



12-2003

## **Analysis and Design of a Gear Shifting Mechanism for Transmission Based Actuators**

Kalyana Bhargava Ganti  
*University of Tennessee - Knoxville*

Follow this and additional works at: [https://trace.tennessee.edu/utk\\_gradthes](https://trace.tennessee.edu/utk_gradthes)



Part of the [Mechanical Engineering Commons](#)

---

### **Recommended Citation**

Ganti, Kalyana Bhargava, "Analysis and Design of a Gear Shifting Mechanism for Transmission Based Actuators. " Master's Thesis, University of Tennessee, 2003.  
[https://trace.tennessee.edu/utk\\_gradthes/1943](https://trace.tennessee.edu/utk_gradthes/1943)

This Thesis is brought to you for free and open access by the Graduate School at TRACE: Tennessee Research and Creative Exchange. It has been accepted for inclusion in Masters Theses by an authorized administrator of TRACE: Tennessee Research and Creative Exchange. For more information, please contact [trace@utk.edu](mailto:trace@utk.edu).

To the Graduate Council:

I am submitting herewith a thesis written by Kalyana Bhargava Ganti entitled "Analysis and Design of a Gear Shifting Mechanism for Transmission Based Actuators." I have examined the final electronic copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.

William R. Hamel, Major Professor

We have read this thesis and recommend its acceptance:

Gary V. Smith, Arnold Lumsdaine

Accepted for the Council:

Carolyn R. Hodges

Vice Provost and Dean of the Graduate School

(Original signatures are on file with official student records.)

To the Graduate Council:

I am submitting herewith a thesis written by Kalyana Bhargava Ganti entitled "Analysis and Design of a Gear Shifting Mechanism for Transmission Based Actuators." I have examined the final electronic copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.

William R. Hamel  
Major Professor

We have read this thesis  
and recommend its acceptance:

Gary V. Smith

Arnold Lumsdaine

Accepted for the Council:  
Anne Mayhew  
Vice Provost and Dean  
of Graduate Studies

(Original signatures are on file with official student records.)

ANALYSIS AND DESIGN OF A GEAR SHIFTING MECHANISM  
FOR TRANSMISSION BASED SERVO-ACTUATORS

A thesis submitted for the Master of  
Science degree

The University of Tennessee, Knoxville

Kalyana Bhargava Ganti

December 2003

Copyright © Kalyana Bhargava Ganti, 2003

All rights reserved.

*To*

Vani

## ACKNOWLEDGMENTS

I would like to thank my major Professor Dr. Bill Hamel for all the support he gave me during my master's program. I think I have evolved both as an engineer, and as well as a person, under his able guidance, for which I will be grateful to him for the rest of my life. He gave me a new perspective of looking at things. I also want to thank my graduate committee members Dr. Smith and Dr. Lumsdaine for their support and encouragement.

Next I would like to acknowledge all the members who have worked with me on the TBA project including Dr. Douglass, Dr. Seewong Kim, Renbin Zhou, Arun Srikantiah, Sriram Sridharan and Kelley Brown. It's been a pleasure working with you gentlemen. I would also like to thank the other members of the Robotics and Electromechanical Labs. Ge Zhang and Mark Noakes for all the help. I would like to thank DOE-NETL for sponsoring the TBA project.

I would also like to thank Dr. Lyne, who was my initial advisor for introducing me to Dr. Hamel. I am grateful to all of my teachers since my childhood, notably Bhaskara Sharma, Dr. Venkateshwara Rao, Dr. Suryananaraya Rao, faculty members at University of Tennessee, Knoxville, who have contributed to my tree of knowledge for the past 22 years.

I would like to express my gratitude to my parents Bhaskara Sharma and Kamala for all the love, support and guidance. I would like to thank my younger brother Aditya (Bujji) for many things, but most importantly for introducing me to the philosophy of Ayn Rand. I would like to thank my grandparents and the rest of my family viz. Vani, Lakshmi (Babbu), Sudha, Madhu, Phani, Sirisha, Anusha et al for all their love and support.

I would like to show appreciation to all of my friends including Hari, Harish, Srinivas, Jaipal, S.V. Srinivas, Fayaz, Madhu, Murali, Mallesham, Prashanth, Raju, Ravi, Sarath, Bhargav, Vasani, Sharat, Vasuki, Manohari, Vasavi, Harika, Saritha, Anu, Deepti, Vijita and rest of my classmates from my B.E. and high school. We had our moment's guys. I would like to convey my special thanks to Sirisha for all the support during my stay at Knoxville and making it a pleasurable one. Lastly, I would like to offer my profound gratitude to my cousin Vani by dedicating this thesis to her, for believing in me when nobody else did. Thank you very much, Potti.



## ABSTRACT

The fundamental idea of Transmission Based Actuators (TBA) is to incorporate a multi-speed transmission to spread a servomotor's torque speed characteristics across a wider output speed range. TBA uses multi-speed transmissions such that heavy, high-torque motors can be traded for high-speed, reduced mass motor-transmission combinations. TBA design consists of a D.C. Brushless motor, discrete gear transmission and a fixed reduction. Of these, the fundamental to the proof of principle of TBA is the design of the discrete gear transmission.

Basically the DVT is a three-speed gearbox consisting of 3 sets of epicyclic gears. Every planetary gear set is a two-degree of freedom system. In the present design one input is always the sun gear and the other is the ring gear. The motor gives one input to the planetary gear and the second input to the planetary gear set is obtained by locking the band brakes such that the ring gear speed is zero. Three different speeds are obtained by selectively and synchronously locking the three annular ring gears using corresponding external flexible band brakes. The complete dynamic model of the Gear Shifting Mechanism (GSM) has been developed and the braking torque has been estimated.

The gear shifting mechanism primarily consists of a band brake, a series of mechanical linkages and electrical actuator. It is used to stop the ring gear, rotating at a very high speed and hold it to obtain the desired gear ratio. . Then a brake test stand has been designed and built to test the functionality of the GSM. The effective braking torque and the system time constant were measured for both dry and wet conditions. The obtained test results have been analyzed with respect to the predicted simulated results.

## TABLE OF CONTENTS

1. INTRODUCTION.....	1
1.1 OVERVIEW .....	1
1.2 BACKGROUND.....	4
1.3 BASELINE REQUIREMENTS .....	7
2. DESIGN OF TRANSMISSION BASED SERVO-ACTUATORS .....	9
2.1 CONCEPT.....	9
2.2 OVERVIEW OF GEAR TRAINS.....	12
2.3 EPICYCLIC OR PLANETARY GEAR TRAIN.....	13
2.4 DESIGN CONFIGURATION .....	15
3. ANALYSIS OF TRANSMISSION BASED SERVO-ACTUATORS ..	17
3.1 KINEMATICS OF DVT.....	17
4. DESIGN OF GEAR SHIFTING MECHANISM.....	26
4.1 INTRODUCTION .....	26
4.2 BAND BRAKES.....	27
4.3 ACTUATION FORCE REQUIREMENTS.....	29
4.3 ACTUATION FORCE REQUIREMENTS.....	32
4.4 FRICTION MATERIALS FOR BAND APPLICATIONS.....	41
4.5 DESIGN OF BAND BRAKE .....	43
4.6 BAND CONSTRUCTION AND PROCESSING .....	45
4.7 BAND FAILURE MODES.....	46
4.8 SURVEY OF DIFFERENT ACTUATORS.....	48
4.9 METHODS OF ACTUATION AND RELEASE.....	50
5. PROTOTYPE EXPERIMENTAL EVALUATION.....	56
5.1 BRAKE TEST STAND.....	56
5.2 EXPERIMENTAL RESULTS.....	60
5.3 CONCLUSIONS .....	66
6. SUMMARY AND FUTURE WORK.....	67
6.1 SUMMARY .....	67
6.2 FUTURE WORK.....	69
BIBLIOGRAPHY.....	71
VITA .....	74

## LIST OF TABLES

Number.....	Page
Table 1.1 Characteristics of Titan II.....	8
Table 5.1 Brake test datasheet 1.....	61
Table 5.2 Brake test datasheet 2.....	62
Table 5.3 Brake test datasheet 3.....	63
Table 5.4 Brake test datasheet 4.....	64

## LIST OF FIGURES

Number.....	Page
Figure 1.1 Complete Design of TBA.....	2
Figure 1.2 Overall Comparison of Actuator Technologies.....	5
Figure 1.3 Machinery Comparison: Power and Response Time.....	6
Figure 2.1 Torque-Speed Characteristics of TBA vs. Fixed Ratio Motor.....	11
Figure 2.2 Cutout View of Single Stage Planetary Gear.....	14
Figure 2.3 Schematic of Single Stage Planetary Gear Set.....	14
Figure 2.4 3-Speed DVT.....	15
Figure 4.1 Band Wrap Angle.....	30
Figure 4.2 Differential Brake.....	31
Figure 4.3 Band Tension.....	32
Figure 4.4 Simple Band Brake.....	36
Figure 4.5 Band Brake for Rotation in Both Directions.....	39
Figure 4.6 Forces Acting on a Band Brake.....	40
Figure 4.7 Band Brake.....	44
Figure 4.8 Conceptual Representation of a Simple Lever Mechanism.....	51
Figure 4.9 Conceptual Design of Gear Shifting Mechanism.....	53
Figure 4.10 Plot of Actuator Force w.r.t. Angle of Contact.....	54
Figure 4.11 Detailed Design of Gear Shifting Mechanism.....	54
Figure 4.12 Exploded View of the Band Brake Assembly.....	55
Figure 5.1 Brake Band Static Test Stand.....	57
Figure 5.2 Static Test Stand.....	58
Figure 5.3 Close-Up Views of the Crank, Yoke and the Other Linkages.....	59
Figure 5.4 Band Brake Test Procedure.....	60
Figure 5.5 Plots for Experimental Results Obtained for Dry Band Brake.....	65
Figure 5.6 Plots for Experimental Results Obtained for Wet Band Brake.....	65

## **INTRODUCTION**

### **1.1 Overview**

This section introduces the concept, design and modeling philosophy of Transmission-Based Servo Actuators (TBA). TBA is a new approach for extending the operating range of electrical servo actuators in applications such as high payload robotics and remote handling systems. The fundamental idea of TBA is to incorporate a multi-speed transmission to spread a servomotor's torque speed characteristics across a wider output speed range. TBA uses multi-speed transmissions such that heavy, high-torque motors can be traded for high-speed, reduced mass motor-transmission combinations [1].

TBA design consists of a D.C. Brushless motor, discrete gear transmission and a fixed reduction. Of these, the fundamental to the proof of principle of TBA is the design of the discrete gear transmission (Figure 1.1). The transmission can be either Continuously Variable Transmission (CVT) or a Discrete Speed Variable Transmission (DVT). This work is based on the concept of Discrete Speed Variable Transmission.

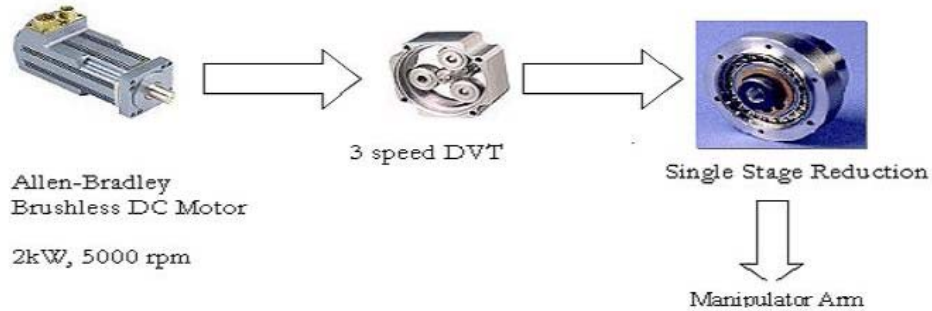


Figure 1.1 Complete Design of TBA

Initially the conceptual design of the DVT has been developed which forms as a foundation to the entire TBA project. Then the kinematic and dynamic analysis of the DVT has been performed in order to determine the rigid-body dynamic behavior of the system.

Basically the DVT is a three-speed gearbox consisting of 3 sets of epicyclic gears. An epicyclic or planetary gear consists of a sun gear (S), a concentric annulus gear or ring gear (A or R) and planet gears, which meshes with both S and A. Every planetary gear set is a 2 degrees of freedom system i.e., it should be provided with two inputs to obtain a determined output. In the present design, we have 6 degrees of freedom corresponding to 3 sets of epicyclic gears. The equivalence of the 3 sun gears, which are connected by a common shaft, provides 2 constraint equations whereas the carrier to ring connection provides another 3 constraint

equations. So the entire system is again reduced to 2 degrees of freedom system, where one input is always the common sun gear shaft and the other input is any one of the three ring gears. The motor gives one input to the planetary gear and the second input to the planetary gear set is obtained by locking the band brakes such that the ring gear speed is zero. Three different speeds are obtained by selectively and synchronously locking the three annular ring gears using corresponding external flexible band brakes. It is analogous to gear shifting in an automobile. Hence the name “Gear Shifting Mechanism (GSM)”. This mechanism should be able to produce the necessary braking torque (around 25 N-m) with minimum modular space and weight requirements.

