Tension variation reduction in high speed cone winding

Tension Variation Reduction in High Speed Cone Winding

by

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A Doctoral Thesis

Submitted in partial fulfilment of the requirements

for the award of

Doctor of Philosophy of Loughborough University of Technology

May 1992

ACKNOWLEDGEMENTS

I first of all wish to thank my supervisor Dr. I. C. Wright for his continuous guidence throughout this period of study and for his comments which helped a lot in the writing of this thesis.

Thanks are due to my directors of research Mr. T. H. Davies for his help in many respects and Dr. T. G. King for his help and comments in writing this thesis.

I also wish to think Mr. W. F. Silvester and Dr. C. P. Garner for their encouragement and generous help.

I wish to extend thanks to many members of the Loughborough University of Technology for their help and advice.

I gratefully acknowledge the financial support provided by the British Council, Chinese Education Committee and the Committee of Vice-Chancellor and Principals for supporting this research work.

DECLARATION

This is to certify that I am responsible for the work submitted in this thesis and that the original work is my own except where due reference to previous work has been noted. I also certify that neither the thesis nor the original work contained therein has been submitted to this or any other institution for a higher degree.

Sen Yang

SYNOPSIS

The research work was concerned with the design of tension compensators for spinning-winding machines for the purposes of reducing winding tension variation during high speed cone winding.

Initially, the research was directed to investigating the 'winding error', i.e. the difference between yarn supply and demand, which was the main cause of winding tension variation. Mathematical models were established to analyse the winding velocity variation and yarn path length variation. Several methods have been investigated to reduce winding tension variation and to develop tension compensators. These include using a servo motor to keep constant winding velocity and using a curved distribution bar to keep constant path length. Two promising tension compensators were selected for detailed investigation. they were a mechanical compensator and a mechatronic compensator.

Based on the analysis of 'winding error', A multiple bar mechanism was devised, analysed and optimised to provide tension compensation. Further consideration has been given to a microprocessor controlled mechanism, that works according to predetermined look-up tables and sensor signals to reduce winding tension variation. The computer simulation of yarn winding process, the software design for the optimisation of mechanical compensator and the control of mechatronic compensator, and the results of winding tension experimentation are also presented in the thesis.

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CHAPTER 1 INTRODUCTION

The textile industry is one of the most prosperous industries in the world. Textile production has a long history dating back many thousands of years. Human beings have clothed themselves with woven materials since the dawn of history, and the history of civilisation is also, to some extent, associated with fabrics. There is evidence that Egyptians made woven fabrics over 6000 years ago and it is believed that in prehistoric times lake dwellers in Europe made nets from twisted threads. The fact that filaments extruded by silkworm were used by the ancient Chinese to make the finest of fabrics is also well known to the world. Old mural paintings and carvings, china and other ancient artifacts make it clear that providing himself with clothes was an important facet of man's early life.[1] Until comparatively recent times spinning, winding, and weaving were skills associated with domestic life and there are references to these familiar occupations in the literature of all civilised societies.[2~4] With the development of the textile industry, various textile machineries and new textile technologies are also developing rapidly, they promote the textile industry to a more efficient, economical, and productive stage.

The first stage in the production of a fabric from fibres is to clean and mix them thoroughly. The fibres are generally straightened, but for the production of certain types of fabrics they must be brought into a condition in which they are all parallel, the fibres are then drawn out into the form of a sliver, which resembles a rope, although the fibres have no twist. During spinning, the fibres are twisted and become yarn. The twist is given to provide sufficient strength to prevent breakage in manipulation. The yarn is wound onto a package and is ultimately used to produce fabrics by either knitting or weaving.

1.1 Introduction to Masterspinner

The Masterspinner, which is the concern of this study, is a spinning-winding machine. As is shown in Fig 1.1, the machine is mainly composed of two units, one is a spinning unit, the other is a winding unit. At first, the sliver goes into the spinning unit, from where it comes out in the form of yarn. The yarn then passes through delivery rollers and enters a "tension device". Finally, in the winding unit, the yarn is led by a traverse guide and wound on to a rotating package.

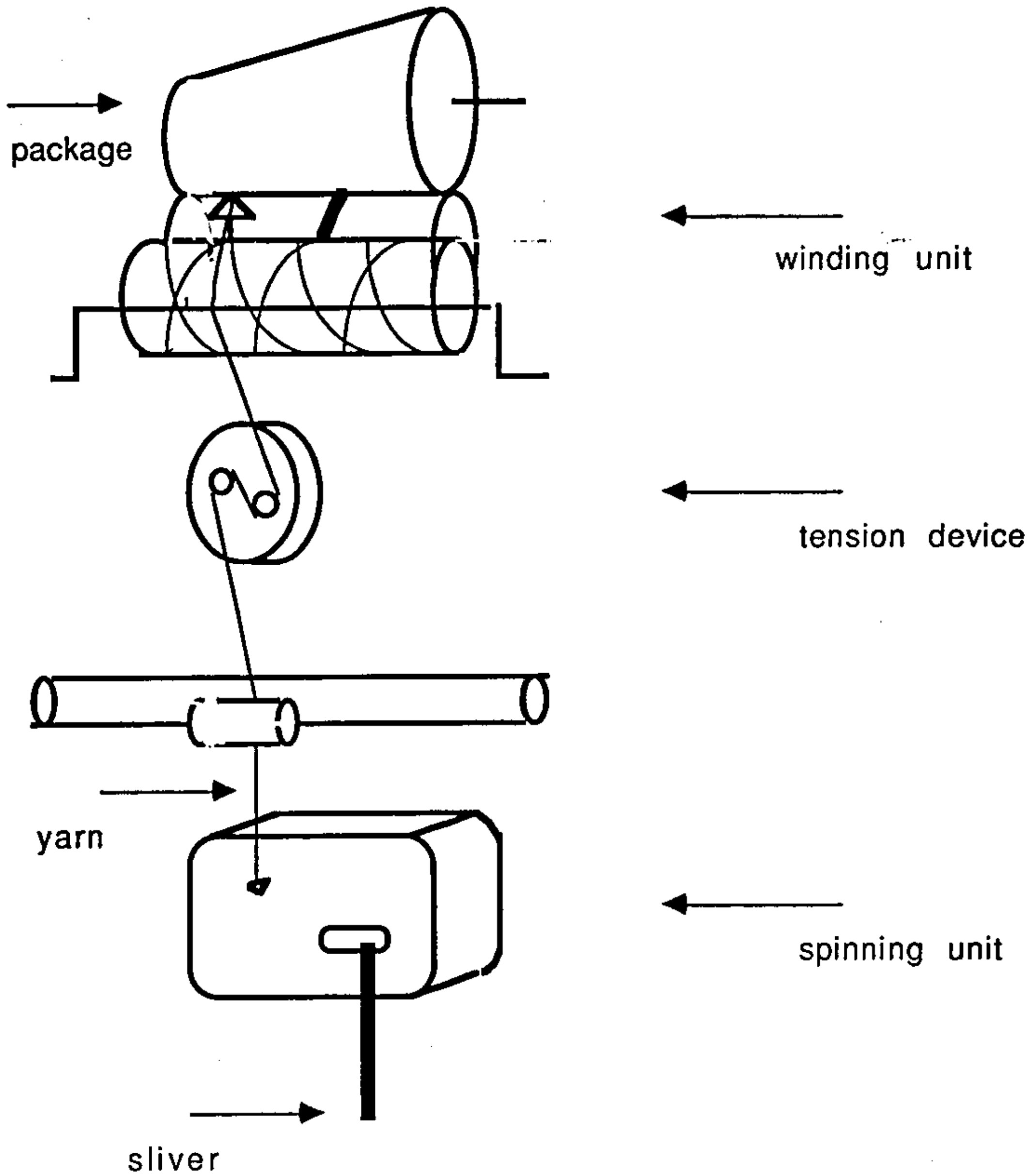


Fig 1.1 Spinning Unit and Winding Unit

1.1.1 The Spinning Unit

The object of spinning is to draft or draw fibres into yarn of a specified fineness and insert sufficient twist to bind the fibres together, the yarn must be evenly spun and strong enough to withstand all further strain imposed on it.

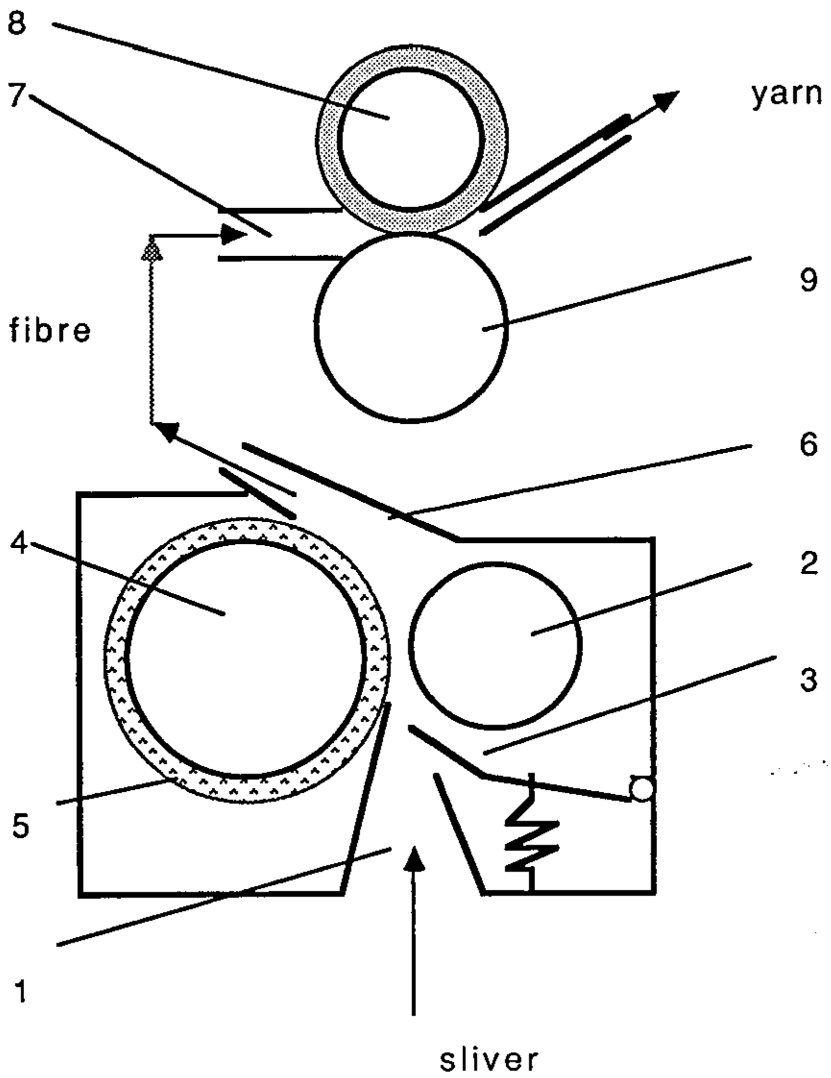


Fig 1.2 Yarn Spinning Device

Fig 1.2 is a drawing of an "open end"(OE) spinning unit, which is composed of a fibre-separating device and a spinning device.

The fibre-separating device consists of the inlet condenser (1), a feeding roller (2) with pressing element (3), a combing roller (beater) (4) covered with wire pins (5), a direct channel (6) for the transport of fibres into the spinning device. The spinning device consists of a transfer tube (7), a perforated roller (8) and a solid roller (9).

The spinning process and the velocities of fibres in a spinning unit of this type are as follows. The raw material is fed into the fibre-separating device in the form of a sliver at velocity v_0, which is imparted by the surface speed of the feeding roller. The fringe coming from the nip between the feeding roller and the pressing element is acted on by the wire pins of the combing roller, which has a surface speed v_1 which is greater than v_0. By this action, the fibres from the fringe are separated and attain a velocity v_1. The zone in which the combing roller interferes with the direct channel is the fibre-doffing area. The separated fibres leave the wire pins by the action of a centrifugal force and by the air stream which has a velocity v' in the direct channel. The velocity v' must be higher than v_1. The fibres in the direct channel are accelerated to a velocity v_2 by the influence of the air stream. The fibres are transported in the air stream towards the spinning rollers. The yarn formation takes place in the nip of the rollers, fibres and yarn being pressed against the surfaces by the suction from the inside of the perforated roller. The finished yarn is pulled off in the direction of the roller axis.[5]

1.1.2 The Winding Unit

When yarn is formed, it is deliveried to the winding unit. Winding was formerly regarded as a process, the primary purpose of which was to transfer yarn from a small capacity primary package to a larger package with greater length. This size of package is more suitable for further processing.

