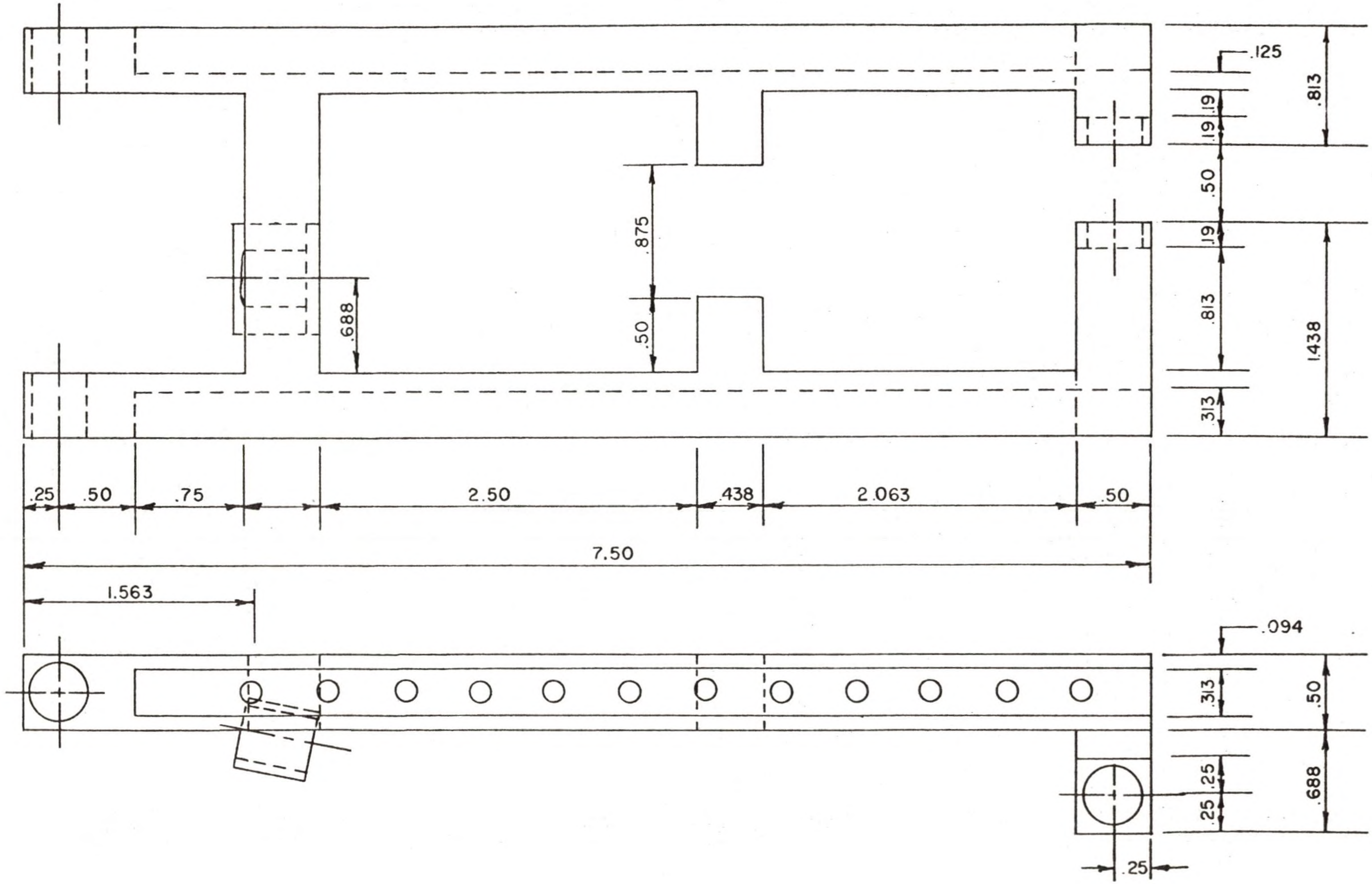
Myoelectric control involves monitoring the electric potentials of a specific group of muscles and, from this output, activating the prosthesis [10]. The primary advantage of myoelectric control over manual control is that no physical motion of the patient is required for prosthesis activation. Myoelectric control systems can be made very sensitive and therefore can be useful for controlling small movements. Also, the effort required to use this system is much less, thereby reducing patient fatigue. Surface electrodes are used in a majority of cases over a group of control muscles [11]. These muscles do not necessarily have to be in the stump; however, since these muscle fibers were originally used to control the arm, the learning process can be shortened by locating the electrodes here. Myoelectric control systems are still fairly experimental and the associated equipment is expensive; however, as technology increases, this type of control should be investigated for use with a prototype.