Any attempt to design a system must begin with a prediction of its performance before the system itself can be designed in detail or accurately built. Such prediction is based on a mathematical description of the system’s dynamic characteristics. This mathematical description is called a mathematical model. For many physical systems, useful mathematical models are described in terms of differential equations [2]. The complete dynamic model of the GSM has been developed and the design carried out.

Then a test stand was designed and built to test the functionality of the GSM. The effective braking torque and the system time constant were measured for both dry and wet conditions. The obtained test results have been analyzed with respect to the predicted simulated results. This present work is the application of

standard analytical and simulation techniques available in the literature to design a Gear Shifting Mechanism for TBA.

## **1.2 Background**

In applications such as high payload robotics and remote handling systems that require high payload and speed capacity, the actuating systems are almost always hydraulic in nature. The hydraulic actuators provide a much higher power density (i.e., power-to weight ratio) and much higher torque density (i.e., torque-to weight ratio) than that is achievable with other conventional sources. But compared to an electric drive, hydraulic actuators are mechanically complex and require periodic maintenance to keep them in working order. These characteristics like, torque density and power density, are especially important to applications such as robotics where actuators may add to the overall payload of the system; in other words, in most cases robot actuators are responsible for lifting the robot arm and external payload, plus themselves and any actuators “downstream”[1]. An overall comparison of various actuators is shown in Figure 1.2 [3].

The graph in the figure summarizes some of the characteristics of a pure hydraulic actuator, a typical electrical actuator, ARM<sup>®</sup> modular Electro - Mechanical Actuator (EMA), and an Electro-Hydraulic Actuator (EHA). The characteristics presented on a scale of 1 to 10 with 10 being the optimal characteristic, are payload-to-weight, dexterity, repairability etc. In almost all the parameters the electro-hydrostatic is equal to or better than the other actuator



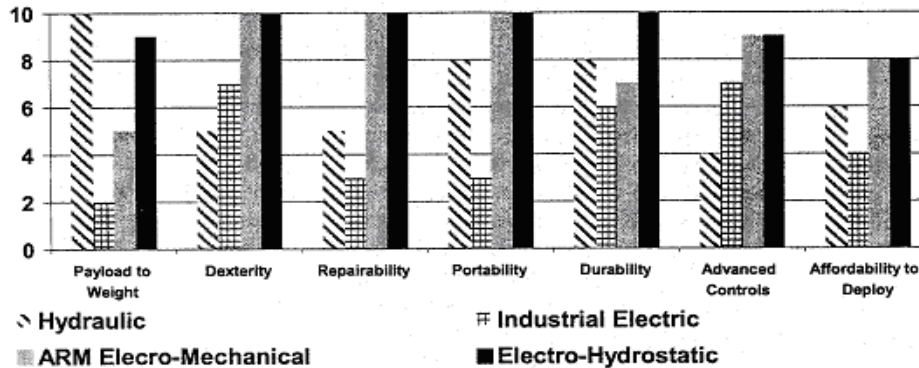


Figure 1.2 Overall Comparison of Actuator Technologies

technologies. Hence if all these characteristics can be realized, then the EHA will provide an overall cost savings as estimated by the aircraft industry. Next a nice comparison of how various types of machinery compare is shown in Figure 1.3, which is a graphical representation of several types of machinery concerning their power capacity and reciprocal of response time [4].

“There are many application domains requiring large scale or high-performance manipulators, and some of these that have currently brought attention to potential robot applications are the environmental cleanup operations within the U.S. Department of Energy nuclear complex. Many environmental restoration projects will involve the use of robotics and remote handling systems- especially when radiation exposure levels are high. The scale of typical operations covers a very large range in terms of the payloads, which must be lifted, and the distance

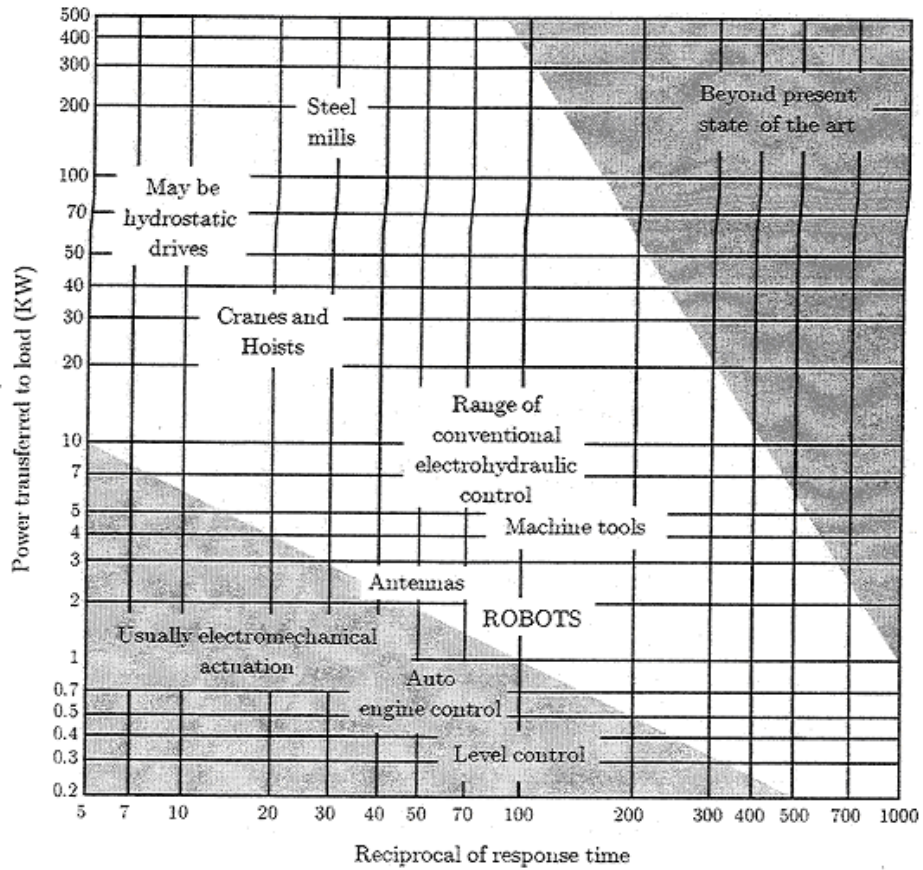


Figure 1.3 Machinery Comparison:  
Power and Response Time

over which they must be moved. In applications associated with decontamination and decommissioning of old facilities, mobile robot work systems with manipulator payloads of at least 100 kg are considered essential” [5].

In most design scenarios, one would prefer to use electrical servoactuators in the interest of cost and reliability. Servomotor improvements, primarily in the area of rare earth magnetic materials have increased motor power density by a factor of 3 over the past 10 years. It is not likely that improvements in motor technology alone will close the fundamental gap between electric and hydraulic actuation [5]. Hamel and Widner [5] introduced the Transmission Based Servo Actuator (TBA) concept for expanding the torque and speed range of electric actuators.

### **1.3 Baseline Requirements**

The baseline for the TBA is the Titan series Electro-hydraulic manipulator from Schilling Development Corporation. It has high manipulative power and is extensively used in DOE projects. The Schilling Titan II is a commercially available robot that was developed originally for under sea applications. Commonly used on submersibles, it has also found favor in the nuclear industry due to its titanium construction, and high performance. The Titan II has a 6-foot reach, and a 240 lb. Payload at full extension. It is also a very agile robot, and very precise in operations. It is powered by a hydraulic pump capable of providing 5 gallons per minute at 3000 psi. The Titan II is a very versatile robot for

manipulating heavy objects and demolition. It can be mounted to a vehicle as needed for mobile deployments (Table 1.1).

Total payload is defined in terms of the shoulder pitch actuator of the Schilling Titan II arm, operating at maximum payload and speed which includes: manipulator arm weight of 200 lbs (90.72 kg) length of 71 inches (~ 1.8m) and a payload of 240 lbs (108.86 kg). With the assumption of a rigid lower arm link, the load torque can be calculated as in equation (1.1) [6]:

$$T_L = \left( M_a + \frac{7 M_p}{12} \right) l \ddot{\theta} + (M_a + 0.5 M_p) l g \cos \theta \quad (1.1)$$

where  $T_L$  = Load torque,  $M_a$  = Mass of Baseline,  $M_p$  = Mass of Load,

$l$  = length of baseline,  $g$  = acceleration due to gravity,  $\theta$  = joint position

$\ddot{\theta}$  = joint acceleration

Table 1.1 Characteristics of Titan II

Manipulator Motion	Max. Torque	Max. Speed	Max. Power
Shoulder Pitch	350Ft-lb	20deg/s	0.7 hp or 500 W

## **DESIGN OF TRANSMISSION BASED SERVO-ACTUATORS**

### **2.1 Concept**

This chapter discusses the concept and design of Transmission-Based Servo Actuators (TBA) in detail. As seen earlier, TBA is a new concept for the purpose of increasing the torque and speed range of an electric actuator in such a way that the power and torque densities of electric actuators are on the order of hydraulics. Electrical actuators based on multi-speed transmissions will inherently be operable over a wider load and speed range because of their capability of exchanging speed and torque.

The fundamental idea of TBA is to incorporate a multi-speed transmission to spread a servomotor's torque speed characteristics across a wider output speed range. TBA uses multi-speed transmissions such that heavy, high-torque motors can be traded for high-speed, reduced mass motor-transmission combinations. Use of transmission-based actuators can eliminate the need to employ centralized actuators. Use of distributed actuators can substantially simplify fundamental manipulator design (hence cost) and also expand the feasible scope of electric drives to a much broader range of applications.

“When a fixed ratio gearing system is designed for an actuator, the proper choice of the transmission ratio is centered around meeting the payload and speed requirement for the joint. In some cases, the transmission ratio is selected such

that maximum acceleration is achieved for the nominal loading conditions. Systems often have varying loading conditions as well as varying geometry (i.e., arm orientation), and if the design space is limited to a single transmission ratio, the ratio is generally designed to meet torque and speed requirements at a position near the center of workspace” [7].

Designing an actuator system that has multiple gear ratios available makes the system much more robust. Each ratio can be optimized with respect to a given payload-response requirement. And further, since several ratios are available, several “nominal” loading conditions may be accommodated. For example, an actuator with a range of loading conditions, as is almost the case for robot actuators because of varying arm configurations, is no longer limited to one gear ratio. The compromises in fixed-ratio design are obviously reduced as the number of available gear ratios increases. This is also true if the transmission is never shifted during motion; each motor and reducer pair are literally another available system. An example would be optimizing gear sizes with respect to several “nominal” loading conditions. Ranges of loading and speed requirements could correspond to high payloads, various arm configurations for several “stations” within the overall workspace, travel between two tasks with low end-effector loads, etc. Single, optimized gear selections can be chosen for each of these situations and coupled to the same electric motor; this in effect (as an absolute minimum) turns one actuator into several “fixed-reduction” actuators (FRA’s).

The system is made even more robust when the transmission ratio is allowed to change during joint motions. This effectively expands both the torque and speed characteristics of the actuator. When a loaded joint is accelerated from rest, a significant fraction of the available actuator torque is needed to overcome load inertia. However, when the joint is set into motion, inertial effects diminish and less torque is needed. Ideally, the actuator power can be manipulated such that torque capacity is exchanged for speed capacity when less torque is needed. This is exactly what a transmission does.

From Figure 2.1 [6] it can be seen that the incorporation of a multi-speed transmission expands the capacity of the actuator in terms of both speed and torque simply because of the design modularity; a higher number of motor/transmission combinations along with very strategic ways using them can lead to greatly improved actuators in terms of both speed and torque.

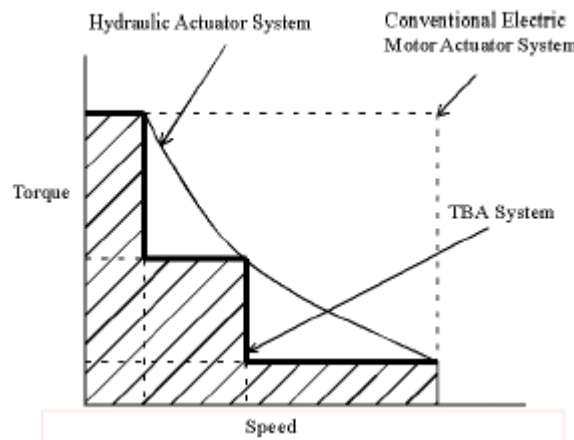


Figure 2.1 Torque-Speed Characteristics of TBA vs. Fixed Ratio Motor

Historically for electric actuators, high torque requirement has demanded that large motors be used with relatively small, and fixed-ratio, gear reductions. The TBA concept suggests that these systems can be replaced with smaller motor-transmission combinations. The desired result is an overall increase in the actuator power density.