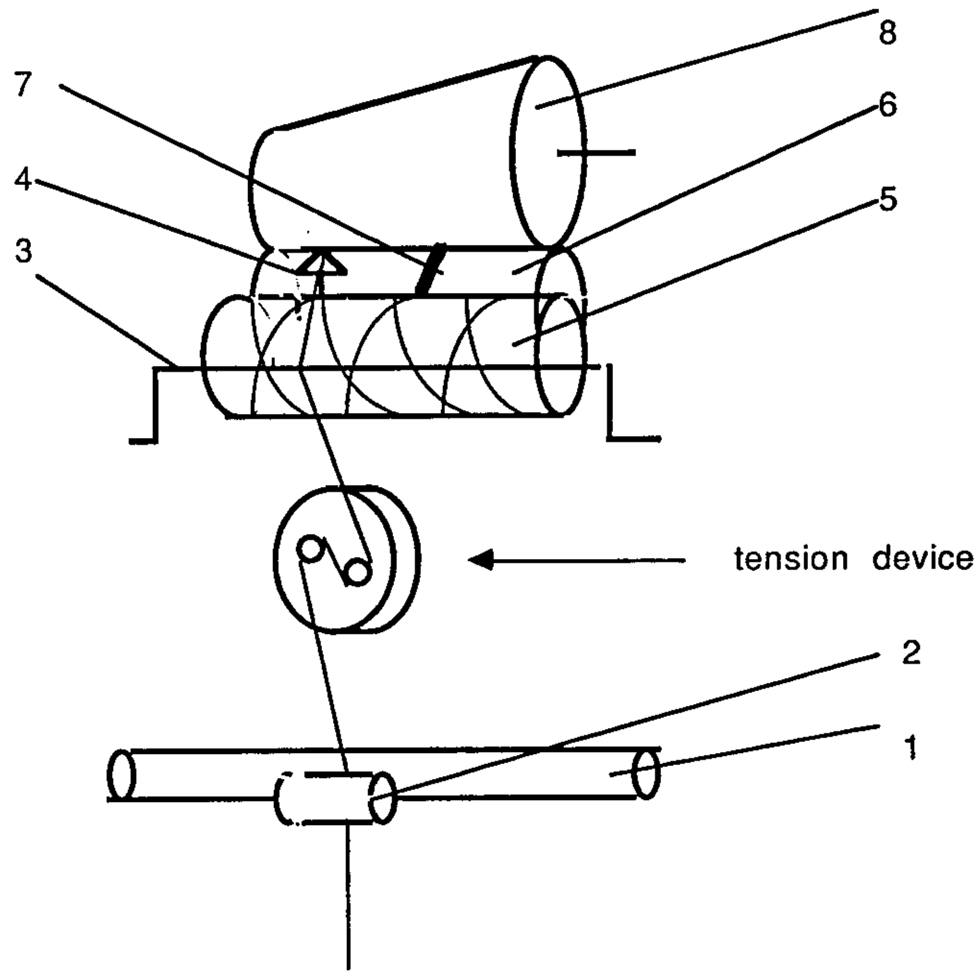


Fig 1.3 Yarn Winding Process

In the spinning-winding machine, after the yarn comes out of the spinning unit, it is deliveried to the winding unit. It is wound directly onto the package which can be made with sufficient yarn content of correct build and dimension for all subsequent routines. So, after the winding , the package can be used directly for knitting and weaving.

-Fig 1.3 is a drawing of the winding-unit. The winding unit consists of a delivery shaft (1), delivery roller (2), a straight distribution bar (3), a traverse guide (4) reciprocating in a slot cam (5), and a driving drum (6) on which fits a rubber tyre (7) which drives the package(8) during winding.

In the winding process, the velocities of the yarn are important. After coming out of the spinning unit, the yarn is delivered upward by a delivery shaft (1) and a delivery roller (2) at constant speed. The yarn passes through a tension device which is used to reduce tension variation in the yarn. The yarn then passes over a straight distribution bar (3) to the traverse guide (4). The traverse guide is driven by a follower riding in a cam slot. The profile of the slot cam(5) is so designed that the guide lays the yarn on the package at an appropriate helix angle. The layers of yarn on the package (8) cross one another to give stability. The package, normally has a conical shape, and is driven by the friction force between the outer surface of the package and a rubber tyre (7) on a driving drum (6). The drum rotates, and the rubber tyre drives the conical package to rotate it on its own axis. The rotating package then takes up the yarn from the traverse guide which is moving to and fro periodically along the package, therefore a cross-wound package can be obtained.[6]

1.2 Description of the Project

The working principle of the spinning-winding machine is as described above. During the working process, the fibres in the form of a sliver are made into yarn and then wound onto a conical package.

A conical package has different diameters at different cross-sections. When the package rotates, the peripheral speeds along its surface are therefore different. The smaller the diameter is, the lower peripheral speed it has. If a conical package is driven by a rotating cylindrical drum it is obvious that at only one point between the shoulder (small end) of the package and the base (big end) of the package , can it be driven without slip.

In this particular spinning-winding machine. the conical package is driven by frictional force exerted by a narrow rubber tyre which fits on the drum. Therefore the conical package contacts the drum at only one point (or over a very small area), and at this point the peripheral speed of the drum and that of the conical package are equal, but for other points on the surface of package, the peripheral speeds are different from that of the drum. Towards the small end, the peripheral speed of the package is lower than that of the drum, towards the large end , the peripheral speed is higher.

As mentioned before, in this spinning unit, the yarn production rate is constant and in a winding unit the yarn delivery rate is also constant. The yarn production rate is equal to the yarn delivery rate. If the yarn delivery rate is equal to the drum peripheral speed ,then at the shoulder of the package, because the package peripheral speed is lower than yarn delivery rate, yarn slack will occur between the guide and the delivery roller. On the other hand, at the base, yarn stretch will occur.

During winding, the size of the package is built up. As the diameters of the cross-section of the conical package grow bigger, the peripheral speeds change correspondingly. This also contributes to the difference between the yarn delivery speed and the yarn taken up speed(the winding speed), and thus therefore also affects the amplitude of the yarn slack or stretch which occurs between the guide and delivery roller.

Yarn is made of elastic materials. Any changes of yarn length, during winding, will certainly cause tension variation in the yarn. The importance of tension in the building of a stable package of the required density is undoubted, but excessive tension variation in the yarn is unacceptable as well. Excessive tension variation may cause yarn breaking, partial breaking, variable package density and package collapse etc. So the control of tension variation during yarn winding is critical.

1.3 Purpose of the Research

In the spinning-winding machine to which this research relates, there is a tension device which is used to reduce the tension variation during yarn winding. This is a passive tension compensator and is mainly composed of a spring and a bollard disc (Fig.1.4). In the winding process , when yarn is stretched and tension builds up, the yarn pulls the bollard disc to rotate against the spring in the counterclockwise direction. By releasing some yarn stored in the compensator, the maximum value of tension variation will be reduced. When yarn is loosened and the tension drops, the bollard disc driven by the spring rotates in the clockwise direction to take up some yarn, and therefore helps to keep a certain tension in the yarn.

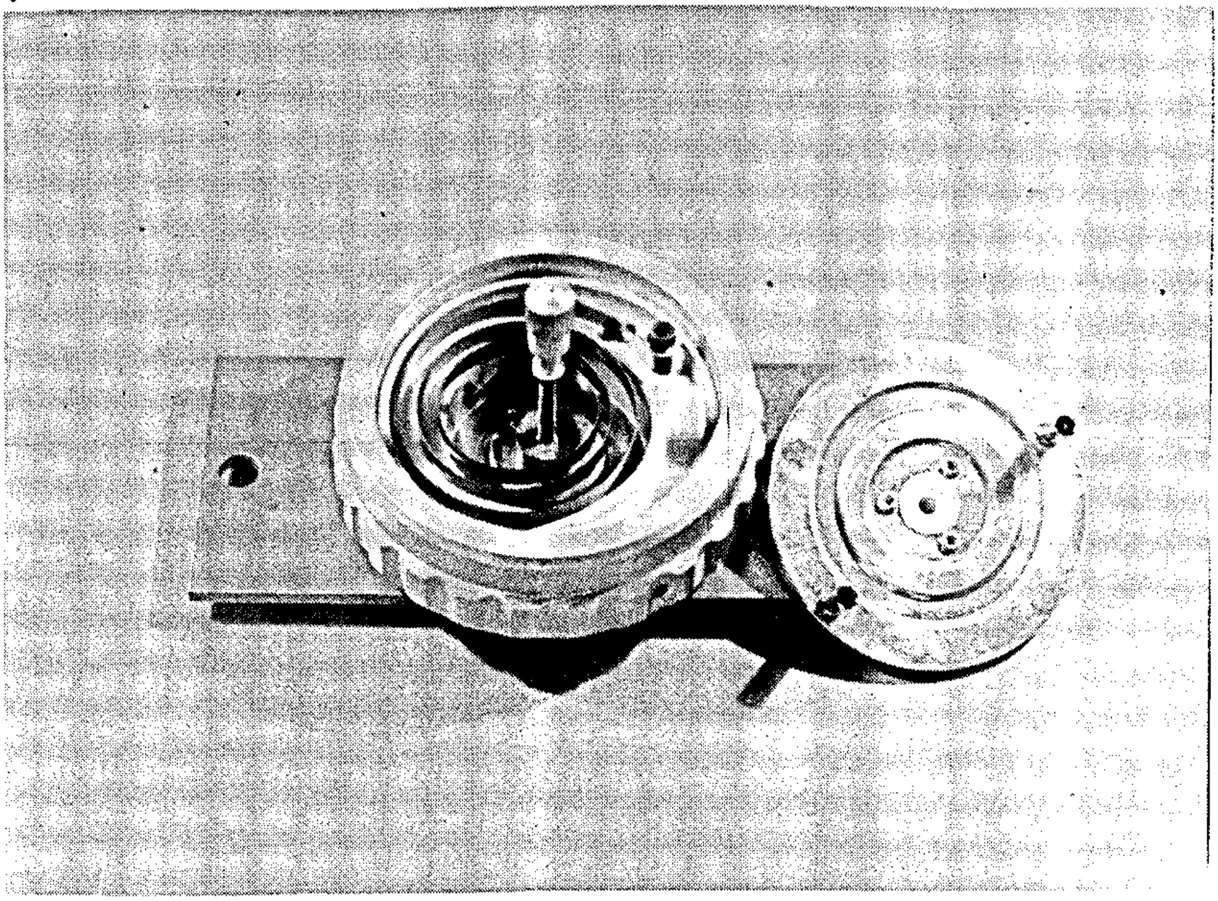


Fig 1.4 Spring Tension Compensator

This tension compensator currently used in the machine is inadequate, especially at high winding speed. Firstly, it is a passive compensator, in that, it is the variation of yarn tension that makes it move. So inevitably there is still some tension variation left. Secondly, because of the inertia of the spring and bollard disc, the compensator cannot respond to the tension variation instantly. Any delay in its response will cause additional tension variation in the yarn.[7]

With the development of textile machinery, the machine speed is getting higher .There is a demand for raising the winding speed from its current value of150 m/min. to 500 m/min. to cope with the increasing productivity of the spinning process. In high speed winding, the tensile force in the yarn changes its direction and amplitude at even higher frequency. However, it is demanded that the amplitude of tension variation should be kept at least at, or under, the current level. The control of yarn tension variation during cone winding is made more important by ever increasing winding speed, and the development of devices to provide this tension control for high speed cone winding is the purpose of this research project.