In conclusion, based on the research performed, it is believed that this device has a high potential and should be pursued further.

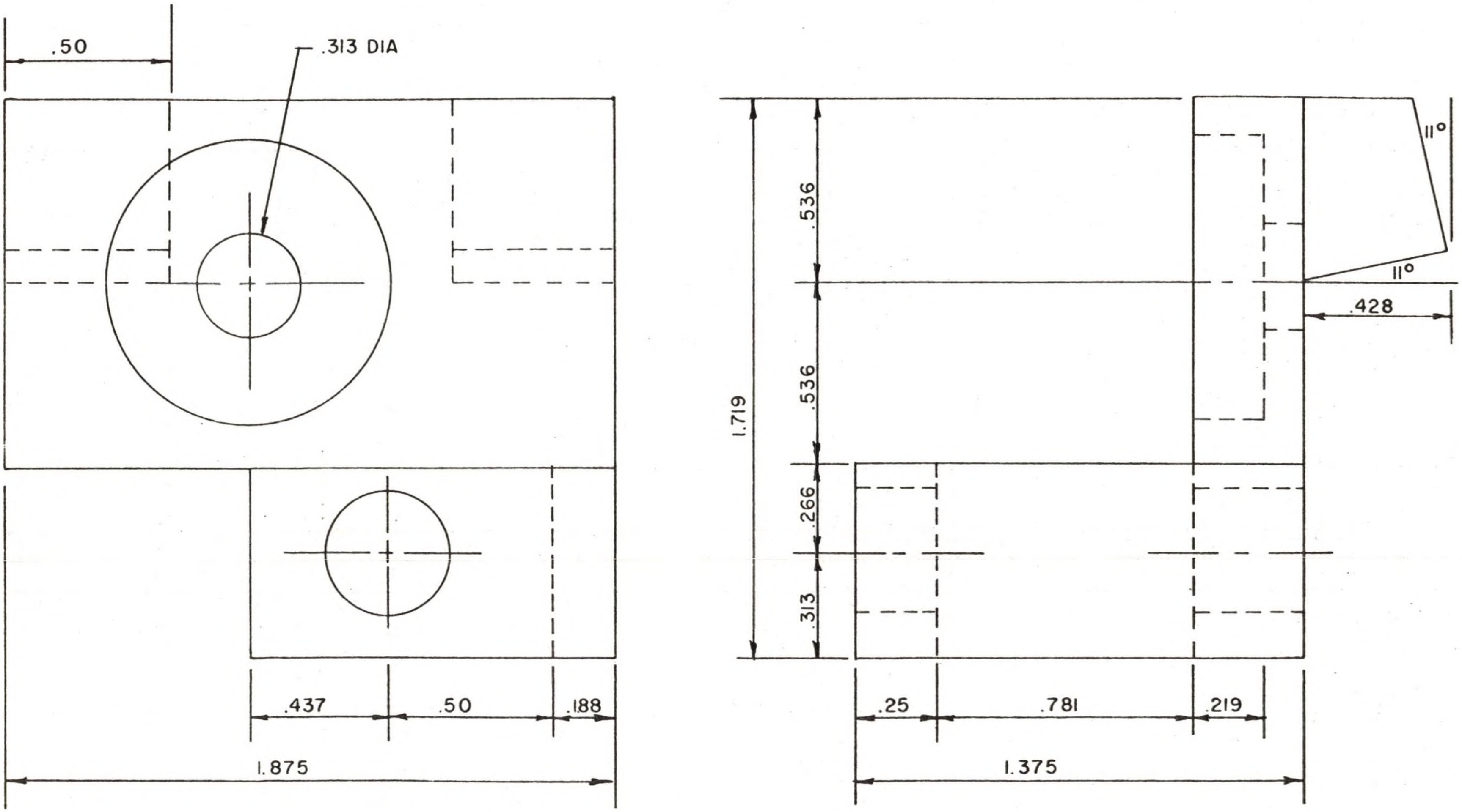
APPENDICES

APPENDIX 1

The following pages are copies of the shop drawings used to construct the prototype.

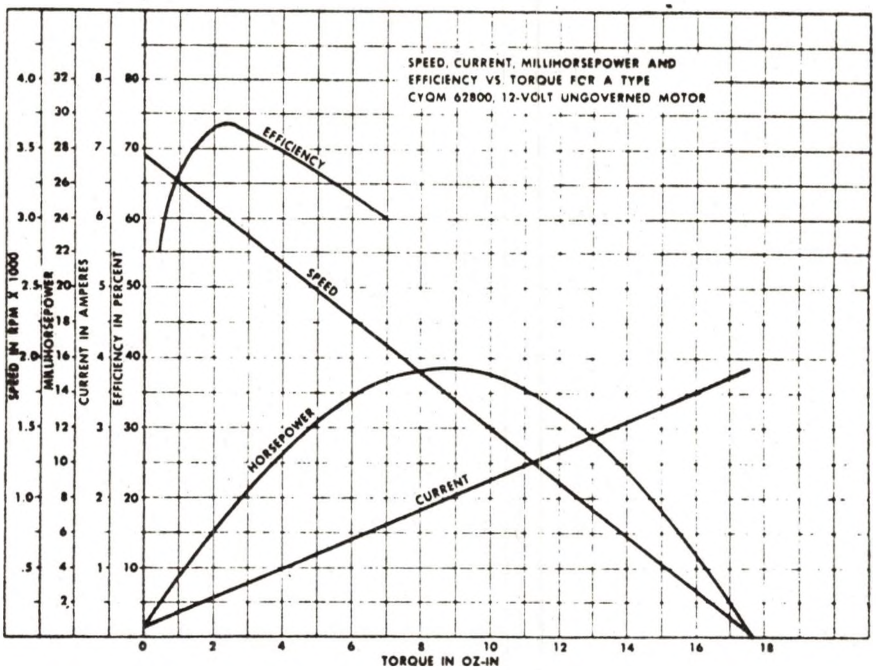


.125
.813
.19
.19
.19
.50
.19
.875
.688
.50
.813
1.438
.313
.25
.50
.75
2.50
.438
2.063
.50
7.50
1.563
.094
.313
.50
.25
.25
.688
.25


.50
.313 DIA
.437
.50
.188
1.875
1.719
.536
.536
.266
.313
11°
11°
.428
.25
.781
.219
1.375

APPENDIX 2

The following curves are a reprint of the manufacturer's ratings for the motor used in the prototype. The information was supplied by Barber-Colman Company, Electro-Mechanical Products Division, Rockford Illinois.



This is a listing of all of the components used in the prototype by manufacturers' name and part number.