## **2.2 Overview of Gear Trains**

The multi-speed transmission is the most crucial component to the concept of TBA. So choosing the appropriate gear train forms the first basic step towards the conceptual design. A *gear train or train of toothed wheels* is the mesh of two or more gears with each other to transmit power from one shaft to another. The nature of the train used depends upon the velocity ratio required and the relative position of the axes of the shafts. A gear train may consist of spur, bevel or spiral gears. The following are the different types of gear trains, depending upon the arrangement of the wheels:

1. Simple gear train
2. Compound gear train
3. Reverted gear train
4. Epicyclic gear train



### 2.3 Epicyclic or Planetary Gear Train

An epicyclic or planetary gear set (Figures 2.2 and 2.3) consists of one or more gears, which are attached to an arm, which can itself rotate about the main axis of the train. This is a two-dof device. Two inputs (one usually being a zero velocity, i.e., a fixed gear) are needed to obtain a predictable output. The simplest type of epicyclic train consists of a sun gear (S), a concentric annulus gear or ring gear (A) and planet gears, which mesh with both S and A.

The following are some of the advantages of epicyclic gear trains over conventional gear trains:

- They are compact
- High torques can be transmitted
- Reversible
- Variable gear ratios can be achieved by controlling the motion of individual gears.
- Simultaneous, concentric, bi-directional outputs are available from a single unidirectional input.

These advantages make epicyclic gears suitable for applications like the TBA where high torques are required to be transmitted while maintaining the compactness of the system.

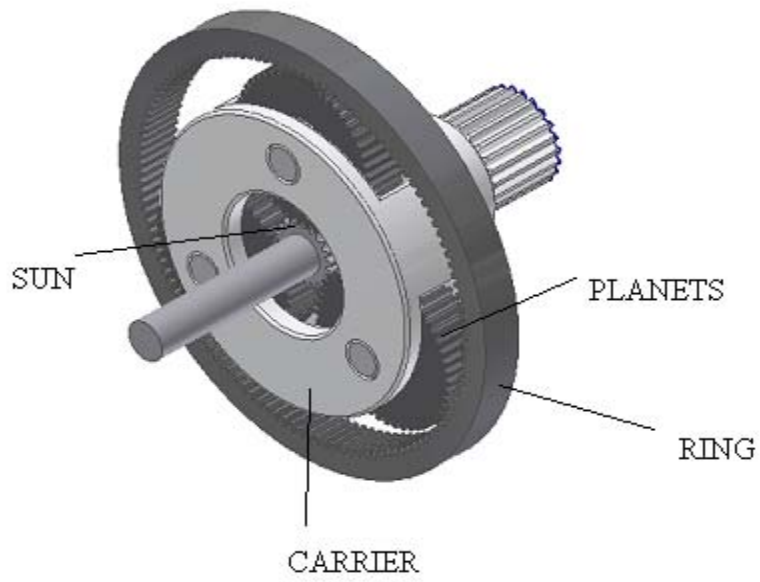


Figure 2.2 Cutout View of Single Stage Planetary Gear

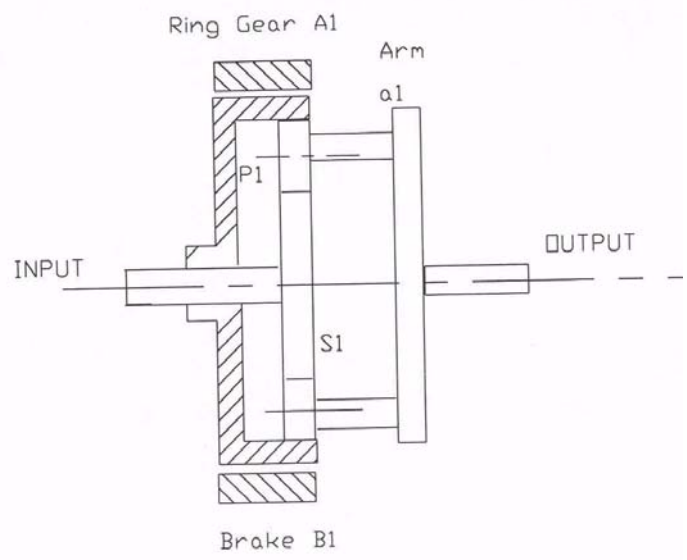


Figure 2.3 Schematic of Single Stage Planetary Gear Set

## 2.4 Design Configuration\*

TBA consists of a D.C. Brushless motor, 3-speed planetary gear transmission and a fixed reduction. The 3-speed planetary gear transmission is illustrated in Figure 2.4. Referring to Figure 2.4, the three-speed gearbox consists of 3 sets of epicyclic gears. The input from the electric motor is given to the sun gears  $S_1$ ,  $S_2$  and  $S_3$ . The output of the system is at the arm  $a_1$  to which are attached planetary gears  $P_1$ . These can rotate freely about their axis, as well as inside the ring gear  $A_1$ . Another set of planetary gears,  $P_2$ , is attached to the arm  $a_2$ . These rotate inside

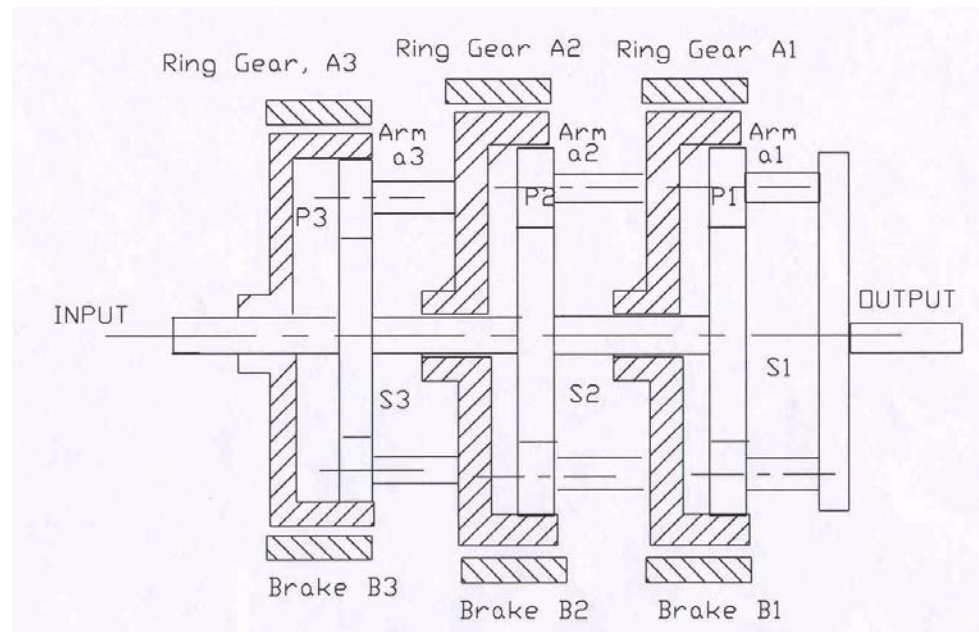


Figure 2.4 3-Speed DVT

---

\* Arun Srikanthiah was also involved in developing this conceptual design for TBA

the annular ring gear  $A_2$ . The arm  $a_2$  is in-turn made integral with the annular ring gear  $A_1$ . Similarly, another set of planetary gears,  $P_3$ , is attached to the arm  $a_3$ . These rotate inside the Annular ring gear  $A_3$  and the arm  $a_3$  is integral with annular ring gear  $A_2$ . Three different speeds are obtained at the Arm  $a_3$  by selectively locking annular ring gears  $A_1$ ,  $A_2$  and  $A_3$ , using brake bands  $B_1$ ,  $B_2$  and  $B_3$  [8]. The detailed analysis of the transmission will be addressed in Chapter 3.

## ANALYSIS OF TRANSMISSION BASED SERVO-ACTUATORS

### 3.1 Kinematics of DVT

The basic kinematic equation for planetary gear train analysis is [8]:

$$\omega_c = \frac{N_s \omega_s + N_r \omega_r}{N_s + N_r} \quad (3.1)$$

where  $\omega_s$  = speed of the sun

$\omega_c$  = speed of the carrier

$\omega_r$  = speed of the ring

$N_s$  = number of teeth on the sun

$N_r$  = number of teeth on the ring

The number of gear teeth is related to the gear diameter by equation 3.2

$$N = \frac{D}{m} \quad (3.2)$$

where  $N$  = number of teeth on a gear,  $D$  = diameter of the gear and  $m$  = module

Basically a planetary gear set is a two-degree of freedom system i.e., it should be provided with two inputs to obtain a determined output. In the present design one input is always the common sun gear shaft and the other input is any one of the three ring gears. In the present configuration, the DVT comprises of three planetary gear sets attached to each other. The carrier of the 1<sup>st</sup> gear set is attached to the ring of the 2<sup>nd</sup> gear set and the carrier of the 2<sup>nd</sup> gear set is attached to the ring of the 3<sup>rd</sup>. The carrier of the 3<sup>rd</sup> is connected to the output shaft. Also braking the ring gear such that the ring gear speed is zero provides the second input to the planetary gear set. In this way one gear ratio is obtained. The next gear ratio is obtained by braking the next planetary gear set while releasing the first one.

The speeds of all the sun gears are equal i.e.,

$$\omega_{s1} = \omega_{s2} = \omega_{s3} = \omega_s \quad (3.3)$$

**Gear #1:** The analysis assumes that the DVT is in 1<sup>st</sup> gear i.e., the 1<sup>st</sup> ring gear is braked

$$\omega_{r1} = 0$$

$$\text{Hence } \omega_{c1} = \frac{N_{s1} \omega_s}{(N_{s1} + N_{r1})} \quad (3.4)$$

But the carrier of the planetary gear set #1 is attached to the ring of the planetary gear set #2

$$\text{Hence} \quad \omega_{r2} = \omega_{c1} \quad (3.5)$$

Substituting (3.5) in (3.1)

$$\omega_{c2} = \frac{N_{s2} \omega_{s2} + N_{r2} \omega_{r2}}{N_{s2} + N_{r2}}$$

$$\omega_{c2} = \frac{N_{s2} \omega_s + \frac{N_{r2} N_{s1} \omega_s}{(N_{s1} + N_{r1})}}{N_{s2} + N_{r2}}$$

$$\omega_{c2} = \frac{\left[ N_{s2} (N_{s1} + N_{r1}) + N_{s1} N_{r2} \right] \omega_s}{(N_{s1} + N_{r1}) (N_{s2} + N_{r2})} \quad (3.6)$$

Again the carrier of #2 is connected to the ring of #3.

Hence  $\omega_{r3} = \omega_{c2}$  (3.7)

Substituting (3.7) in (3.1)

$$\omega_{c3} = \frac{N_{s3} \omega_{s3} + N_{r3} \omega_{r3}}{N_{s3} + N_{r3}}$$

$$= \frac{N_{s3} \omega_s + \frac{N_{r3} \left[ N_{s2} \binom{N_{s1} + N_{r1}}{N_{s1} + N_{r1}} + N_{s1} N_{r2} \right] \omega_s}{\binom{N_{s1} + N_{r1}}{N_{s1} + N_{r1}} \binom{N_{s2} + N_{r2}}{N_{s2} + N_{r2}}}}{N_{s3} + N_{r3}}$$

$$\omega_{c3} = \frac{\left\{ N_{s3} \binom{N_{s1} + N_{r1}}{N_{s1} + N_{r1}} \binom{N_{s2} + N_{r2}}{N_{s2} + N_{r2}} + N_{r3} \left[ N_{s2} \binom{N_{s1} + N_{r1}}{N_{s1} + N_{r1}} + N_{s1} N_{r2} \right] \right\} \omega_s}{\binom{N_{s1} + N_{r1}}{N_{s1} + N_{r1}} \binom{N_{s2} + N_{r2}}{N_{s2} + N_{r2}} \binom{N_{s3} + N_{r3}}{N_{s3} + N_{r3}}} \quad (3.8)$$



**Gear #2:**

$$\omega_{r2} = 0$$

$$\text{Hence } \omega_{c2} = \frac{N_{s2} \omega_s}{(N_{s2} + N_{r2})} \quad (3.9)$$

But the carrier of the planetary gear set #2 is attached to the ring of the planetary gear set #3

$$\text{Hence } \omega_{r3} = \omega_{c2} \quad (3.10)$$

Substituting (3.10) in (3.1)

$$\begin{aligned} \omega_{c3} &= \frac{N_{s3} \omega_{s3} + N_{r3} \omega_{r3}}{N_{s3} + N_{r3}} \\ \omega_{c3} &= \frac{N_{s3} \omega_s + N_{r3} \left( \frac{N_{s2} \omega_s}{(N_{s2} + N_{r2})} \right)}{N_{s3} + N_{r3}} \\ \therefore \omega_{c3} &= \frac{\left[ N_{s3} (N_{s2} + N_{r2}) + N_{s2} N_{r3} \right] \omega_s}{(N_{s2} + N_{r2})(N_{s3} + N_{r3})} \quad (3.11) \end{aligned}$$

Also the ring of #2 is attached to the carrier of #1

$$\omega_{r2} = \omega_{c1} = 0 \quad (3.12)$$

Substituting (3.12) in (3.1)

$$\omega_{c1} = \frac{N_{s1} \omega_{s1} + N_{r1} \omega_{r1}}{N_{s1} + N_{r1}} = 0$$

$$\Rightarrow \omega_{r1} = \frac{-N_{s1} \omega_s}{N_{r1}} \quad (3.13)$$

**Gear #3:**

$$\omega_{r3} = 0$$

$$\text{Hence} \quad \omega_{c3} = \frac{N_{s3} \omega_s}{(N_{s3} + N_{r3})} \quad (3.14)$$