CHAPTER 2 YARN WINDING VELOCITY VARIATION

In the yarn winding process, yarn is delivered upwards by a delivery shaft, the yarn delivery velocity is constant. However, the yarn winding velocity is not constant. This velocity difference causes an imbalance between yarn supply and demand, and so, yarn surplus or shortage occurs in certain stages of cone winding, this causes yarn winding tension variation. This yarn surplus and shortage are called yarn winding error. The former is defined as negative error, the latter, positive error.

2.1 Winding Velocity Definition and Assumption

Yarn winding velocity is the velocity by which yarn is wound on to a package. As shown in Fig 2.1, yarn winding velocity is composed of package peripheral velocity Vp and guide velocity Vg. Both of them are proportional to yarn delivery velocity. Yarn winding velocity can thus be expressed as:

$$V_w^2 = V_g^2 + V_p^2 \qquad (2.1)$$

The package peripheral velocity depends on the "winding-on" position and the size of the package, the guide velocity does not only depend on the driving cam profile, but is also influenced by pattern breaking (discussed below). All these make the calculation of yarn winding velocity very complicated.

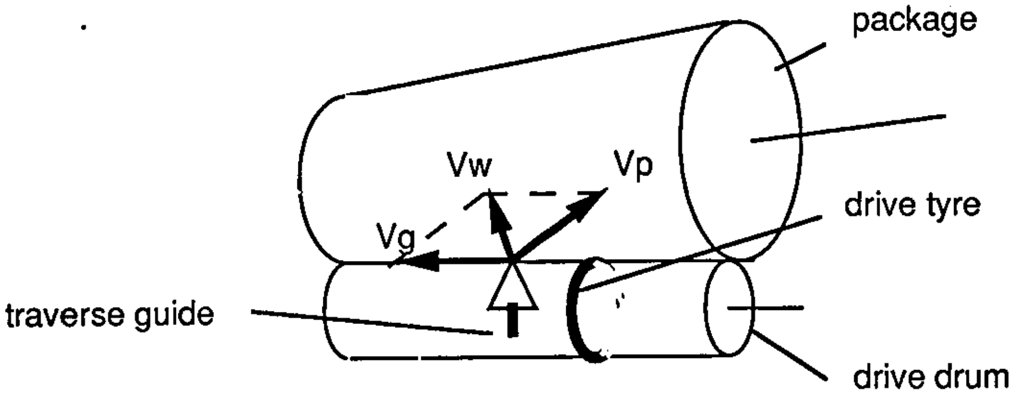


Fig 2.1 Yarn Winding Velocity Calculation

Before calculating yarn winding velocity, some assumption must be made in order to take only the main factors into account, and to avoid some minor variables which are tedious to calculate but have only very little effect on the amplitude of winding velocity.

The following are the assumptions:

1. The conical package contacts the drive drum only at the tyre.
2. There is no slip at the contact point during yarn winding.
3. There is no yarn package deformation at the contact point.
4. There is no gap between the guide and the winding-on point.

2.2 Traverse Guide Velocity

Fig 2.2 shows schematically the motion transmission in the winding machine. It indicates that the traverse guide, which leads yarn to the winding-on position, is driven by a cam.

The guide displacement S depends on the cam profile and cam angular displacement ϕ_C.

$$S = S(\varnothing_c) \tag{2.2}$$

where $\varnothing_c$ is cam angular displacement

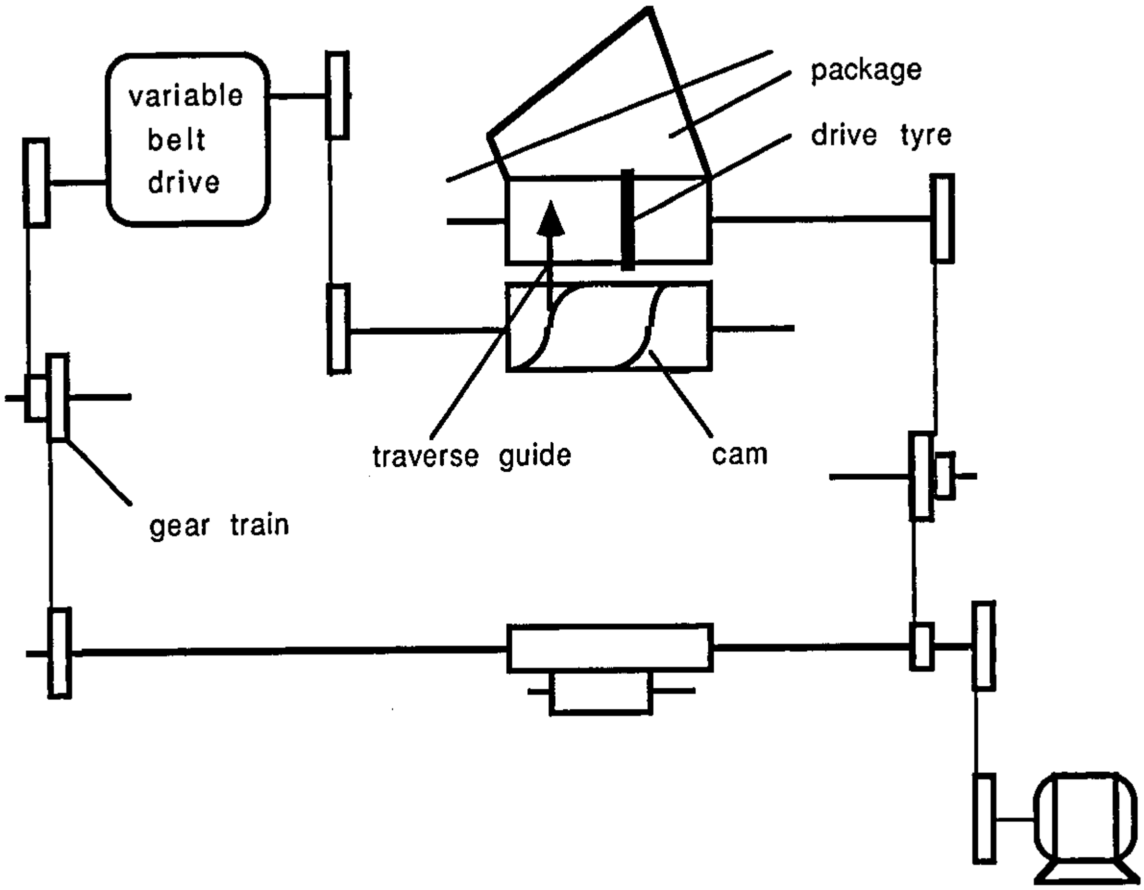


Fig 2.2 Motion Transmission Diagram

The guide velocity can then be expressed as:

$$V_g = \dot{S} = \dot{S}(\varnothing_c) * \omega_c \tag{2.3}$$

where ω_c is cam angular velocity

For this cam which has a "two turn" groove configuration, a complete cycle of guide motion includes two to-and-fro guide motions and one complete cycle of guide motion corresponds to thirteen revolutions of the cam shaft. Fig 2.3 shows the traverse guide displacement in one complete cycle of guide motion.

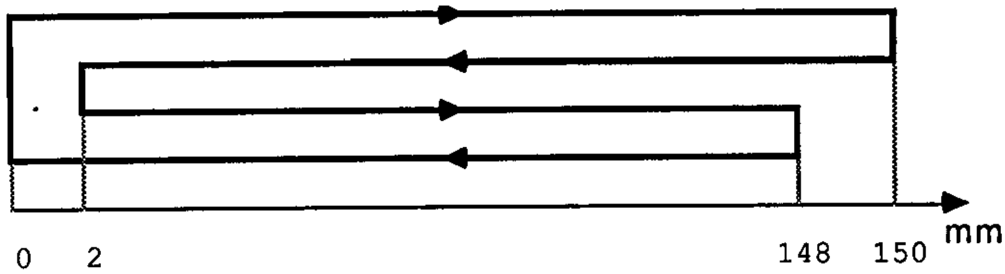


Fig 2.3a Guide Displacement in Two Half Cycles

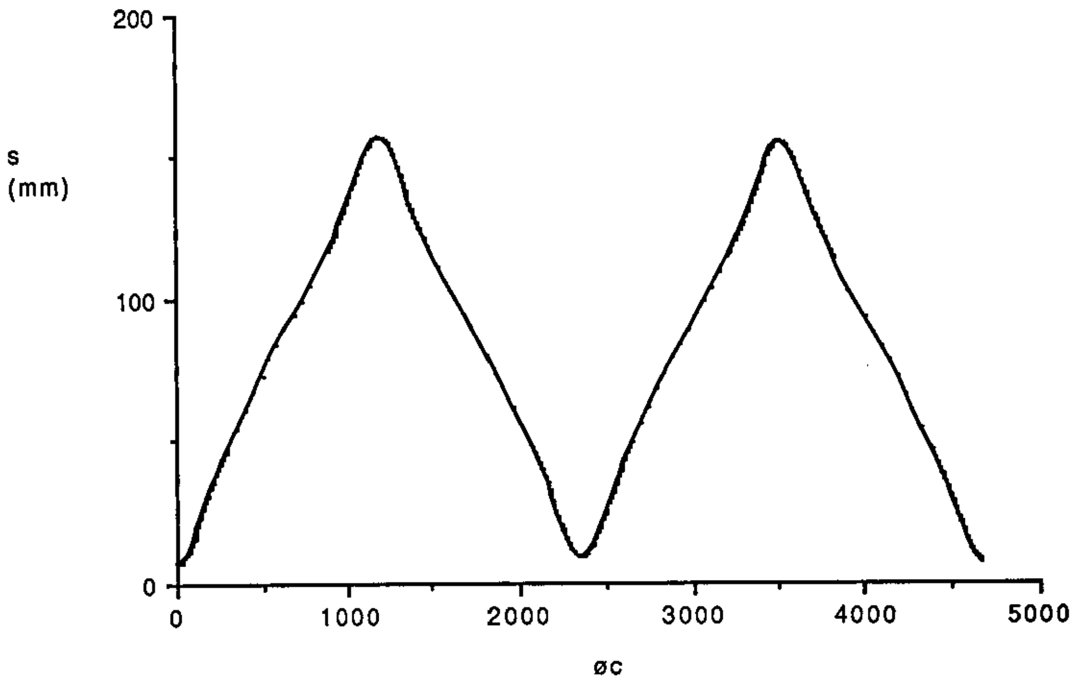


Fig 2.3b Guide Displacement in One Cycle of Motion

Fig 2.3a indicates that the traverse guide displacements in two half cycles are slightly different. The arrows represent the guide movement direction. Fig 2.3b indicates the traverse guide's displacement versus cam angular displacement. It can be seen that in two half cycles the guide moves in a similar way.