Boston Gear Company

Gears
- bevel pinions - G486Y-P
- bevel gear - G486Y-G
- spur pinion - Y4830
- spur gear - Y4860

W. M. Berg, Inc.

Bearings
B1-25
B1-39
Flexible Coupling
CC5-19-P
Sprockets
3MP19A-40
3MP19S-15
Cable Chain
3CCF-E

Warner Electric Company

Ball Bearing Screw
Warner 3/16" Diameter, 0.05 lead

Barber-Colman Company

D-C Motor
CYQM 62800

APPENDIX 3

MOTION SIMULATION PROGRAM

The following two pages contain a reprint of the computer program used to simulate the motion of the prototype. Basically the program uses a subroutine to solve the differential equation presented in Chapter 2.

Symbol notation:

FL	- is the weight load (F_ℓ)
RL	- is the weight load radius measured from the elbow pivot (R_ℓ)
ML	- is the inertial load m R_ℓ^2
XO,YO,XN,YN	- are iterative values of the displacement and velocity of the arm.
H	- is the time step used in solving the equation

```
      COMMON FL,ML,RL
      DIMENSION X(300),Y(300)
      DO 13 LL=1,3
      DO 12 L=1,4
      MM=0.
      XO=0.
      YO=0.
      XN=0.
      YN=0.
      FL=1.*LL
      RL=6.+L*3.
      WRITE(6,20)FL,RL
      ML=(FL/386.4)*RL*RL
      H=.0001
    1 DO 8 M=1,120
      IF(MM.GT.20.) GO TO 9
      DO 7 N=1,3
      CALL RUNGE(H,XO,YO,XN,YN)
      XO=XN
      YO=YN
      X(N)=XN
      Y(N)=YN
    7 CONTINUE
      WRITE(6,10)X(1),Y(1),X(2),Y(2),X(3),Y(3)
      GO TO 11
    9 DO 5 N=1,300
      CALL RUNGE(H,XO,YO,XN,YN)
      XO=XN
      YO=YN
      X(N)=XN
      Y(N)=YN
    5 CONTINUE
      WRITE(6,10)X(100),Y(100),X(200),Y(200),X(300),Y(3
00)
      IF(MM.EQ.120.) GO TO 12
   11 MM=MM+1.
    8 CONTINUE
   12 CONTINUE
   13 CONTINUE
   10 FORMAT(3X,F7.4,3X,F7.4,6X,F7.4,3X,F7.4,6X,F7.4,3X
,F7.4)
   20 FORMAT(//3X,'THE LOAD =',F4.0,3X,'LOCATED AT',F4.
0,1X,'INCHES',//)
      END
```

```
      SUBROUTINE RUNGE(H,X0,Y0,XN,YN)
      COMMON FL,RL,ML
      A1=H*Y0
      B1=H*(104.55-218.9*Y0-2.51*SIN(X0)-FL*RL*SIN(X0))
     /(.02755+ML)
      X=X0+.5*A1
      Y=Y0+.5*B1
      A2=H*Y
      B2=H*(104.55-218.9*Y-2.51*SIN(X)-FL*RL*SIN(X))/(.
     02755+ML)
      X=X0+.5*A2
      Y=Y0+.5*B2
      A3=H*Y
      B3=H*(104.55-218.9*Y-2.51*SIN(X)-FL*RL*SIN(X))/(.
     02755+ML)
      X=X0+A3
      Y=Y0+B3
      A4=H*Y
      B4=H*(104.55-218.9*Y-2.51*SIN(X)-FL*RL*SIN(X))/(.
     02755+ML)
      XN=X0+(A1+2.*A2+2.*A3+A4)/6.
      YN=Y0+(B1+2.*B2+2.*B3+B4)/6.
      RETURN
      END
```

Appendix 4

TEST EQUIPMENT LIST

1. Weston D.C. Voltmeter, Model 622, Serial Number 15077, Weston Electrical Instrument Comp., Newark, N.J., USA.

2. Hoyt D.C. Ammeter, Type 515, Hoyt Electrical Instrument Works, Penacook, N.H., USA.

3. Varian Aerograph Strip Chart Recorder, Series G-2000, Varian Aerograph, Walnut Creek, California.

4. Electrogoniometer, Constructed at UND, 10K and 50K ohm variable resistors purchased from Radio Shack locally.

BIBLIOGRAPHY

1. Clynes, M., D.Sc., Milsum, J.H., Sc.D., Biomedical Engineering Systems, Vol. 10, McGraw-Hill, New York, 1970, pp.515, 527, 530-531.