But the carrier of the planetary gear set #2 is attached to the ring of the planetary gear set #3

$$\text{Hence } \omega_{r3} = \omega_{c2} = 0 \quad (3.15)$$

Substituting (3.15) in (3.1)

$$\omega_{c2} = \frac{N_{s2} \omega_{s2} + N_{r2} \omega_{r2}}{N_{s2} + N_{r2}} = 0$$

$$\Rightarrow \omega_{r2} = \frac{-N_{s2} \omega_s}{N_{r2}} \quad (3.16)$$

But the carrier of the planetary gear set #1 is attached to the ring of the planetary gear set #2

$$\omega_{r2} = \omega_{c1} \quad (3.17)$$

Substituting (3.17) in (3.1)

$$\omega_{c1} = \frac{N_{s1} \omega_s + N_{r1} \omega_{r1}}{N_{s1} + N_{r1}} = \frac{-N_{s2} \omega_s}{N_{r2}}$$

$$\therefore \omega_{r1} = \frac{-\left[ N_{s2} (N_{s1} + N_{r1}) + N_{s1} N_{r2} \right] \omega_s}{N_{r1} N_{r2}} \quad (3.18)$$

The final governing equation for the output speed is:

$$\omega_{c3} = \delta_1 \frac{\left\{ \begin{array}{l} N_{s3} (N_{s1} + N_{r1})(N_{s2} + N_{r2}) + \\ N_{r3} [N_{s2} (N_{s1} + N_{r1}) + N_{s1} N_{r2}] \end{array} \right\} \omega_s}{(N_{s1} + N_{r1})(N_{s2} + N_{r2})(N_{s3} + N_{r3})} + \delta_2 \frac{[N_{s3} (N_{s2} + N_{r2}) + N_{s2} N_{r3}] \omega_s}{(N_{s2} + N_{r2})(N_{s3} + N_{r3})} + \delta_3 \frac{N_{s3} \omega_s}{(N_{s3} + N_{r3})} \quad (3.19)$$

The delta functions  $\delta_1, \delta_2, \delta_3$  give the state of the band brake. If  $\delta_1=1$ , then band brake1 is engaged and when  $\delta_1=0$ , band brake1 is disengaged.

Correspondingly,  $\delta_2$  and  $\delta_3$  indicate whether band brakes 2 and 3 are engaged or disengaged respectively.

**Planet Speeds:** In all the three gears the respective planet speeds are given by the following equations:

$$\omega_{p1} = \frac{\omega_{c1} (N_{s1} + N_{p1}) - \omega_s N_{s1}}{N_{p1}} \quad (3.20)$$

$$\omega_{p2} = \frac{\omega_{c2} (N_{s2} + N_{p2}) - \omega_s N_{s2}}{N_{p2}} \quad (3.21)$$

$$\omega_{p3} = \frac{\omega_{c3} (N_{s3} + N_{p3}) - \omega_s N_{s3}}{N_{p3}} \quad (3.22)$$

## **DESIGN OF GEAR SHIFTING MECHANISM**

### **4.1 Introduction**

The gear shifting mechanism is very important to the functionality of TBA. It primarily consists of a band brake, a series of mechanical linkages and electrical actuator. It is used to stop the ring gear, rotating at a very high speed and hold it to obtain the desired gear ratio. The mechanism should have the following attributes:

- i. Should be able to provide a minimum of 25 N-m\* (number obtained from Dynamic Simulation of the entire system using MathWorks Simulink) braking torque.
- ii. Should be modular in regards to size and weight requirements. The design should be such that the entire TBA envelope should be as small as possible.
- iii. Should be able to withstand operating speeds up to 3000 rpm.

---

\* Dr. Seewong Kim, Research Associate at MABE department, UT Knoxville obtained this result

## 4.2 Band Brakes

The most fundamental component of the gear shifting mechanism is the band brake. It is used to stop and hold the ring gear of a planetary gear train. Also called transmission bands, they serve an important role in many transmission designs, since they provide unusual packaging geometries, and handle much higher loads than that of friction clutches. “Bands are found in at least four formats: single-wrap rigid bands, double-wrap rigid bands, flexible bands, and a hybrid cross double-wrap rigid band and flexible” [9].

The major advantage of the band brake is the relatively large frictional surface area it provides while consuming very little space in the transmission. Along with this advantage, there is a drawback associated with the band brake. The basic disadvantage of the band brake is the degree of the control that can be maintained over the engagement when it is used to accomplish a “shift”. The control of the shift is complicated by the geometry of the band-ring system. As a result, an exponential relationship exists between band actuation force and the pressure exerted by the band lining on the ring. This problem is compounded by the fact that the band can be applied in either an “energized” or “de-energized” mode, depending on the direction of the ring rotation. But these problems can be reduced by using electronic control.

Usually a band brake has two modes viz., energized mode and de-energized mode. Energized mode is defined as the mode in which the direction of ring

rotation coincides with direction of the band actuator motion. Application of force causes the band to be tightened around the ring. A direct result of the wrapping motion is that, in energized mode, the ring drag force adds to the band actuation force. De-energized mode is defined as the mode in which the direction of ring rotation is opposite to the direction of band actuator motion. In this mode, the two forces i.e., ring drag force and band actuation force are subtracted.

A band is used to absorb the reaction force of a planetary gear train. In modern planetary-gear automatic transmissions, bands always transmit reaction torque to ground through the case. Band applications may take the form of a static or dynamic engagement. A static condition exists when there is little or no rotation of the ring prior to band application. A dynamic condition exists when there is drum rotation due motor torque prior to the band application.

In the usual design situation involving the use of a band, the factors that must be evaluated are as follows [9]:

- i. Magnitude and direction of torque input and band actuator force.
- ii. Magnitude of the loading at the reaction end of the band.
- iii. Magnitude of the maximum unit pressure to which lining material will be subjected.

The forces required for a particular application depend on the following:



- i. Reaction torque: determined by motor torque and planetary gear train ratio(s).
- ii. Inertia of rotating masses.
- iii. Effective radius of ring.
- iv. Coefficient of friction of materials.
- v. Angle of band wrap.

The angle of wrap of a band is that angle of band contact with the ring defined by the lines drawn through the center of ring rotating to the apply loads (Figure 4.1) [9]. The angle of wrap will be less than 360 degrees for a single-wrap band and greater than 360 degrees for a multiple-wrap band. Band ends should be designed to give a true tangential pull from the point of band lining contact with the ring. A design where the band ends are longer than the tangential actuation force point on the drum does not fully utilize the possible band capacity available, because of the reduction in maximum possible angle of wrap. A design where the band ends are shorter than the tangential actuation force point on the ring should be avoided to prevent excess wear to the band ends.

### **4.3 Actuation Force Requirements**

The following section is reproduced from reference [10] for the sake of convenience. A general arrangement of band brakes is shown in Figure 4.2.

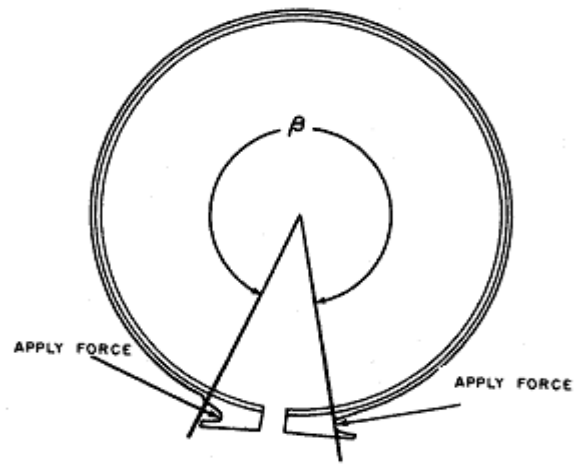


Figure 4.1 Band Wrap Angle

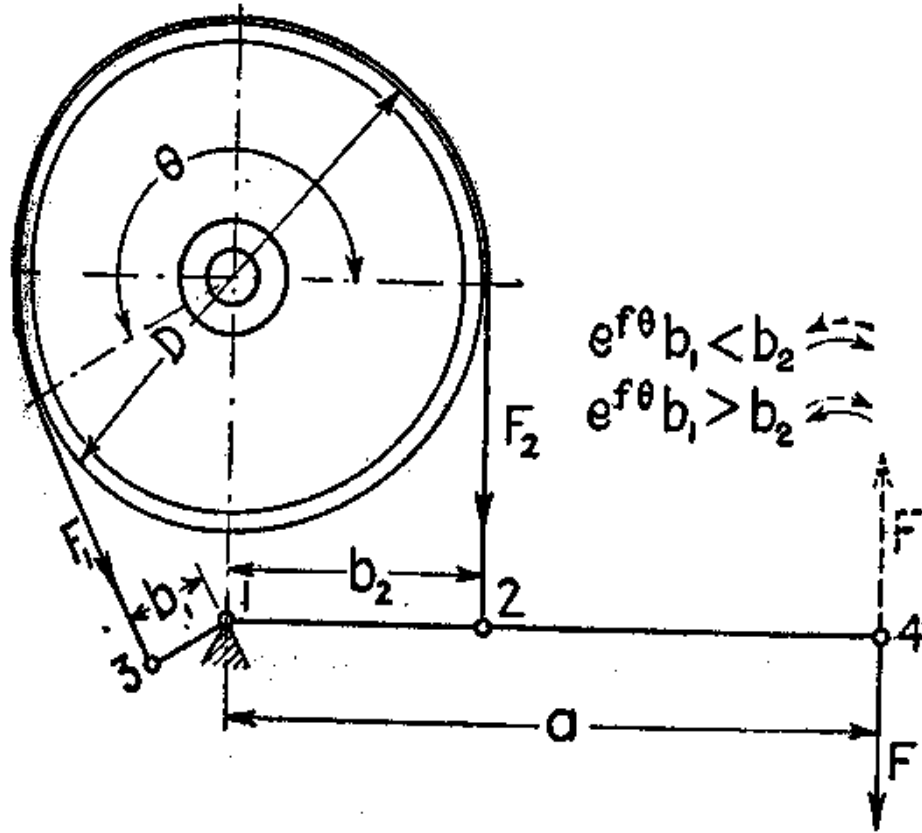


Figure 4.2 Differential Brake

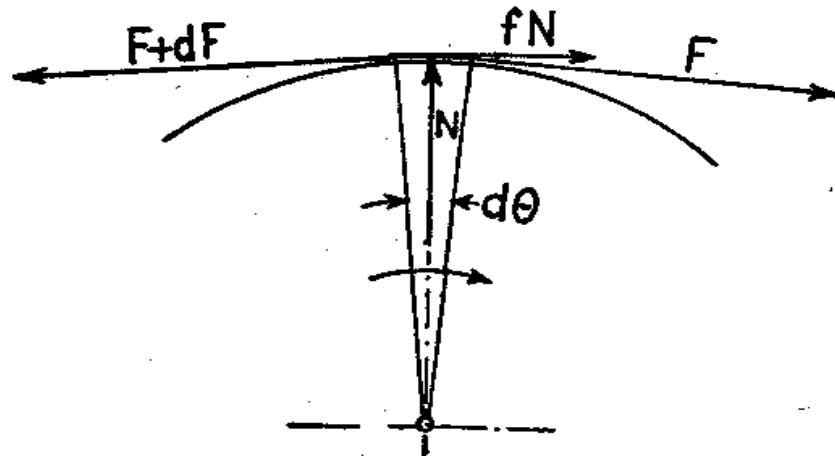


Figure 4.3 Band Tension

avoided to prevent excess wear to the band ends.

#### 4.3 Actuation Force Requirements

The following section is reproduced from reference [10] for the sake of convenience. A general arrangement of band brakes is shown in Figure 4.2. Before a relation is deduced between the tangential force  $F_t$  and the pull  $F$  at the lever, it should be noted that owing to friction between the sheave and the band the tensions  $F_1$  and  $F_2$  in the two band ends are different. Such brakes are called *differential brakes* [10].

*Differential brakes:* The desired relation for differential brakes may be found using Figure 4.3. An infinitesimal length of the band is subtended by the angle  $d\theta$ . The

tension at one end is  $F$ , and at the other end is  $(F+dF)$ ; each of these tensions makes an angle of  $(\frac{1}{2} \pi - \frac{1}{2} d\theta)$  with the vertical centerline. The pressure between the band and the ring is designated by  $N$ , and with a friction coefficient of  $f$  the friction force is  $fN$ .

This piece of band is held in equilibrium by the four forces  $F$ ,  $(F+dF)$ ,  $N$  and  $fN$ .