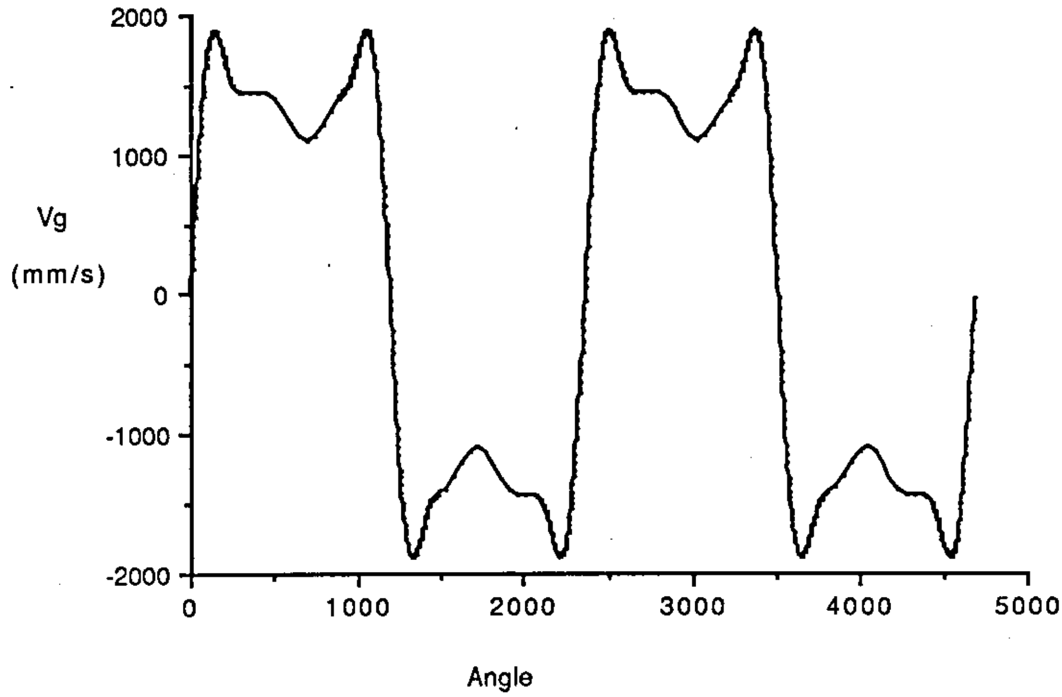


Fig 2.4 Traverse Guide Velocity Diagram

Fig 2.4 is a computer simulation of the traverse guide velocity diagram(yarn delivery velocity Vd=250m/min.). At the beginning of guide displacement, the guide velocity goes up quickly, and for a short period of time the velocity stays high . The high guide velocity produces a large yarn winding angle, that makes the end of the package more stable, the guide velocity then slows down and levels off. Near the midpoint of the guide displacement, the guide velocity drops down again, the lowest guide velocity is obtained at the position where the rubber tyre contacts the package. The low guide velocity produces a small winding angle, which makes this part of package harder. Because the rubber tyre contacts this part of package and drives it by a frictional force, the hard contact area enables the friction drive to work more efficiently. After that, the guide velocity increases again to make a large winding angle at the base of the conical package. The guide velocity then drops down sharply to zero. At this time, the guide changes its direction and begins to move back. On the return journey, the guide velocity is almost symmetrical to the forward journey. The negative velocity means that the guide moves in the opposite direction. It can also be seen that in two to-and-fro motions the guide velocity varies in a very similar way.

If we neglect the effect of the variable belt drive(described below) and assume the ratio of its input and output is 1, then the cam angular velocity is proportional to the delivery shaft angular velocity.

$$\omega_c = \zeta_c * \omega_d \qquad (2.4)$$

where ω_d is delivery shaft angular velocity

ζ_c is the ratio of the gear train from the delivery shaft to the cam shaft

Because the profile of the cam is fixed, the guide velocity at each position is a function of cam angular displacement ϕ_c and delivery shaft angular velocity ω_d.

$$V_g = \dot{S}(\phi_c) * \omega_c = \dot{S}(\phi_c) * \zeta_c * \omega_d \qquad (2.5)$$

The guide velocity itself changes a lot during one cycle of guide motion. However, compared with the package peripheral velocity, the traverse guide velocity is low and is only a small portion of the winding velocity.

2.3 Package Peripheral Velocity

In this winding machine, the conical package is pressed onto the drum by a spring, so that a reasonably constant pressure is maintained at the contact point throughout the building of the package. Around the drum is fitted a rubber tyre. The diameter of this tyre is a little bit bigger than the diameter of the drum, so that the conical package contacts the drum only at the tyre. When the drum rotates, the frictional force between the rubber tyre and the package drives the conical package. The conical package then rotates on its own axis. Although different points along the package surface have different peripheral velocities, because of this type of drive, slip due to velocity difference between other non-contacting points on the package and on the drum can be avoided.

The drive drum is driven by the delivery shaft through a gear train. The ratio of drive drum linear velocity V and the delivery shaft linear velocity V_d is called tension draft ζ_d. Tension draft is a very important parameter because of its influence on the yarn winding tension.

$$V = \zeta_d * V_d = \zeta_d * r_d * \omega_d \qquad (2.6)$$

where r_d is the radius of the delivery shaft

If there is no slip between drive drum and package, then V is also the peripheral velocity of the contact point of the package.

The calculation of package peripheral velocity is illustrated in Fig.2.5. The conical package contacts the drive drum at the position

A, where the radius of rotation of the package is r_0.

The angular velocity of the package can be written as:

$$\omega_p = \frac{V}{r_0} \tag{2.7}$$

The package peripheral velocity is

$$V_i = \omega_p * r_i \tag{2.8}$$

According to Fig.2.5, radius of rotation can be expressed as:

$$r_i = R + S_i * \sin(\varnothing) + z * \cos(\varnothing) \tag{2.9}$$

where S_i is the guide displacement.

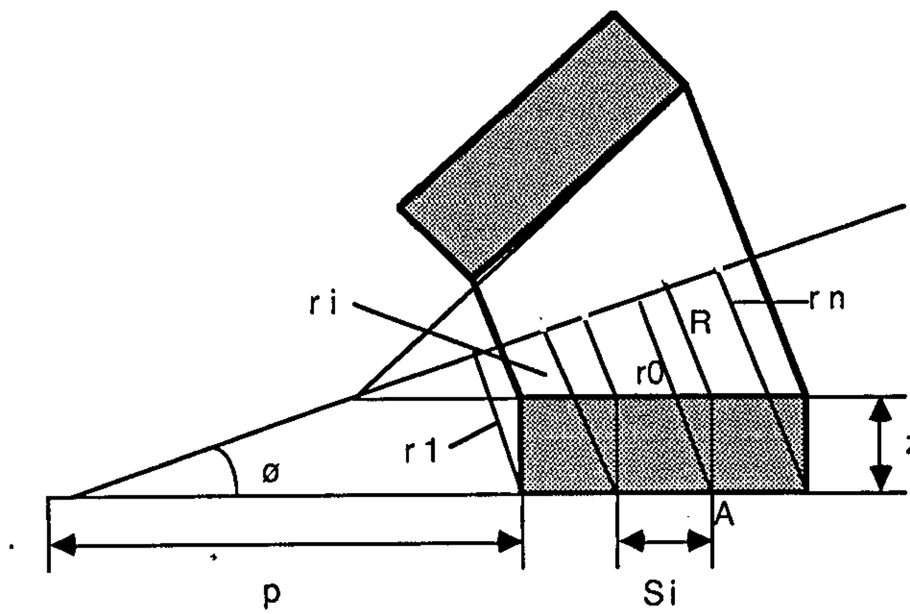


Fig. 2.5 Package Peripheral Velocity Calculation

We define that at the contact point Si= 0, to the right of the contact point Si is positive, to the left Si is negative. The thickness of the yarn layer is z. The package peripheral velocity can be rewritten as:

$$V_i = \omega_p * r_i = V * \frac{r_i}{r_0} = \zeta_d * r_d * \omega_d * \frac{r_i}{r_0} \qquad (2.10)$$

By substituting equation (9) into (10), we can get

$$V_i = \zeta_d * r_d * \omega_d * \frac{R + S_i * \sin(\varnothing) + z * \cos(\varnothing)}{R + z^* \cos(\varnothing)} \qquad (2.11)$$

where $S_i = S_i(\varnothing_c)$. The peripheral velocity is a function of cam angular displacement $\varnothing_c$, delivery shaft angular velocity ω_d and the package size z. At the left end of the package, the peripheral velocity can be expressed as:

$$V_1 = \omega_p * r_1$$

At the right end:

$$V_n = \omega_p * r_n$$

Since $r_n > r_0 > r1$, then, $V_n > V > V_1$. That indicates, at the left, the package peripheral velocity is lower than the drive drum velocity, at the right , peripheral velocity is higher.

2 .4 Pattern Breaking

In the winding process, the guide traverses from one end to the other and lays the yarn onto the package. The guide velocity does not only depend on the cam profile but also depends on the cam rotation speed. Any change of the cam shaft rotational speed will also change the guide traverse velocity. In this spinning-winding machine, there is a variable belt drive which changes the cam rotation speed slightly from time to time. The purpose of this device is to introduce pattern breaking.

The term "pattern" refers to the phenomenon that during winding the yarn is wound on to the package at an exact position repeatedly and therefore piled up at the same place. As stated before , the yarn is wound onto the package at a helix angle, if yarn is piled at the same position, there will be a lot of raised helical ribs on the surface of the package. The package will have different densities at different parts and this kind of package is difficult to unwind. That is not acceptable for subsequent package processing. To obtain a high quality yarn package, pattern breaking is a necessary process in the package winding.

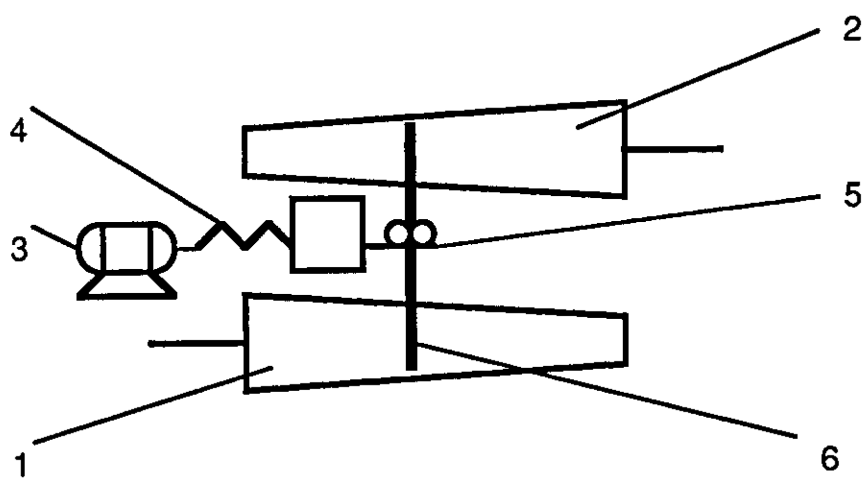


Fig 2.6 Variable Belt Drive Device

Fig 2.6 shows a variable belt drive device.

The variable belt drive device consists of a driving tapered roller(1), driven tapered roller(2), motor(3), screw(4), belt guide roller(5), and flat belt(6) which is mounted on the tapered rollers with a certain amount of tension and transmits peripheral force by friction.

Rotation is transmitted from the input roller(1) to the output roller(2) by means of the flat belt(6). Because the input roller and the output roller have tapered shapes, it is obvious that the position of the belt will decide the input-output rotation ratio. When the belt is at the left side, as shown in Fig 2.6 the diameter of the input roller is bigger than that of the output roller, and the input-output ratio is less than one. When the belt at the right side , the ratio is greater than one.[8]

During the winding process, the motor drives the screw , the rotating screw changes the position of belt guide roller slowly, and consequently this changes the belt position and the input-output ratio, The input angular velocity is constant, but because of this variable belt drive, the output angular velocity varies with time.

Since the driving and driven rollers have small cone angles, the input-output ratio changes only very slightly. The variable angular velocity of the output roller also changes the angular velocity of the cam shaft because they are connected by a toothed belt. The variation of cam shaft angular velocity results in a slight guide velocity variation, so that in the each cycle of winding , yarn does not wind onto the exact position of the previous cycle, thereby introducing the desired pattern breaking.

For this particular device the diameter of the small end of the taper roller is 99mm, the diameter of big end of the taper roller is 100mm. If slip between belt and roller is ignored, then

$$\omega_i * R_i = \omega_o * R_o$$

$$\omega_o = \frac{R_i}{R_o} * \omega_i$$

where ω_i is the input shaft angular velocity

R_i is the radius of the input roller

ω_o is the output shaft angular velocity

R_o is the radius of the output roller

$$\omega_{o1} - \omega_{o2} = \left(\frac{100}{99} - \frac{99}{100}\right) * \omega_i = 2\% \ \omega_i$$

where ω_{o1} is the output shaft angular velocity, when the belt is at the left.

ω_{o2} is the output shaft angular velocity, when the belt is at the right.

Under the action of the variable belt drive, when the belt moves from the left to the right end, the output shaft angular velocity changes about 2%. The output shaft then passes this variation to the cam shaft. Even though the cam shaft angular velocity variation is quite small, it is sufficient for pattern breaking.

Compared with the package peripheral velocity, guide velocity only contributes only a small portion to the winding velocity. Since pattern breaking affects the guide velocity by only 2%, neglecting the effect of pattern breaking will not substantially affect the calculation of yarn winding velocity.