2. Lambert, T.H., Davies, R.M., "Hydraulic Actuation for Upper Limb Prostheses," Proceedings of the 25th Annual Conference on Engineering in Medicine and Biology, Vol. 14, 1972, pp 99.

3. Lewis, E.A., M.A., R.P.T., Sheredos, C.R., B.S., R.P.T., Sowell, T.T., B.S., R.P.T., "Clinical Application Study of Externally Powered Upper-Limb Prosthetic Systems: The VA Elbow, The VA Hand, and the VA/NU Myoelectric Hand Systems," Bulletin of Prosthetics Research, BPR 10-24, Fall 1975, pp. 52-54, 82, 92, 125-127.

4. LeBlane, M.A., "Clinical Evaluation of Externally Powered Prosthetic Elbows," Artificial Limbs, Vol. 15, No. 1, Spring 1971. pp. 70-77.

5. LeBlane, M.A., "Improved Prostheses for Arm Amputees?" Proceedings of the 23 rd Annual Conference on Engineering in Medicine and Biology, Vol. 12, 1970, pp 170.

6. Krump, M., "Design of an Externally Powered Elbow Mechanism for Use in a Prosthesis," Honors Project Report, Department of Mechanical Engineering, University of North Dakota, 1978.

7. Shigley, J.E., Mechanical Engineering Design, 3rd ed., McGraw-Hill, New York, 1977.

8. Hutinger, P.W., "Construction and Utilization of a Simple Electrogoniometer," Selected Topics on Biomechanics, Proceedings of the C.I.C. Symposium on Biomechanics, October 19-20, 1970, pp.92-94.

9. Adrian, M.J., "An Introduction to Electrogoniometry," Kinesiology Review, 1968, pp.14-16.

10. Scott, R.N., "Myoelectric Control of Prostheses and Orthoses," Bulletin of Prosthetics Research, Spring 1967, pp. 93-94.

11. Scott, R.N., "Myoelectric Control Systems," Advances in Biomedical Engineering and Medical Physics, Vol. 2, Interscience Publishers, New York, 1968, pp. 45-69.

12. Barber-Colman Company, "Commercial D-C Motors", Electro-Mechanical Products Division, Manufacturer's Published Data.

13. Beer, Johnston, Mechanics for Engineers: Dynamics, 2nd ed., McGraw-Hill, New York, 1962.

14. Warner Electric, "Ball Bearing Screws and Splines," Manufacturer's Published Data.

Design and Testing of an Externally
Powered Elbow Prosthesis

David Edward Weston, M.S.

The University of North Dakota, 1980

Faculty Advisor: Doctor J. P. Sadler

ABSTRACT

There continues to be a need for an externally powered prothesis which can be used by above elbow amputee patients who cannot effectively operate a conventional body powered prosthesis. This device must be reliable, economically constructed, and easily maintained in the field. A device employing a drive mechanism powered by a small DC motor has been designed to meet this need.

The device is based on an inversion of the belt driven pulley system and is a continuation of previous work employing this mechanism. A prototype was designed using this system in a size suitable for patient application. The model was constructed from commerically available parts and some shop fabrication. Once constructed, a laboratory testing program was devised to subject the prototype to typical tasks it would be required to perform in the field. The test results are included in the thesis. Also included are the kinematic and force analyses of the model and a computer program, based on the design equations, written to simulate the motion of the device under load. A comparison between the simulated and experimental results is also presented.

The major intent of this project was to design and test a reliable externally powered above elbow prosthesis from commerically available parts. The design has proven to be a viable concept and should be pursued further based on the recommendations given in this thesis.