The summation of the horizontal and vertical components, gives the following equations:

$$-dF \cos\left(\frac{1}{2} d\theta\right) + fN = 0 \quad (4.1)$$

$$\text{and } -\left(2F + dF\right) \sin\left(\frac{1}{2} d\theta\right) + N = 0 \quad (4.2)$$

Eliminating  $N$ , and substituting  $\frac{1}{2} d\theta$  for  $\sin(\frac{1}{2} d\theta)$  and 1 for  $\cos(\frac{1}{2} d\theta)$ , gives,

$$fF d\theta - dF = 0 \quad (4.3)$$

Separating the variables and integrating gives,

$$\int_{F_2}^{F_1} \frac{dF}{F} = f \int_0^{\theta} d\theta$$

$$\Rightarrow \frac{F_1}{F_2} = e^{f\theta} \quad (4.4)$$

The net tension is evidently equal to the tangential braking torque, or

$$F_1 - F_2 = F_t \quad (4.5)$$

From equations (4.4) and (4.5)

$$F_1 = \frac{F_t e^{f\theta}}{e^{f\theta} - 1} \quad (4.6)$$

$$\text{and } F_2 = \frac{F_t}{e^{f\theta} - 1} \quad (4.7)$$

The equations (4.6) and (4.7) show that for a required  $F_t$  the magnitudes of  $F_1$  and  $F_2$  decrease with an increase of the friction coefficient  $f$ , and particularly with an increase of the angle of contact  $\theta$ .

Considering the operating lever as a free body and taking moments about the fulcrum  $l$ , an assuming clockwise rotation in Fig: 4.2, there results

$$F_1 a + F_2 b_1 = F_t b_2 \quad (4.8)$$

Substituting the values for  $F_1$  and  $F_2$  from equations (4.6) and (4.7), and solving for  $F$ , gives

$$F = \frac{F_t \left( \frac{b_2 - e^{f\theta} b_1}{e^{f\theta} - 1} \right)}{\left( \frac{f\theta}{e^{f\theta} - 1} \right) a} \quad (4.9)$$

$$\text{But } F_t = T / R \quad (4.10)$$

where  $T$  is the required braking torque and

R is effective ring radius.

$$F = \frac{T \begin{pmatrix} f\theta \\ b_2 - e & b_1 \end{pmatrix}}{R a \begin{pmatrix} f\theta \\ e & -1 \end{pmatrix}} \quad (4.11)$$

The conditions represented in Fig.4.2 require that

$$b_2 > e \frac{f\theta}{b_1} \text{ or } \frac{b_2}{b_1} > e \frac{f\theta}{b_1} \quad (4.12)$$

If  $\frac{b_2}{b_1} = e \frac{f\theta}{b_1}$ , then F is zero and the brake becomes self-locking. This is

undesirable and even dangerous because of the possible fluctuation of the friction coefficient  $f$ .

If  $\frac{b_2}{b_1} > e \frac{f\theta}{b_1}$ , the pull F becomes negative, the brake is applied automatically,

and a pull must be applied in the opposite direction, as shown by dotted lines in fig.4.2, in order to allow the ring to turn and thus to lower the load.

If the direction of rotation is reversed, or is counterclockwise, the greater tension F1 will act as the right end of the band, and the smaller tension F2 will act at the left end. A similar analysis gives

$$F = \frac{T \left( e^{f\theta} b_2 - b_1 \right)}{R a \left( e^{f\theta} - 1 \right)} \quad (4.13)$$

If  $b_2$  does not differ much from  $b_1$ , the influence of  $e^{f\theta}$  according equations 4.11 and 4.13, is small. The main factor determining the magnitude of  $F$  for a given  $T$  or  $F_t$  is the average ration of the lever arms, or the ratio  $(b_1 + b_2)/2a$ .

*Simple band brake:* A simple band brake is a special case of the differential brake, in which one of the band ends is fastened to the fixed fulcrum. One of these arrangements is shown in Figure 4.4 [10]. The expression for  $F$  is obtained from equation 4.11 or equation 4.13, the proper depending on the direction of rotation of the sheave. In the present case,  $b_1 = 0$  and  $b_2 = b$ . Thus, for clockwise rotation, equation 4.11 gives

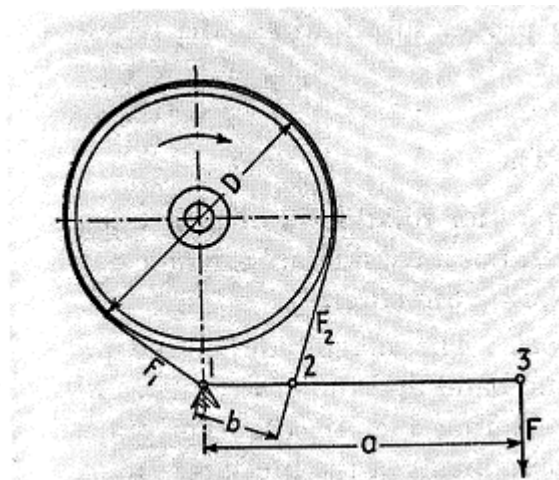


Figure 4.4 Simple Band Brake



$$F = \frac{T b}{R a \left( e^{f\theta} - 1 \right)} \quad (4.14)$$

Equation 4.14 shows a certain advantage of a simple band brake over a differential brake. By making the angle of contact  $\theta$  large, the necessary pull  $F$  can be made small without increasing unduly the lever arm  $a$ .

For counterclockwise rotation, equation 4.13 gives

$$F = \frac{T b e^{f\theta}}{R a \left( e^{f\theta} - 1 \right)} \quad (4.15)$$

The direction of reaction torque and band apply forces determines if a band is self-energizing or not. A band is self-energizing if the direction of the forces is such that the gear train reaction torque input to the ring is acting in the same direction as the torque moment about the center of the ring due to apply force. This means that the summation of moments about the center of rotation will give a band anchor force always greater than the apply load. Conversely, if the band is not self-energizing, the apply force will be greater than the band anchor force. Typically for single-wrap bands in current use, the anchor force is approximately double the apply force when operating in a self-energizing direction, and approximately half the apply force when operating in the de-energized direction. In order not to increase  $F$  unnecessarily, the rotation should be clockwise. This means that the band end with higher tension must be fastened to the fixed

fulcrum. This self-energizing feature, although a definite capacity in most of the transmission designs is a big disadvantage in the present scenario because the band is also required to have torque capacity in the non-energizing direction. Hence the need for a band brake mechanism that can have high torque capacity in both the directions to stop the ring gear.

*Band brakes for rotation in both directions:* For brakes in which the rotation is reversible, it is desirable to have the same pull  $F$ , regardless of the direction of the band. A study of the Fig.4.2 shows that this can be accomplished if both the moment of the tension  $F_1$  and the moment of tension  $F_2$  act in the same direction and in the opposite direction to the moment of the pull,  $F$ . To obtain this result the overhanging lever end  $3-1$  must be turned to another position and the lever arms  $b_1$  and  $b_2$  must be equal. A brake with this arrangement is shown in Figure 4.5 [10]. By applying the same method of analysis, or by using either equation 4.11 or equation 4.13, with proper substitution, the required magnitude of the pull is determined to be

$$F = \frac{T b \left( e^{f\theta} + 1 \right)}{R a \left( e^{f\theta} - 1 \right)} \quad (4.16)$$

With this arrangement the required pull for rotation in one direction is heavier than the required by other brakes, but a considerably lighter pull is sufficient when rotation occurs in the opposite direction.

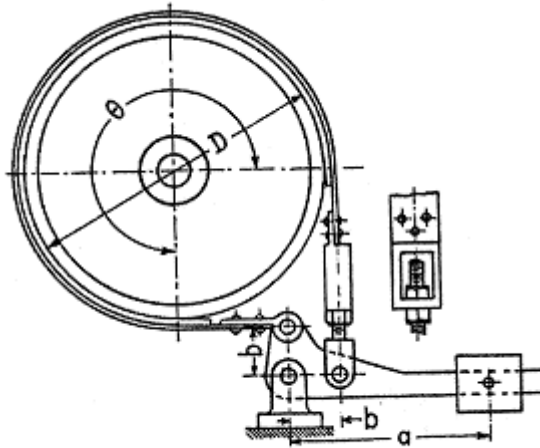


Figure 4.5 Band Brake for Rotation in Both Directions

For the present design using eq 4.16, the following is the force actuator requirement:

$F$  - Actuator force =?

$T$  - Torque applied on the ring = 25 N-m (obtained from Dynamic Simulation)

$f$  - coefficient of friction = 0.15

$\theta$  - angle of wrap =  $5\pi/3 = 300^\circ$

$R$  - radius of the ring = 2.75in = 0.06985 m

$b/a$  - lever arm = 1/1

**$F = 957.785 \text{ N} = 215.32 \text{ lb-f}$**  (1 pound-force = 4.448222 N)

*Pressure on band:* The magnitude of the pressure  $p$  between the band and the ring may be found considering Figure 4.6 [10]. The sum of the horizontal components of the pressures is equal to the product  $pD$ . Thus, the sum of the forces applied to the band ends, or  $2F$ , is equal to  $pDw$ , where  $w$  is the band width. For a rotating ring,  $2F = F_1 + F_2$  and the average pressure is

$$p = \frac{F_1 + F_2}{Dw} \quad (4.17)$$

Because of stretching of the band, the pressure around the sheave is not constant.

It is highest at the tight end of the band, where it is

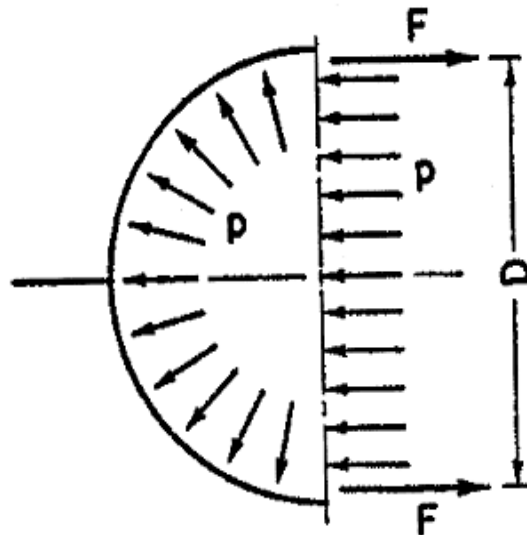


Figure 4.6 Forces Acting on a Band Brake

$$p = \frac{F_1}{2Dw} \quad (4.18)$$

It decreases gradually toward the other end to the minimum value of

$$p = \frac{F_2}{2Dw} \quad (4.19)$$

#### **4.4 Friction Materials for Band Applications**

Friction materials available for band lining ranges from semi-metallic to organic, with static coefficients of friction in the 0.10 to 0.17 range for wet application and  $\mu$  to 0.55 for dry applications for bands and rings in good condition. The values for the dynamic coefficients of friction may be lower.

Semi-metallic materials are capable of withstanding high unit pressure, although there is a tendency to abrade the ring, and conformity is limited because of their hardness. The use of materials in this class generally is limited to applications that require high static torque capacity and a minimum service under dynamic conditions [9].

Organic band lining materials are available with a wide range of physical characteristics. The hardness may vary from that of the semi-metallic materials to relatively soft paper based compositions. With the harder organic materials, it is

possible to ensure uniform contact on the ring by controlling the ring finish so that the band lining is abraded during initial running.

The hardness of a band lining must be compatible with the ring hardness, because many organic materials are available that will wear the ring to the point of surface failure before the lining itself exhibits significant wear. A softer material will have less abrasive effect on a ring and exhibit more consistent shift quality because of better compliance to the ring. Possible disadvantages of the softer lining material are greater wear and more compressive set. The latter factor is especially important if a relatively thick, soft material is used with a high unit loading. Current practice shows that unit loadings in the 400 psi (anchor end) range are practical. A band of this type absorbs heat energy at a maximum rate of about 0.9 Btu/s for each square inch of contact area. Since unit loading varies, heat energy is not distributed equally over the band length, however ring rotation tends to minimize this effect [9].

In the present design, a woven non-asbestos band lining material manufactured by Midwest Brake® is used. The carcass consists of brass wire inserted yarns containing organic and inorganic fibres including Kevlar®. The woven type is vacuum impregnated for thorough saturation with an oleoresinous compound. Long, slow cooling with alternate rolling and compressing for a final high temperature step cure bonds the material into an extremely dense and strong product with excellent friction and wear characteristics.

The coefficient of friction in dry condition is given as 0.40 for dynamic and 0.42 for static.

#### 4.5 Design of Band Brake

The external diameter of the ring gear is 5.5 inches. Hence the internal diameter of the band brake  $D=5.55$  inches.

A practical rule is to determine the band thickness  $h$  by the relation

$$h = 0.005 D \quad (4.20)$$

$$\therefore h = 0.005 \times 5.55 = 0.02775 \text{ inches}$$

After  $h$  has been selected, the width  $w$  is determined from considerations of strength by applying the equation

$$wh = \frac{F_1}{S_d} \quad (4.21)$$

where  $S_d$  is the design stress in psi. In selection  $S_d$ , a safety factor  $n = 2 \times 2 = 4$  should be used because of the possible sudden applications of a band brake. For the present application tensile strength = 60000 psi. Hence design stress is  $60000/4=15000$  psi.

But from equations 4.6 and 4.10

$$F_1 = \frac{T(e^{f\theta})}{R(e^{f\theta} - 1)} = \frac{25 \times e^{0.15 \times 5 \times \pi \div 3}}{0.06985(e^{0.15 \times 5 \times \pi \div 3} - 1)} = 658 \text{ N} = 147.92 \text{ lb-f}$$

Hence  $w = 0.35$  inch

But from manufacturing view point the width of the band is determined as 0.75 inches. The brake-band ends are fastened to the lugs by welding. Figure 4.7 shows the designed band brake.