2.5. Yarn Winding Velocity

Winding velocity is the velocity at which yarn is wound onto the package. Both of its components Vg and Vp change continually during cone winding.

substituting equation (2.5), (2.10), (2.11) into (2.1), we can can get

$$V_w = \sqrt{V_g^2 + V_p^2} = \omega_d * \sqrt{[\dot{S}(\varnothing_c)*\zeta_c]^2 + (\zeta_d * r_d * \frac{r_i}{r_0})^2} \quad (2.12)$$

and

$$V_w = \omega_d * \sqrt{[\dot{S}(\varnothing_c)*\zeta_c]^2 + [\zeta_d * r_d * \frac{R + S_i(\varnothing_c)*\sin(\varnothing) + z*\cos(\varnothing)}{R + z*\cos(\varnothing)}]^2} \quad (2.13)$$

From equation (2.13), it can be seen that winding velocity is a function of cam angular displacement $\varnothing_c$, yarn layer thickness z and delivery shaft angular velocity ω_d. For a particular machine setting, the cam profile is fixed and ζ_c, ζ_d, $\varnothing$, R are constant.

Fig2.7 is a computer simulation of the winding velocity diagram (Vd=250m/min., z=30mm). It indicates the winding velocity variation in one cycle of guide motion. The winding velocity goes up and down in a very similar way in the two half cycles. At the shoulder of the package winding velocity goes up sharply, then drops for a short time due to the guide velocity decreasing there. After that, the velocity goes up steadily towards the base of the package. Contrary to expectation, the highest winding velocity doesn't occur right at the big end of the package. Near the very end the winding velocity drops down noticeably, because the traverse guide velocity decelerates there, and changes its direction of movement.

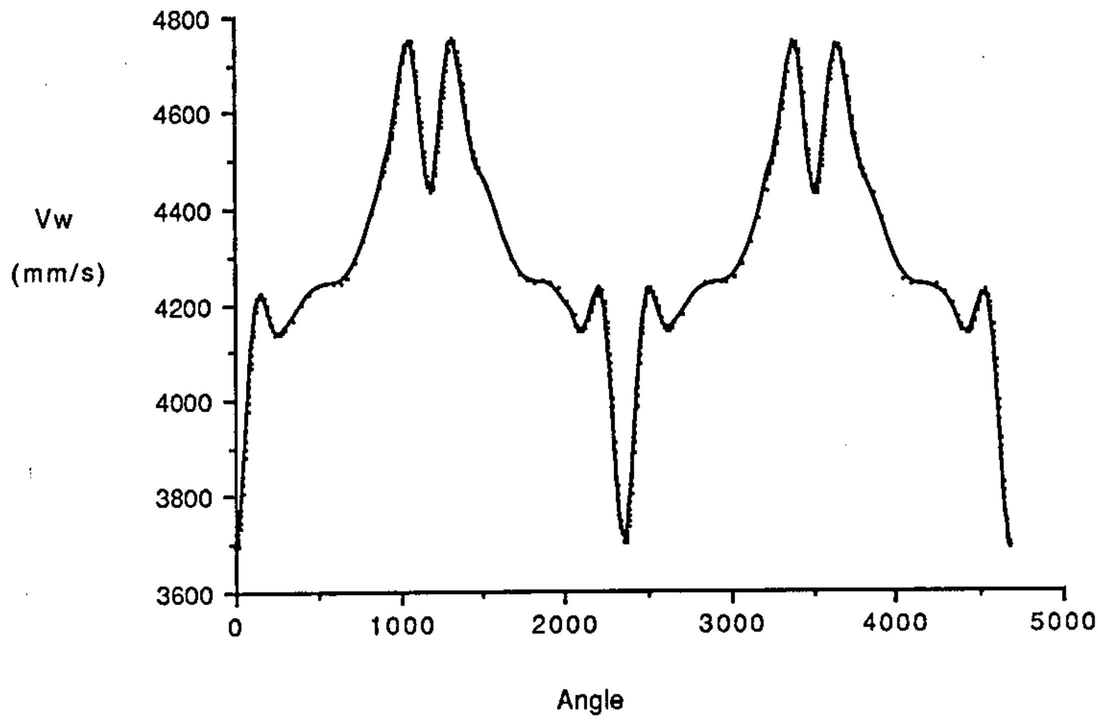


Fig 2.7 Winding Velocity Diagram

2.6 Yarn Winding Error

Since yarn winding velocity is a function of yarn delivery velocity, traverse guide displacement and the package size, and is not constant, yarn winding error, which is the difference between yarn supply and demand, will occur in the cone winding. As stated before, yarn winding error is due to the difference between yarn winding velocity and yarn delivery velocity. Yarn winding error in one cycle of guide motion can be expressed as:

$$e = \int_0^T (V_w - V_d) * dt$$

$$= \int_0^T \{\sqrt{[S_i(\varnothing_c)*\zeta_c]^2 + [\zeta_d * r_d * \frac{R+S_i(\varnothing_c)* \sin(\varnothing)+z*\cos(\varnothing)}{R + z*\cos(\varnothing)}]^2}$$

$$- r_d\} * \omega_d * dt \qquad (2.14)$$

$$\varnothing_c = \omega_c * t$$

$$t = \frac{\varnothing_c}{\omega_c}$$

So, equation (2.14) can be rewritten as:

$$e = \int_0^{\omega_c * T} \{\sqrt{[S_i(\varnothing_c)*\zeta_c]^2+[\zeta_d*r_d*\frac{R+S_i(\varnothing_c)*\sin(\varnothing)+z*\cos(\varnothing)}{R + z*\cos(\varnothing)}]^2}$$

$$- r_d\} * \frac{\omega_d}{\omega_c} * d\varnothing_c$$

because $\omega_c=\zeta_c*\omega_d$, then

$$e = \int_0^{\varnothing_{cc}} \{\sqrt{[S_i(\varnothing_c)]^2+[\frac{\zeta_d}{\zeta_c}* r_d * \frac{R+S_i(\varnothing_c)*\sin(\varnothing)+z*\cos(\varnothing)}{R+z*\cos(\varnothing)}]^2} - r_d\}* d\varnothing_c \qquad (2.15)$$

where $\varnothing_{cc}$ is the cam angular displacement in one cycle of guide motion.

From equation(2.15), it can be seen that yarn winding error is only a function of cam angular displacement $\varnothing_c$ and package size z. Unlike winding velocity, yarn winding error, which affects the yarn winding tension variation, is not a function of delivery shaft angular velocity . In other words, the yarn delivery velocity should not affect the yarn winding tension variation..

Fig 2.8 Yarn Winding Error Diagram

Fig 2.8 is a computer simulation of the yarn winding error in one cycle of traverse guide motion(Vd=250m/min., z=0mm). The traverse guide starts at the small end of package, at the beginning stage, the winding error is negative, which indicates that more yarn is supplied to the winding unit than required. As the guide moves toward the big end of the package, winding error becomes positive, which means that more yarn is required than is supplied, which also implies that winding tension builds up at this stage. As the second half cycle starts, the magnitude of winding error drops due to the low winding velocity.

Because of the winding error accumulation in the previous stage, the value of winding error remains positive. If we neglect the starting value of the two half cycles, the two error curves look almost identical. So in the tension compensator design, it is possible to take only half the cycle of yarn winding error into consideration. The other half can be regarded as similar.

CHAPTER 3 YARN WINDING TENSION VARIATION

3.1 Introduction to Yarn Winding Tension

When yarn is formed and pulled out of the spinning unit, tension has already been produced in the yarn. Yarn tension plays an important role in winding. Too high a tension can damage yarn, whereas too low a tension can lead to an unstable package which will not be unwind cleanly. A common fault associated with certain loosely wound packages is the tendency to "slough off" more than one turn to give a tangle. On the other hand, excessive winding tension can produce elongation, and possible breakage and yarn imperfection. . Yarn tension variation in different parts of a wound package can cause undesirable effects. For instance, with many man-made fibres, high tension can cause fibre structure change which affects the dye-ability, so variation in tension ultimately shows as apparently random variations in colour shading.[9]

"Winding tension should be maintained constant at as low a figure as is consistent with the build of a suitable package which will readily unwind at the next process and which will withstand internal transport without yarn sloughing off." ---- F. Millard[10]

The tension in the yarn is created mainly by twisting, bending, drawing and winding . Many publications have dealt with most of these subjects, however very few systematic works on the winding tension variation in friction spinning of yarn were found after an extensive survey of the literature. There is no doubt that the yarn should be wound under a uniform tension. This both creates a consistent package and minimises any variational effects in the yarn which may be a function of tension. In this chapter, winding tension variation will therefore be discussed in turn.

There are a lot of factors which affect the winding tension variation in the yarn . Among them are yarn winding error variation, yarn path length variation and frictional force variation. These factors will be discussed respectively.

3.2 Tension Variation Due to Winding Velocity Variation

The analysis of yarn winding error in chapter 2 has indicated that the difference between the amount of yarn being delivered up and wound onto the conical package contributes to the yarn winding tension. However, some tension is necessary to keep a stable conical package. What we need to do is to eliminate winding tension variation not winding tension itself.

Fig2.8 is a computer simulation of the yarn winding error in one complete cycle of guide motion, the traverse guide starts at the small end of an empty package. If a straight line is drawn to connect the starting point and the end point of the curve, then the yarn winding error can be divided into two components. The straight line represents the first component of yarn winding error which increases at a constant rate. In the yarn winding process, this part of yarn winding error contributes a constant winding tension to the yarn. The constant winding tension is desired for further package processing. If we neglect this component of the yarn winding error and take the straight line as the axis on which to redraw this diagram, then we get Fig3.1 which represents the second component of yarn winding error. This part of the yarn winding error keeps changing its amplitude as the traverse guide moves back and forth. It is this part of the yarn winding error that causes winding tension variation. From now on, we will only concern ourselves with this part of yarn winding error .

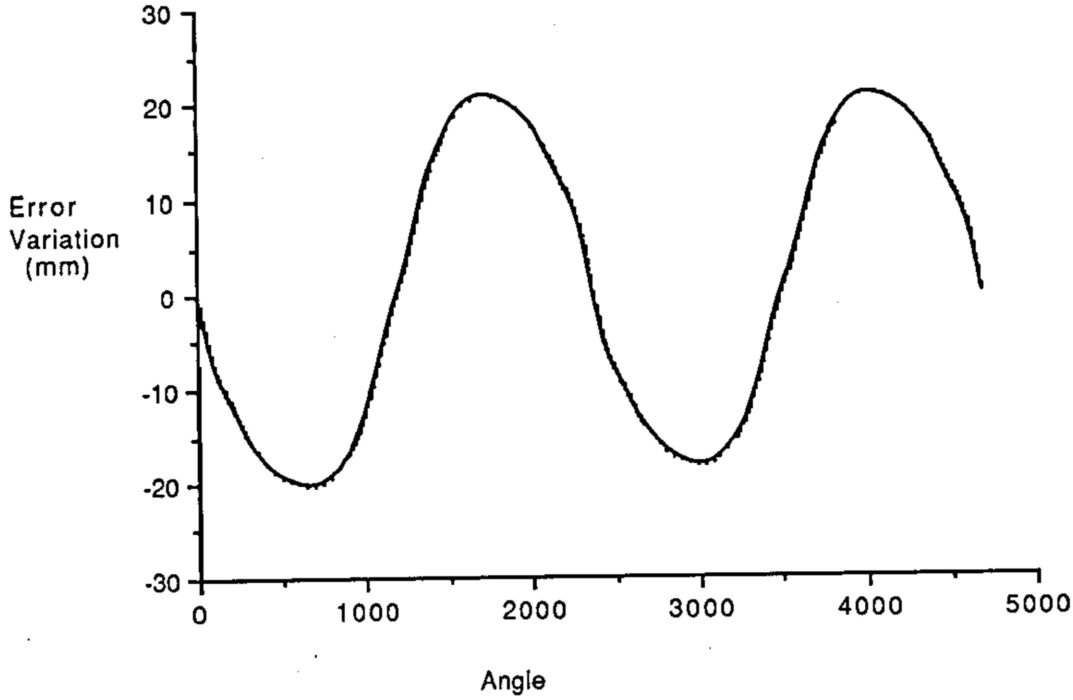


Fig 3.1 Winding Error Variation Diagram

Fig3.1 is a computer simulation of yarn winding error variation in one cycle of guide motion. The winding condition is: yarn delivery velocity Vd=150 m/min., package size z=0 mm, traverse guide initial position at the shoulder of the package. It can be seen that at first the winding error is negative. In practice, yarn winding tension should never be negative, if yarn winding error is negative, that simply means that more yarn is provided to the winding unit than is actually needed. Consequently, yarn slack occurs and the yarn winding tension drops to zero. As the traverse guide moves toward the base of the package, the winding error changes to positive, which means extra winding tension has been built up in the yarn and yarn is further stretched. In the second half cycle of guide motion, the above procedure repeats in a similar way.