Figure 4.7 Band Brake



#### **4.6 Band Construction and Processing**

Bands generally are fabricated by rolling a partial hoop from steel strip stock and attaching suitable band ends by one of several methods available. In the case of bands that carry a relatively light load, the band ends may be stampings attached to the band loop either by spot welding or riveting. Where strength requirements are more stringent, a composite forging, incorporating both band ends is butt-welded to the band loop [11].

“The fabrication of a quality single-wrap band requires careful processing, including a number of interesting steps. The steps involved in manufacturing a typical band, made up from a composite band end forging and a formed steel hoop, may be described as follows:

1. Blank and form the band hoop section from steel strip stock. The length of the hoop must allow for burnoff during welding and be shorter than the hoop section of the finished band, to allow for stretching the welded assembly to size.
2. Coin composite band end forging to conform to nominal drum radius plus lining thickness, and trim to length for welding. Apply and reaction pockets maybe finish machined if lining is to be bored to size.
3. Weld composite forging to band hoop and trim flash.

4. Heat-treat the assembly to relieve welding stresses.
5. Expand the band assembly to a nominal diameter of ring or drum plus lining thickness.
6. Shot blast the inside of the band assembly. This operation prepares the surface for bonding of lining material and also builds up a compressive stress that will cause the band to spring open when the band ends are separated.
7. Bond the lining in place and bore to size, if necessary.
8. Finish machine the apply and reaction pockets, if not done before welding. Finish band to width if required.
9. Separate composite forging to form a finished band” [11].

#### **4.7 Band Failure Modes**

Band failures can be divided into four categories, as follows [11]:

1. A failure may result from excessive wear. Wear at the anchor end of a self-energizing band, and wear at the apply end of a band that operates in the de-energized direction, can be anticipated because of the highest unit loading being at the ends of the bands. Excessive wear, however, may be the result of the geometry of the struts imposing forces at the band ends,

or a lining material incompatible with the drum (ring). In some cases, the linkage geometry is intentionally designed with toe-in effect to improve the conformity of the band with the drum (ring). In this type of an installation, the wear problem can be alleviated by using segments of a harder lining material near the ends of the bands.

2. A failure because of glazing an organic band lining and drum (ring) surface probably results from a soft drum (ring) with a relatively smooth initial finish. The band burnishes the ring, and vice versa, continually lowering the coefficient of friction. Before this state is attained, the engagement becomes exceedingly smooth and stretched out. Improper fluids also have been known to induce this type of failure.
3. The abuse and overheating of a band may cause a bonding failure such that the lining is freed from the steel band. This failure usually occurs with the thin, semi-metallic linings, while the insulating effect of thick organic linings prevents the temperature at the bond line from becoming critical.
4. Mechanical failure of apply tab, anchor(s), fasteners or welds are rare, but do occur. They are typically the result of metal fatigue or severe shock loading.

#### 4.8 Survey of Different Actuators

Most systems involve motion or action of some sort. This motion is created by a force or torque that results in acceleration and displacement. *Actuators* are devices used to produce this motion or action. Actuators produce physical changes as linear or angular displacement. They also modulate the rate and power associated with these changes. Usually the following actuators are used to actuate and release a band brake:

- i. *Hydraulic*: It is the most common form of band actuator mechanism used. Hydraulic systems are designed to move large loads by controlling a high-pressure fluid in distribution lines and pistons with mechanical or electromechanical valves. A typical hydraulic servo consists of a cylinder, a piston, and usually a return spring. Force from the servo usually acts directly on the end of the band. These types of actuators are capable of generating very high force but they are very complex in nature and require regular maintenance.
- ii. *Mechanical*: This is the simplest, least costly way to engage a clutch or brake. Actuation is by rods, cables, levers, or cams. Besides cost, a major advantage of this type of actuation is the “feel” of engagement that the operator gets through a pedal or lever, and the ability to judge the amount of braking force or slip needed. Because mechanical actuation depends on human strength, actuation force is limited to about 75 lb. This low clamping force limits response times and cycling rates.

- iii. *Pneumatic:* Principles of operation of pneumatically operated brakes are similar to hydraulically operated units, but they use compressed air as the working medium rather than hydraulic fluid.
- iv. *Electromagnetic:* Electromagnetic brakes permit extremely fast cycling rates (to 1,600 cpm in some uses). However, they do not provide the torque range of pneumatic or hydraulic units. They are more convenient for much automatic machinery, where control commands are electric. An electromagnetic brake has two basic parts: an annular electromagnet and an armature. The electromagnet is a wound copper coil embedded in a donut-shaped iron shell, and fitted with lining material to retard wear. When voltage is applied to the coil, the magnetic field formed by the electromagnet shell and armature engages the two friction surfaces and locks them together as a rotating unit. If the armature is allowed to rotate, and the electromagnet is held stationary, the unit works as a brake.
- vi. *Solenoids:* Solenoid consists of a coil and a movable iron core called the armature. When the coil is energized with current, the core moves to increase the flux linkage by closing the air gap between the cores. The movable core is usually spring-loaded to allow the core to retract when the current is switched off. The force generated is approximately proportional to the square of the current and inversely proportional to the square of the length of the air gap. Solenoids are inexpensive, and their use is primarily limited to on-off applications. *Electric Motors:*

“Electric motors are by far the most ubiquitous of the actuators, occurring in virtually all electromechanical systems. Electric motors can be classified either by function or by electrical configuration. In the functional classification, motors are given names suggesting how the motor is to be used. Examples of functional classifications include torque, gear, servo, instrument servo and stepping” [12]. Examples of electrical configurations are ac motor, dc motor and stepper motor.

*viii. Piezoelectric:* A piezoelectric actuator (PZT) can produce extremely fine position changes down to the sub-nanometer range. The smallest changes in operating voltage are converted into smooth movements. Motion is not influenced by stiction/friction or threshold voltages. PZTs can generate a force of several 10,000N. Piezo actuators offer the fastest response time available.

#### **4.9 Methods of Actuation and Release**

*Design 1:Lever Mechanism with Solenoid Actuation:* The design is a simple lever mechanism similar to that in Figure 4.5. The lever mechanism is “Y- shaped”. The small ends are attached to the ends of band brake and the longer end is joined to an actuator, which is a solenoid in the present scenario. The conceptual representation of such a design is presented in Figure 4.8. The band ends are attached to end of links E, and  $F_i$  is the actuator force. The force amplification is given by the ratio of the links D/E.

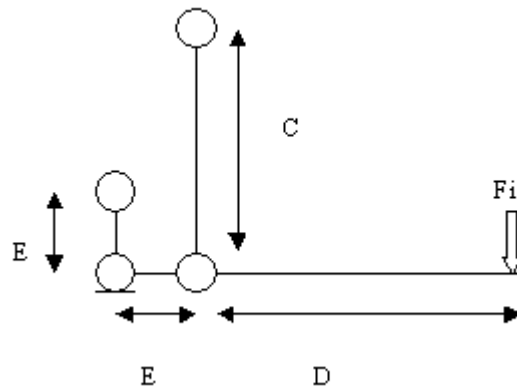


Figure 4.8 Conceptual Representation of a Simple Lever Mechanism

For the present design using eq 4.16, the following is the force actuator requirement is:  $F = 957.785 \text{ N} = 215.32 \text{ lb-f}$  (1 pound-force = 4.448222 N).

But the solenoid actuator considered exerts 25 lbf at 0.1 inch stroke i.e.,  $F_i = 25$  lbf. Hence the effective lever arm required is  $F/F_i = 215/25 \sim 9$ . So  $D/E = 9:1$ .

For design considerations if  $E = 0.5$  inch, then  $D = 4.5$  inches. So this design adds 4.5 inches to the envelope of DVT at minimum. But as seen earlier, design modularity is one of the key features of DVT. Hence this design is not suitable for DVT.

*Design 2:Slider-Crank Mechanism with Electrical motor Actuation\**: As seen in the earlier section, the design modularity requirements of TBA warrant a completely different design and this section discusses one such design. The conceptual design is illustrated in Figure 4.9. Basically it consists of two links attached to the ends of band brake and a third link running through a bearing is attached to the junction of the other two links. Using Fig.4.9, the actuator force is deduced as

$$F_a = \frac{\tau \cdot b \left( e^{f\theta} + 1 \right)}{R a \left( e^{f\theta} - 1 \right)} \sin \left( \frac{\theta}{2} \right) - \mu \frac{\tau}{R} \cos \left( \frac{\theta}{2} \right) \quad (4.22)$$

where  $F_a$  - Actuator force =?

T - Torque applied on the ring = 25 N-m (obtained from Dynamic Simulation)

$f$ - coefficient of friction of the band brake= 0.15

$\theta$  - angle of wrap =  $5\pi/3 = 300^\circ$

R - radius of the ring = 2.75in = 0.06985 m

b/a - lever arm=1/1

$\mu$ - friction due to the bearing or bushing

For different values of  $\mu$ , the actuator force is calculated w.r.t covering angle  $\theta$

---

\* The initial conceptual design and mathematical formulation is the work of Renbin Zhou. The detailed design also involves the works of Sriram Sridharan and Kelley Brown



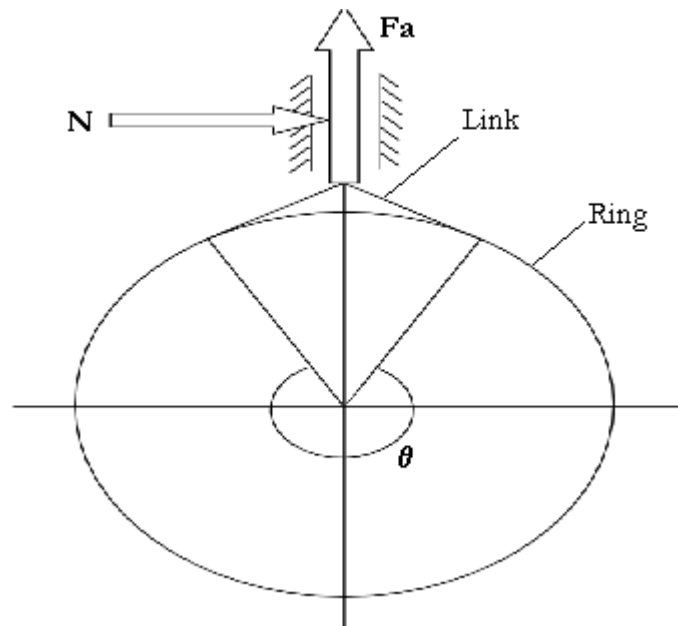


Figure 4.9 Conceptual Design of Gear Shifting Mechanism

and the plot is illustrated in Figure 4.10. The stated conceptual design is now discussed in detailed as shown in Figure 4.11. It consists of a band brake running around the ring gear; two links attached to the ends of the band brake; the other end of the links is attached to the two symmetric sides of a 3-way yoke piece; the 3<sup>rd</sup> side of yoke piece is attached to another link; the other side of this link is attached to one side of the crank; the other side of the crank is connected to the motor shaft. An L-bracket is used for the mounting of the brake motor and as an interface with the TBA housing. The yoke is run through a bushing, which is fitted into the L-bracket for the reduction of the actuator force as discussed earlier in this section.

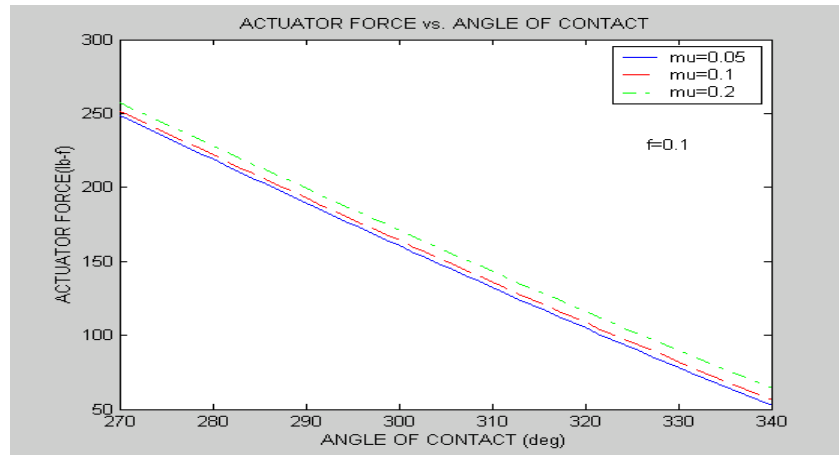


Figure 4.10 Plot of Actuator Force w.r.t. Angle of Contact

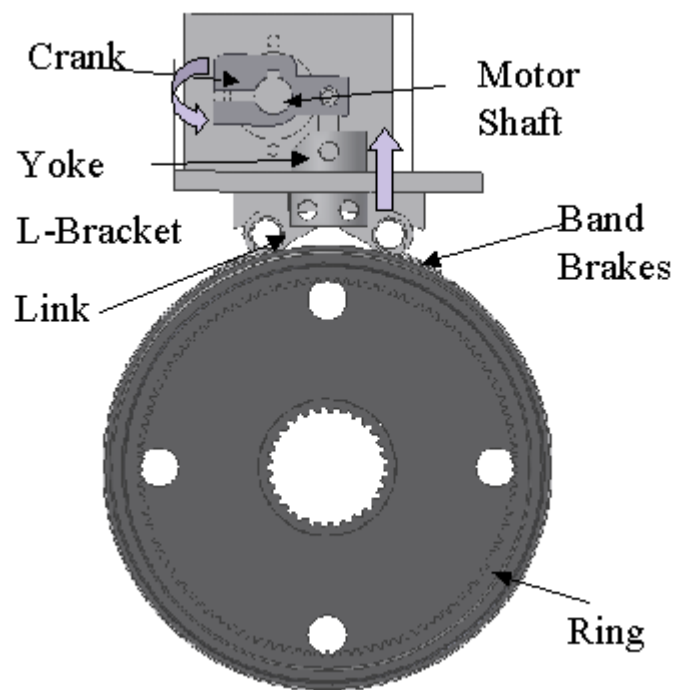


Figure 4.11 Detailed Design of Gear Shifting Mechanism

The brake motor is connected to the amplifier, which in turn is connected to the computer with brake control software (National Instruments LabView is the control software used). When a current control signal is given in the software, the motor rotates at the desired rpm and at the desired torque. Then the motor turns the crank by some angle, which in turn pulls the link-yoke upwards. This pulls the other two links towards each other, which causes the band brake to tighten around the ring gear and hold the ring gear in its place. Figure 4.12 gives the exploded view of the brake mechanism assembly.

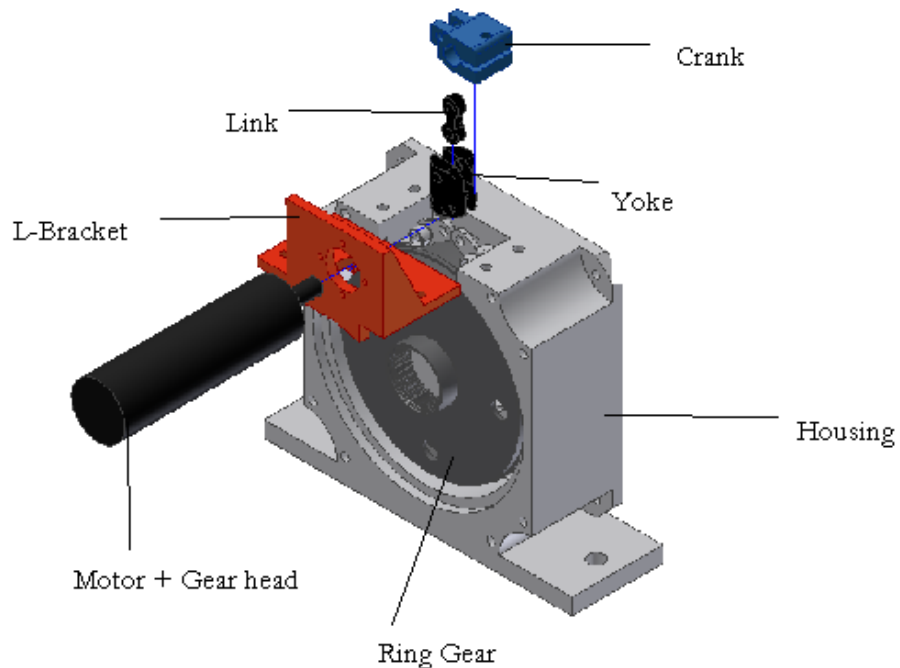


Figure 4.12 Exploded View of the Band Brake Assembly

## PROTOTYPE EXPERIMENTAL EVALUATION

### 5.1 Brake Test Stand

Ultimately, one cannot be sure of the correctness or viability of any design until it is built and tested. This usually involves the construction of a *prototype* physical model. A mathematical model, while being very useful, can never be as complete and as accurate a representation of the actual physical system as a physical model, because of the need to make simplifying assumptions.

*Testing* of the model or prototype may range from simply actuating it and observing its function to attaching expensive instrumentation to accurately measure displacements, velocities, accelerations, forces, temperatures, and other parameters. Tests may need to be done under controlled environmental conditions such as high or low temperature or humidity [13]. A *test stand* is the piece of equipment designed and built to test the functionality of the prototype. Figure 5.1 illustrates such a test stand designed to test the functionality of the gear shifting mechanism.

Prototype Testing consists of six stages starting with the test rig design, rig fabrication, installation and checkout, parametric testing and the comparison of the collected data with computer predictions.

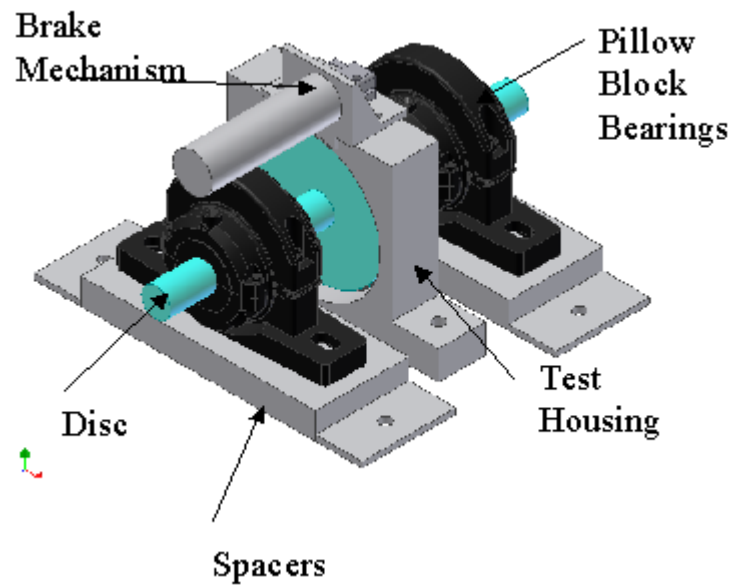


Figure 5.1 Brake Band Static Test Stand

*Test Stand Design:* The test stand basically consists of the following equipment:

- i. Band brake mechanism i.e., band brake, yoke, links, l-bracket, bushing, shaft extender and support, brake motor and amplifier.
- ii. Computer with brake control software (LabView is used in the present experiment).
- iii. Disc with embedded shafts on both sides.
- iv. Test housing
- v. Mounting parts like pillow block bearings and spacers.
- vi. Torque wrench.
- vii. Transmission Fluid

*Test Stand Description:* In this testing equipment, a disc with embedded shafts on both sides is used to mimic the ring gear. The two shafts are mounted on the pillow block bearings. The band brake is mounted on the disc in the same way as done in the real scenario. The rest of the mechanism is mounted on to the test housing as discussed in detail in section 4.6- *Design 2*. The height of the pillow block bearings is adjusted using spacers to obtain the same ring center height from the ground as in the original TBA design. The brake motor, which is mounted, on the L-bracket is connected to the amplifier, which in turn is connected to the computer with brake control software. Figures 5.2 and 5.3 illustrate some pictures of the designed test stand.

*Test Procedure\**: The testing is done in two modes viz. dry mode and the wet mode i.e., in which the band brake is lubricated with transmission fluid and tested.

- i. The initial testing was carried out when the band brake is dry i.e., when it is not lubricated.



Figure 5.2 Static Test Stand

---

\* This section and the following section is the combined work of the author and Renbin Zhou

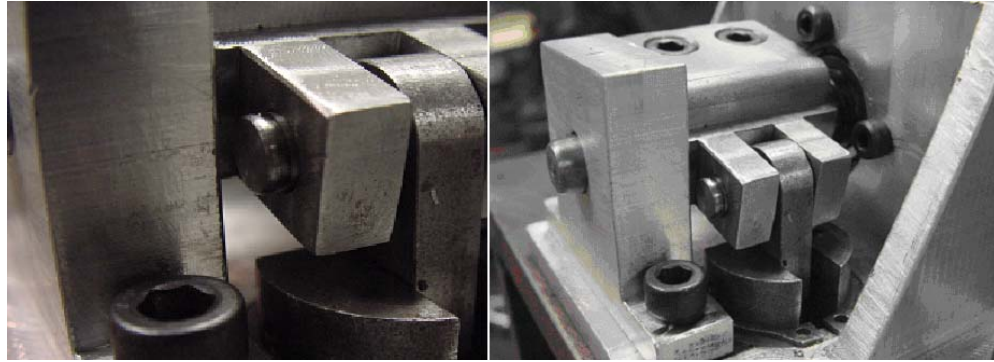


Figure 5.3 Close-Up Views of the Crank, Yoke and the Other Linkages

- ii. A very low current control signal corresponding to a brake motor torque of 1.5 N-m was given in the software. This makes the motor rotate at the desired rpm and engages the band brake.
- iii. Now one end of the shaft is turned and held with a torque wrench and both static and dynamic braking torque were measured (As illustrated in Figure 5.4).
- iv. The disc is rotated and three torque measurements at three different points were taken for same value of motor current.
- v. The disc was rotated in both the directions so that the effects of disc direction, disc finish, inertia, lubrication, and unit loading can be evaluated.
- vi. The motor current value was incremented such that brake motor torque is incremented by 1.5 N-m and steps (ii) to (v) were repeated and



Figure 5.4 Band Brake Test Procedure

the corresponding braking torques measured. The same procedure was followed until the maximum motor current corresponding to the maximum brake motor torque of 15 N-m was achieved.

- vii. Then the band brake was lubricated with transmission fluid and steps (ii) to (vi) are repeated..
- viii. The following additional parameters were measured when possible:
  - a. Actuation time
  - b. Temperature rise of both motor and band brake
  - c. Visual inspection of the mechanical parts

## 5.2 Experimental Results

The obtained results are listed in Tables 5.1 to 5.4 and corresponding plots in Figures 5.5 and 5.6. From Table 5.1 it can be noted that:

- 1) Consistent data was hard to get due to equipment and human error.



Table 5.1 Brake test datasheet 1  
(08/14/2003)

Run	Surface Condition	Motor Torque (Nm)	Static Torque (Nm)		Braking Torque mean (Nm)		Dynamic Torque (Nm)	
			CW	CCW	CW	CCW	CW	CCW
1	D (Dry)	1.5(0.3A)	8	15	7.67	13	5	5
			8	11				
			7	13				
2	D	3	15	24	13.67	26	8	12
			13	26				
			13	28				
3	D	4.5	21	38	22.67	39.3	10	15
			25	40				
			22	40				
4	D	6	31	41	30.83	41	20	22
			30.5	41				
			31	41				
5	D	7.5	31	41	30.67	41	20	22
			31	41				
			30	41				
6	D	9	34	44	33.83	44	24	25
			33.5	44				
			34	44				
7	D	10.5	49	60	49.17	60.33	33	34
			49.5	61				
			49	60				
8	D	12	63	75	62.67	75.3	45	45
			63	76				
			62	75				
9	D	13.5	72		72.33			
			72					
			73					
10	D	15(3A)	68		68.33			
			68					
			69					

Table 5.2 Brake test datasheet 2  
(08/15/2003)

Run	Surface Condition	Motor Torque (Nm)	Static Torque (Nm)		Braking Torque mean (Nm)		Dynamic Torque (Nm)	
			CW	CCW	CW	CCW	CW	CCW
1	W (Wet)	1.5(0.3A)	3	3	3	3.33	2	2
			3	3				
			3	4				
2	W	3	10	12	10	12	10	11
			10	12				
			10	12				
3	W	4.5	12	13	12.67	13.67	11	13
			13	14				
			13	14				
4	W	6	13	14	13.67	15	12	13
			13	16				
			14	15				
5	W	7.5	21	21	21.33	21.67	20	20
			22	22				
			21	22				
6	W	9	21	22	21.33	21.67	20	20
			22	21				
			21	22				
7	W	10.5(2.1A)	31	32	32.33	32	(25~35)	25~35
			32	32				
			31	32				
8	W	12	43	42	42.67	42.33	35~50	35~50
			43	43				
			42	42				
9	W	13.5	52	52	52.33	52.33	40~60	40~60
			53	52				
			52	53				
10	W	15(3A)	45	45	45	45	35~50	35~50
			45	45				
			45	45				

Table 5.3 Brake test datasheet 3  
(08/28/2003)

Run	Surface Condition	Motor Torque (Nm)	Static Torque (Nm)		Braking Torque mean (Nm)		Dynamic Torque (Nm)	
			CW	CCW	CW	CCW	CW	CCW
1	W (Wet)	1.5(0.3A)	4	3	3.67	3.67	3	3
			4	4				
			3	4				
2	W	3	9	9	9.33	8.67	9	9
			9	8				
			10	9				
3	W	4.5	14	15	14	14.33	13	13
			15	14				
			13	14				
4	W	6	16	17	17	16.67	16	17
			17	16				
			18	17				
5	W	7.5	20	21	20.33	20.67	21	20
			20	20				
			21	21				
6	W	9	28	27	27	27	26	27
			26	28				
			27	26				
7	W	10.5(2.1A)	41	42	42	41.33	35~50*	35~50*
			43	41				
			42	41				
8	W	12	47	48	47.33	47.67	43~55*	43~55*
			47	47				
			48	48				
9	W	13.5	52	52	52.33	53.33	45~60*	45~60*
			53	52				
			52	53				
10	W	15(3A)	47	48	47.33	47	42~55*	42~55*
			48	47				
			47	46				