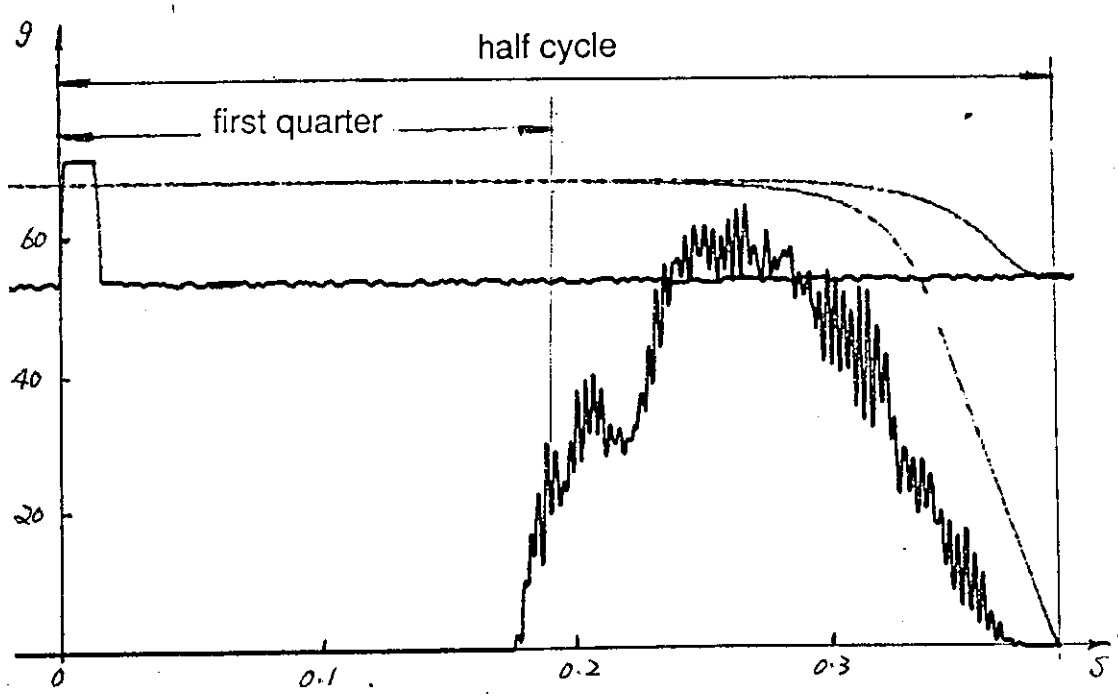


Fig 3.2 Yarn Winding Tension Variation
Tension (gram) Versus Time (second)

Fig3.2 is the record of an experiment of yarn winding tension variation, here, the square curve indicates that the traverse guide is at the shoulder of conical package, at the first quarter of the half cycle, the winding tension in the yarn is almost zero. Only at the second quarter does the tension gradually build up and then come down. Compared to Fig3.1 and 3.2 the tension variation curve is just as we predicted in the computer simulation .

Yarn winding error variation is the major factor which affects the winding tension variation. In the following paragraph, we will discuss some relevant parameters and look at their respective effects on winding tension variation.

3.2.1 Package Size

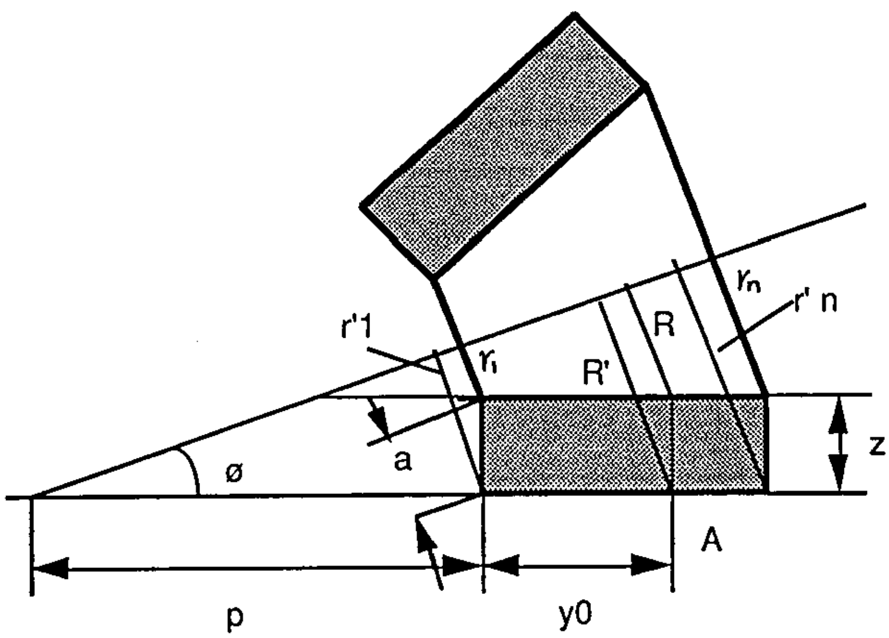


Fig 3.3 Package Size and Peripheral Velocity

In order to describe the problem more clearly, Fig.2.5 is redrawn into Fig3.3. It can be seen that as the winding process proceeds , the package size increases, R becomes R' ,

$$R'=R+a \quad (3.1)$$

where a is the projection of yarn layer thickness on the radius of rotation of the package. According to equation(2.7), (2.8), the angular velocity of the package changes to ω'

$$\omega' = \frac{V}{R+a} \tag{3.2}$$

Consequently, the package peripheral velocity also changes,

$$V_1' = \omega' * r_1' = V * \frac{r_1 + a}{R+a} \tag{3.3}$$

$$V_n' = \omega' * r_n' = V * \frac{r_n + a}{R+a} \tag{3.4}$$

So, after a layer of yarn is laid on the package, a peripheral velocity increment occurs which can be expressed as follows:

$$V_1' - V_1 = \frac{V}{R+a} * (r_1 + a) - \frac{V}{R} * r_1 = \frac{V * a}{(R+a) * R} * (R - r_1) \tag{3.5}$$

$$V_n' - V_n = \frac{V}{R+a} * (r_n + a) - \frac{V}{R} * r_n = \frac{V * a}{(R+a) * R} * (R - r_n) \tag{3.6}$$

Since $r_n > R > r1$, we have

$$V_1' - V_1 > 0 \text{ and } V_n' - V_n < 0$$

If we take the contact point between package and drive drum as a boundary, as the package size builds up, the peripheral velocity at the left part increases, whereas, at the right part it decreases.

Based on the analysis of guide velocity and package peripheral velocity, we know that peripheral velocity is a major component of winding velocity. As package size increases, at the left side, because the peripheral velocity increases, the negative yarn winding error increases(the amplitude becomes small), at the right side, because the peripheral velocity decreases, the positive yarn error decreases as well.

Fig3.4 is a computer simulation of the yarn winding error variation at different package sizes. For a small package, the yarn winding error is much bigger than for a big package. It indicates that when the package size is small the winding tension variation is much more violent.

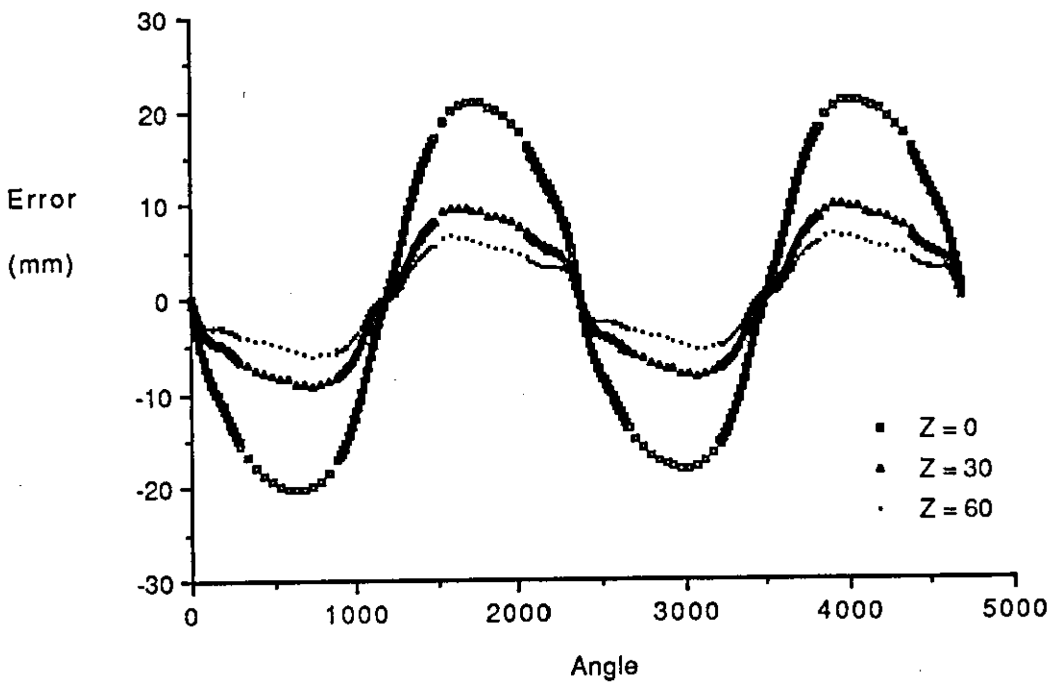


Fig 3.4 Yarn Winding Error at Different Package Size

As package size increases, the amplitude of yarn winding error decreases. The winding tension variation , which is due to the yarn winding error variation, will gradually reduce.

3.2.2 Guide Displacement

Guide displacement $S=S(\phi_c)$ depends on the cam angular displacement ϕ_c. In one cycle of guide motion, z is constant, according to the equation (2.15), S_i is the only variable which affects the winding error variation. According to the definition of S_i, at the left side of the contact point, S is negative, at the right, S is positive. the effect of S is inferred in the equation (2.13). When yarn is wound on the left part of the package, the winding velocity is low. So it tends to get a negative winding error ,whereas at the right part, a positive error. However, due to the winding error accumulation, in the first quarter of the half cycle, it is more likely to get a negative winding error and the winding tension remains zero. In the second quarter of the half cycle, a positive winding error gradually builds up and then drops down. This indicates the extra tension goes up and down in the second quarter of the guide motion cycle. Fig3.1 is a computer simulation of yarn winding error variation at the different guide displacement. Fig3.2 is the record of the tension experiment. Both of them have proved the above explanation.

3.2.3 Delivery Velocity

Delivery velocity V_d is sometimes called yarn velocity, as shown on the control panel of Masterspinner and depends on the production rate of the spinning unit. There is a belief that the value of V_d influences the winding tension variation, however, this is not true. According to equation (2.5), (2.11), (2.13), V_g, V_p and V_w are proportional to the delivery shaft angular velocity. Any increase in delivery shaft velocity will increase the guide velocity and the peripheral velocity, and, therefore winding velocity. At the same time, the increase of guide velocity consequently reduces the time period T which a complete cycle of guide motion would take. The relation between T and V_g can be expressed as:

$$T = \int_0^{\phi_{cc}} \frac{S(\phi_c)}{V_g(\phi_c)} * d\phi_c \qquad (3.7)$$

From equation (2.13) and (3.7), it is quite clear that the increasing delivery shaft angular velocity ω_d proportionally increases the yarn winding velocity Vw and proportionally decreases the time period T. It does not affect the yarn winding error, therefore, does not affect winding tension variation.