Table 5.4 Brake test datasheet 4

(08/28/2003)

Run	Surface Condition	Motor Torque (Nm)	Static Torque (Nm)		Braking Torque mean (Nm)		Dynamic Torque (Nm)	
			CW	CCW	CW	CCW	CW	CCW
1	W (Wet)	1.5(0.3A)	4	3	3.33	3.33	3	3
			3	3				
			3	4				
2	W	3	10	9	9.67	9.67	10	10
			10	10				
			9	10				
3	W	4.5	12	13	12.67	12.33	12	12
			13	12				
			13	12				
4	W	6	23	24	22.33	23.67	23	23
			22	23				
			22	24				
5	W	7.5	25	25	24	24.33	24	25
			24	24				
			23	24				
6	W	9	32	33	32.33	33	34	33
			33	33				
			32	33				
7	W	10.5(2.1A)	43	42	42.67	42.67	35~50*	35~50*
			43	43				
			42	43				
8	W	12	53	54	53	53.67	43~55*	43~55*
			52	54				
			54	53				
9	W	13.5	58	57	58.67	57.67	45~60*	45~60*
			58	58				
			59	58				
10	W	15(3A)	54	55	55	55	42~55*	42~55*
			55	55				
			56	55				

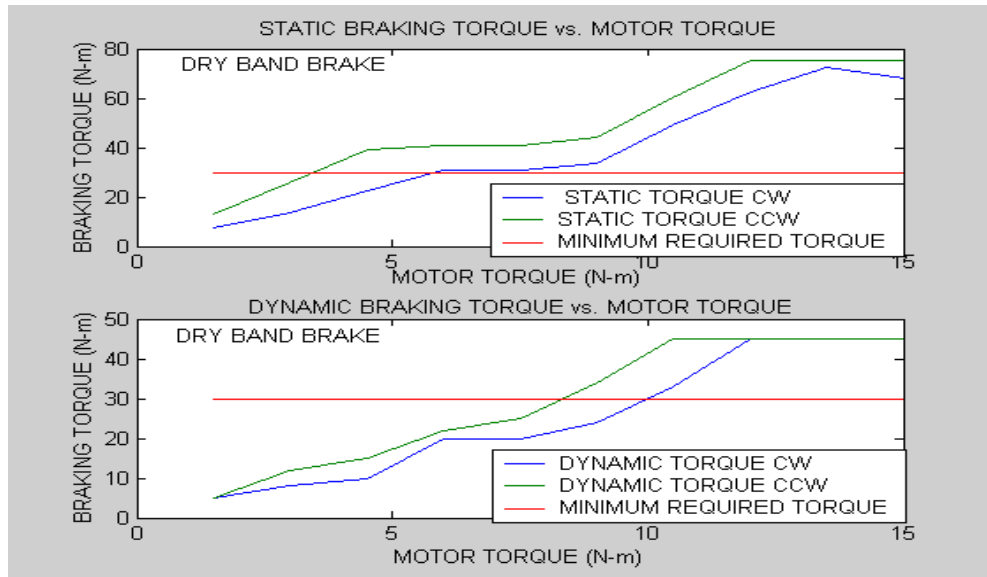


Figure 5.5 Plots for Experimental Results Obtained for Dry Band Brake

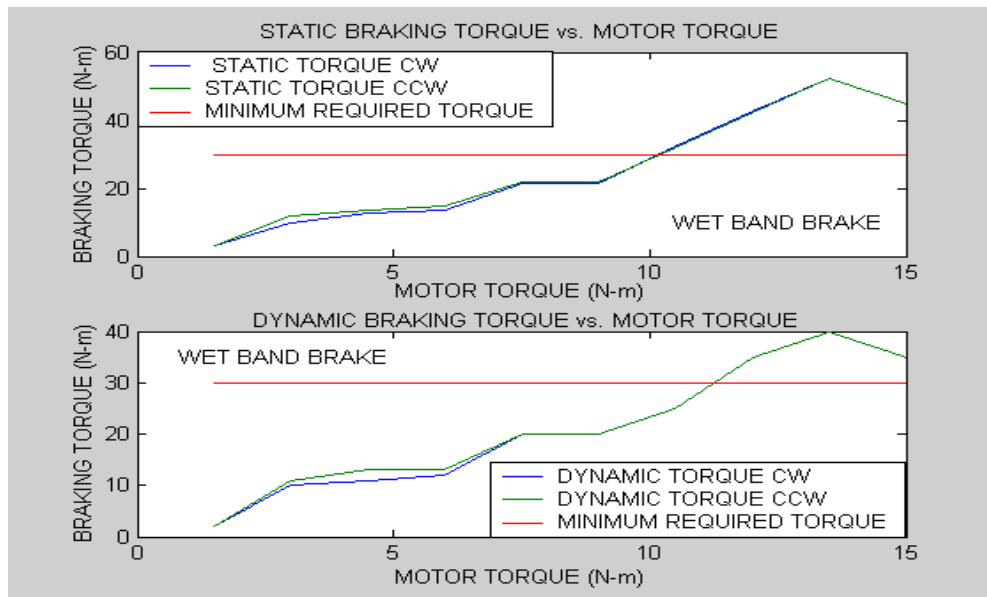


Figure 5.6 Plots for Experimental Results Obtained for Wet Band Brake

- 2) Torque meter (0-70Nm, 1Nm accuracy), Torque wrench (0~150 ft-lbs, 1 ft-lbs accuracy)
- 3) Data missing in Run 9 and 10 due to hardware setup and equipment limit.
- 4) Temperature rises ~40deg in 5 minutes @3Amp

From Table 5.2 it can be noted that at high current ( $>2.1\text{Amp}$ ), the dynamic torque is oscillating due to equipment and human controllability, but the trend shows that in the wet case, the static torque should be similar to the dynamic torque.

### **5.3 Conclusions**

The fundamental design requirement of the GSM is to generate at the least 25 N-m to 30 N-m of braking torque in all conditions with very low response time. From the experimental results it can be seen that the mechanism produces a braking torque of 45 N-m per 12 N-m motor torque in the dry condition and a minimum of 35 N-m braking torque per 12 N-m motor torque in the wet condition. At high current ( $>2.1\text{Amp}$ ) corresponding to high motor torque, the dynamic braking torque is oscillating due to equipment and human controllability. The temperature rise in the system is not significant. From visual inspection of the system it can be noted that all the joints of the mechanism are rigid.

## **SUMMARY AND FUTURE WORK**

### **6.1 Summary**

The fundamental idea of TBA is to incorporate a multi-speed transmission to spread a servomotor's torque speed characteristics across a wider output speed range. TBA uses multi-speed transmissions such that heavy, high-torque motors can be traded for high-speed, reduced mass motor-transmission combinations. TBA design consists of a D.C. Brushless motor, a Discrete Speed Variable Transmission (DVT) and a fixed reduction. Of these, fundamental to the proof of principle of TBA is the design of the DVT.

Initially the conceptual design of the DVT was developed. Then the kinematic and dynamic analysis was performed in order to determine the rigid-body dynamic behavior of the system. The DVT is a three-speed gearbox consisting of 3 sets of epicyclic gears and every planetary gear set is a 2-degree of freedom system i.e., it should be provided with two inputs to obtain a determined output. Hence the present design is 6 degrees of freedom system, which reduces to 2 degrees of freedom system when all the constraint equations are applied. One input is always the common sun gear shaft and the other is the ring gear. The motor gives one input to the planetary gear and the second input to the planetary

gear set is obtained by locking the band brakes such that the ring gear speed is zero. Three different speeds are obtained by selectively and synchronously locking the three annular ring gears using corresponding external flexible band brakes and the gear shifting mechanism (GSM).

The GSM primarily consists of a band brake, a series of mechanical linkages and electrical actuator. The most fundamental component of the gear shifting mechanism, the band brake, was designed, the actuator force requirements were established and a slider-crank mechanism was designed for application and release of the band brakes. Also the 3- models of all the parts were developed using AutoDesk Inventor and the corresponding blue prints generated for manufacturing.

Then a test stand was designed and built to test the functionality of the GSM. The effective braking torque was measured for both dry and wet conditions. The obtained test results were analyzed with respect to the predicted simulated results.

The kinematic analysis of the DVT, the design of the band brake, the estimation of the actuator force requirements, the abandoned lever mechanism with solenoid actuation design for the application and release of band brake, the survey of the different actuator mechanisms, the design of the static test stand and the development of 3-d models and corresponding blue prints using AutoDesk Inventor form the exclusive domain of the author's research. Other



parts of the thesis are the result of teamwork for which the author is grateful to the concerned individuals.

## **6.2 Future Work**

The present Gear Shifting Mechanism design is quite complicated for the function it serves in the TBA. Hence the immediate research topic in this area is the development of less complicated GSM design for the DVT. The simpler design can be the one with the same fundamental principle but with fewer components or an entirely different design. Also the present design occupies a significant amount of space, which in turn increases the net envelope of the DVT. So further research need to carried out to decrease the design envelope of the GSM. This can be accomplished by researching the electric motor industry and gear manufacturing industry for high power density motors with very small size and high precision gearheads respectively.

Furthermore, future research may explore different design alternatives involving electromagnetic clutches or friction clutches instead of band brakes to reduce the complexity of the GSM design. This line of research may result in a 2- planetary gear set DVT able to deliver 3 or more speeds instead of the present 3- speed, 3- planetary gear set one, which will optimize the DVT design itself along with the GSM design. Also research can be performed in the field of high friction materials for clutches and band brakes, which in turn reduces the force and torque requirements of the actuator. Future research also includes development

of more realistic dynamic models of the wet band brake, which includes several non-linear phenomena, and estimation of the braking torque from such a model.

## **BIBLIOGRAPHY**

- [1] T.C.Widner. ANALYSIS OF TRANSMISSION-BASED ROBOTIC SERVO ACTUATOR CONCEPT. Ph.D. dissertation, Mechanical Engineering, The University of Tennessee, Knoxville, 1999
- [2] Katushito Ogata. SYSTEM DYNAMICS, Third Edition. Prentice Hall, 1998.p.2
- [3]Survey of D&D Manipulator Requirements and State of the Art EHA Technology Topical Report.
- [4] Richard C. Dorf. MODERN CONTROL SYSTEMS. Fifth Edition. Addison-Wesley Publishing Company.1989.p.36
- [5] W.R.Hamel and T.C.Widner “Transmission-Based Electrical Actuators: A Concept for Improving Robot Manipulator Performance”. In *IEEE International Conference on Robotics and Automation*, pages 2204-2211, Albuquerque, New Mexico, 1997.
- [6] W.R.Hamel, S.Kim, R. Zhou and A.Lumsdaine “ Dynamic Modeling and Analysis of a Transmission-based Robot Servoactuator”. In *IEEE International Conference on Robotics and Automation*, pages 208-213, Taipei, Taiwan, 2003.
- [7] T.C. Widner and W.R.Hamel “ Control Concepts for Transmission Based Actuators in Robotics”. In *IEEE International Conference on Intelligent Robots and Systems*, pages 1170-1175, Victoria, B.C., Canada, 1998
- [8] S. S. Ratan. THEORY OF MACHINES, Seventh Reprint. Tata McGraw Hill, 1998,p 405.

- [9] Robert J. Fanella. "Design of Bands" in DESIGN PRACTICES: Passenger Car Automatic Transmissions, Third Edition. SAE Inc., 1994, pages: 419-426.
- [10] V.L. Maleev and J.B. Hartman. "Band Brakes" in MACHINE DESIGN. Third edition, International Textbook Company, 1954, pages: 371-375.
- [11] Robert J. Fanella. "Manufacturing Considerations of Bands" in DESIGN PRACTICES: Passenger Car Automatic Transmissions, Third Edition. SAE Inc., 1994, pages: 431-433.
- [12] M.B. Hestand and D.G. Alciatore. "Introduction to Mechatronics and Measurement Systems". WCB/ McGraw-Hill, 1999,p 309.
- [13] R.L. Norton. DESIGN OF MACHINERY: An Introduction to the Synthesis and Analysis of Mechanisms and Machines. Second Edition, McGraw Hill, 1999, p13.

## VITA

Kalyana Bhargava Ganti was born in Hyderabad, India, on November 26<sup>th</sup>, 1977. He completed his high school from St. Andrew's High School, Hyderabad in 1993 and his intermediate from Ideal Junior College in 1995. He had a great time at these two places. He made very good friendships and shared some golden moments during this period, which will be always treasured.

Kalyana started his engineering journey in 1998. He enlisted himself into Production Engineering at Deccan College of Engineering and Technology affiliated to Osmania University. He graduated with his Bachelor's Degree in July 2001. Upon receiving his Bachelor's he wanted to pursue his Master's in Mechanical Engineering and so elevated himself to school of higher learning at University of Tennessee, Knoxville. He will be graduating in December 2003. He has plans of pursuing his Ph.D.

Kalyana's hobbies include books and movies. He is an avid reader and ardent movie watcher. He is profoundly influenced by the philosophy of Ayn Rand known as Objectivism.