3.2.4 Yarn winding tension

A winding tension experiment has been carried out to measure the tension variation in the yarn winding process. A sensitive tension meter was used to pick up the yarn tension during winding, and display it on a digital storage oscilloscope, and then output it to a plotter.

Fig.No.	Delivery Velocity	Yarn Layer Thickness
3.5a	250 m/min	20 mm
3.5b	400 m/min	20 mm
3.6a	150 m/min	0 mm
3.6b	150 m/min	4 mm
3.6c	150 m/min	8mm

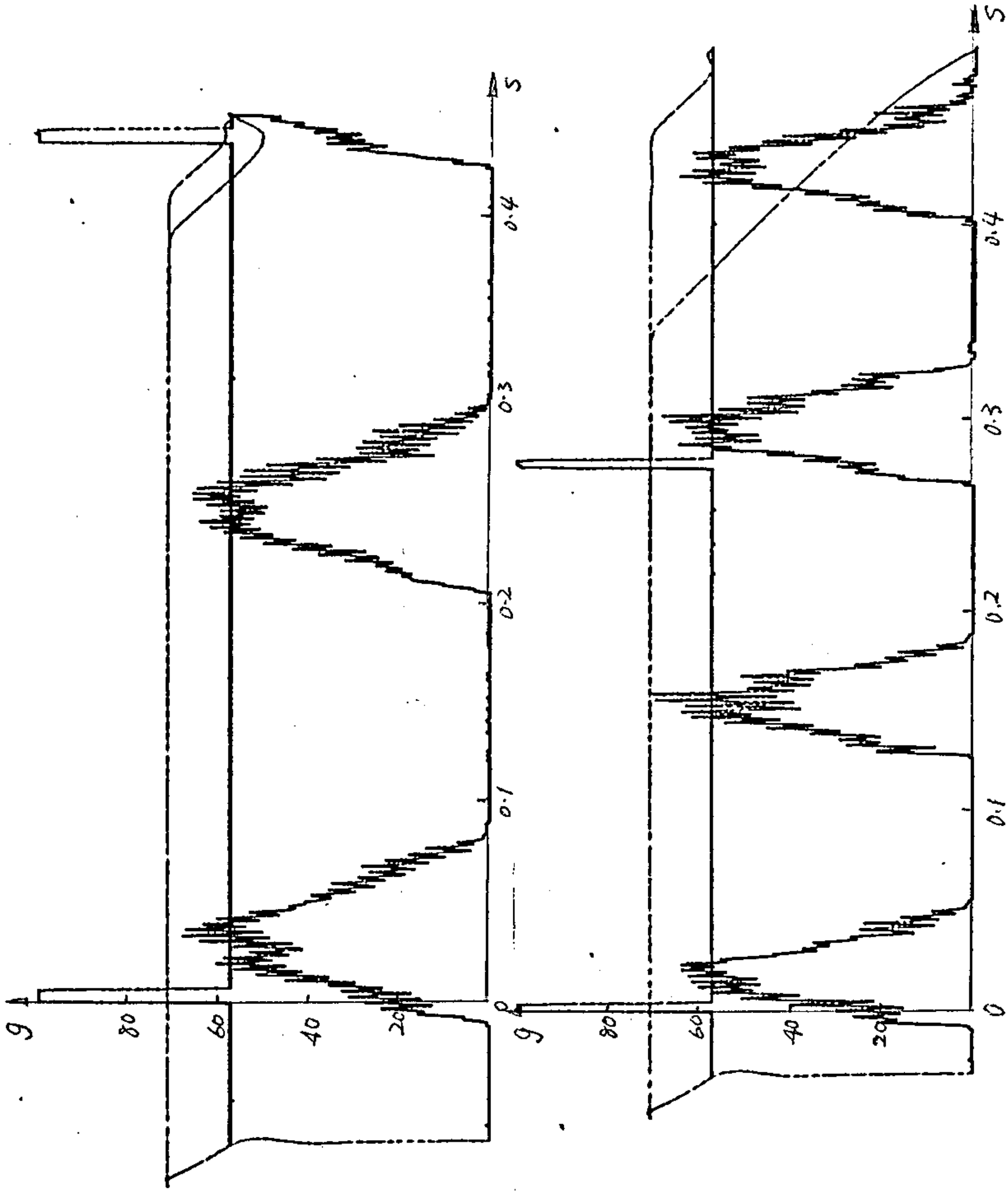
Table 3.1 Conditions of Tension Experiment

The current machine setting is that the tension draft ζ_d=0.981, the cone angle of package $\varnothing$=3.8° and contact point position y_0=80 mm.(see fig3.3) Experiments have been carried out at different yarn delivery velocities and package sizes. Experimental conditions are listed in table3.1.

Fig3.5a and 3.5b show the winding tension variation at the same package size(20 mm), but at different yarn delivery velocity. It can be seen that these tension curves have almost the same amplitude. It indicates that the yarn delivery velocity does not affect the amplitude of winding tension variation.

Fig. 3.6a, 3.6b and 3.6c show the winding tension variation at the same yarn delivery velocity (150 m/min.), but at different package size. It can be seen that when the package size increases from 0mm to 8mm, the winding tension drops to nearly half of its original amplitude. It indicates that as the package size builds up, the yarn tension variation drops dramatically. the smaller the package size is, the more violent the winding tension variation.

The results of these yarn winding tension experiments confirm what was predicted in the yarn winding error analysis and the computer simulation.



(a) (b)

Fig 3.5 Winding Tension at Different Delivery Velocity

(a) Vd = 250 m/min (b) Vd = 400 m/min

Tension (gram) Versus Time (second)

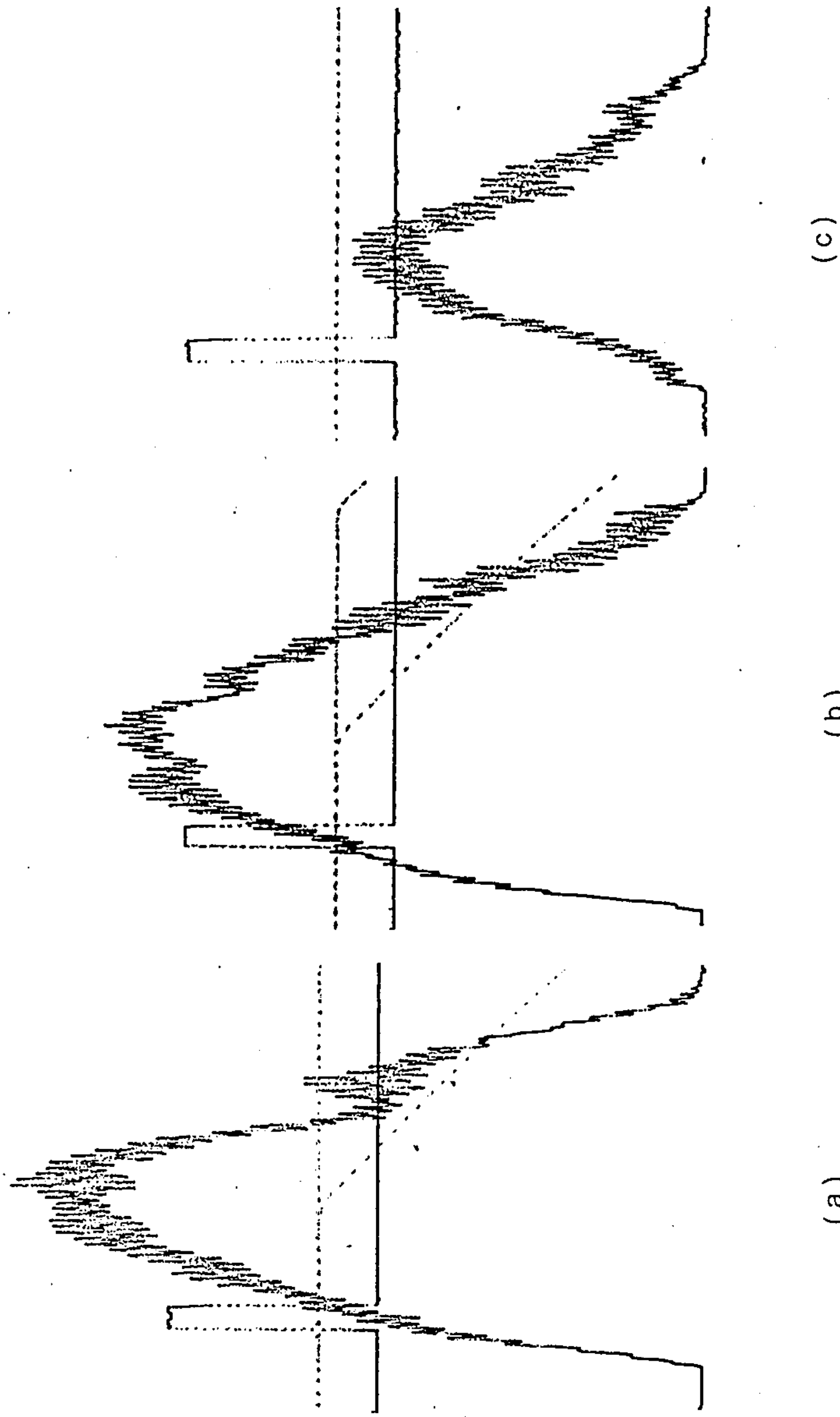


Fig 3.6 Winding Tension at Different Package Size

(a) z = 0 mm (b) z = 4 mm (c) z = 8 mm

3.3 Tension Variation due to Yarn Path Length Variation

In the Masterspinner, after the yarn leaves the spinning unit, it passes a delivery shaft, then goes over a straight distribution bar and is wound onto a conical package. The distance between the traverse guide and the delivery shaft is called the yarn path length.

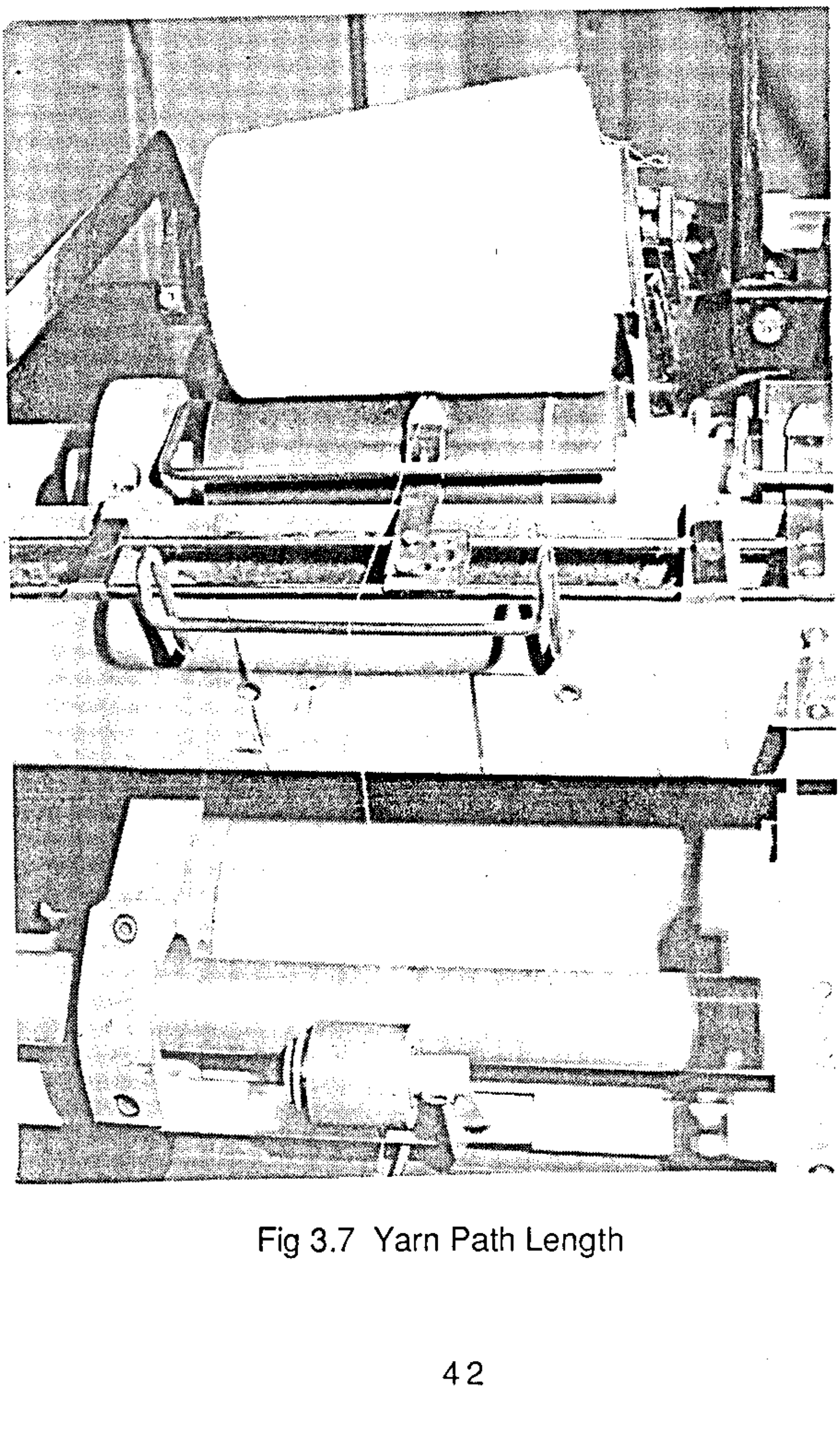


Fig 3.7 Yarn Path Length

The path length changes substantially as the guide traverses from one end of the package to the other. Fig 3.7 shows the yarn path length in the winding machine.

Some machine manufacturers have noticed this yarn path length variation problem. For example, Shlafhorst Company has put fish-belly compensation bows in their machine in order to provide additional yarn in the extreme winding position.[11]

In the following paragraph, the yarn path length is analysed and calculated. The records of tension variation due to the yarn path length variation is also presented.

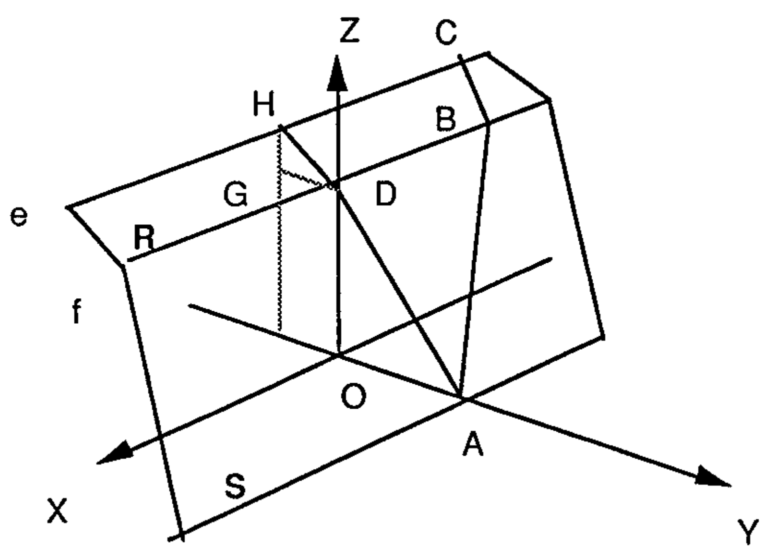


Fig 3.8a Yarn Path Length Parameters

Fig 3.8a is a drawing which shows the machine parameters affecting the yarn path length.

In the above figure:

e is the line along which the drive drum contacts the package. The yarn winds onto the package along this line.

f is the line which represents the straight distribution bar . During winding, yarn passes over and traverses along this line.

C the winding-on point corresponds to the position of the reciprocating traverse guide.

A is a fixed guide through which the yarn passes prior to passing over the distribution bar.

B is the moving point on the distribution bar over which the yarn passes on its way from A to C.

Point A,D,G,H are in the plane x=0

GH is the vertical distance between line e and line f.

DG is the horizontal distance between line e and line f.

The coordinates of points A, B and C are:

A(0, OA, 0), B(DB, 0, OD), C(HC, DG, OD+GH)

For a particular machine setting, the following parameters are fixed:

OA=m, OD=n, DG=p, GH=q

So, the coordinates of A, B and C can be expressed as:

A(0, m, 0)

B(DB, 0, n)

C(x, p, n+q)

During winding, the winding-on point C traverses back and forth along the line e , the point B traverses correspondingly along the line f in a similar way. The guide displacement x is the coordinate of C in X axis.

Before the analysis of yarn path length, some assumption must be made, in order to simplify the calculation:

1. The friction between the yarn and the distribution bar can be neglected.

2. The lag in the yarn-bar contact point can be ignored.

3. Yarn is stretched under positive tension during the winding process.

In fig3.8a, A is a fixed point, and C is a traversing point whose displacement depends on the guide displacement. Since yarn is always stretched during winding, point B must be at such a point that the yarn path length between A and C is at its shortest.

If the plane **R**, in which the intersection lines *e* and DH lie, is rotated to coincide with plane **S** in which *f* and AD lie. (Fig 3.8a) Then, *e* becomes *e'*, C becomes C', and H becomes H' as shown in Fig 3.8b. According to planar geometry, the essential condition for the shortest distance between A, B, C' is that the points A, B, C' should lie on the same straight line.

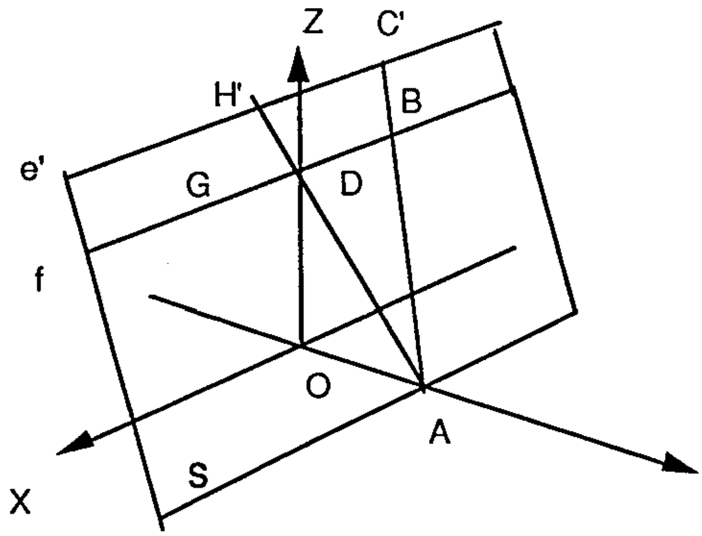


Fig 3.8b Yarn Path Length Calculation

On plane **S**, the yarn path length L can be written as:

$$L^2 = (AH')^2 + (H'C')^2$$
$$= (\sqrt{m^2+n^2} + \sqrt{p^2+q^2})^2 + x^2 \qquad (3.8)$$

where x is the traverse guide displacement.

In the Masterspinner, m=68 mm

n=200 mm

p=165 mm

q=35 mm

the guide maximum displacement is 150 mm. Then the yarn path length can be calculted.

When the guide is at the middle point, the yarn path length is Lo

Lo=380 mm

When the guide is at either end of the package the yarn path length is L

L=387 mm

For this winding machine, in which a straight distribution bar is employed, the yarn path length changes by 7 mm when the traverse guide moves from the end to the middle of the package. So even if we ignore the yarn winding velocity variation, the yarn path length variation will certainly cause yarn winding tension variation.

Fig 3.9a shows a record of yarn tension variation in cone winding when a straight distribution bar is employed. The square pulse indicates that the traverse guide is at the small end of the package. it can be seen that tension peaks occur at each end of the package. This is because where the yarn path length is longer, the demand for more yarn to cover the path length stretches the yarn further and therefore increases the tension in the yarn.

A curved bar which can eliminate the yarn path length variation by changing the position of moving point B is recommended to replace the straight distribution bar. The design of the profile of the curved bar will be illustrated later. The results of the tension experiment shown in Fig 3.9b were obtained by employing a curved bar under exactly the same winding condition as in Fig3.9a. Comparing Fig3.9a with 3.9b, it can be seen that by employing a curved bar, the yarn tension variation due to yarn path length variation can be significantly reduced.

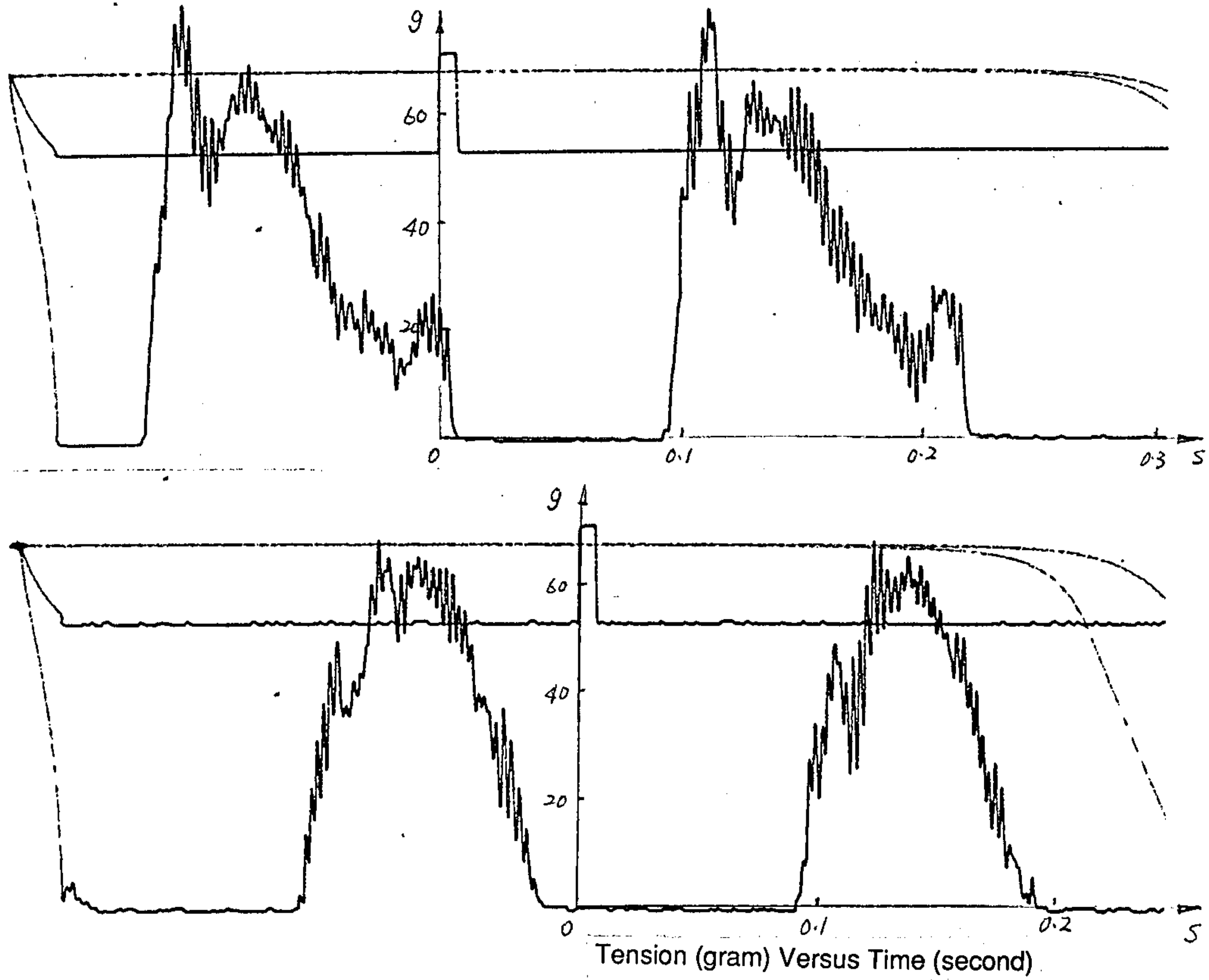


Fig 3.9 Yarn Winding Tension Variation

(a) with a straight bar

(b) with a curved bar

3.4 Tension Variation due to Friction Variation

It is well known that friction assumes an important role in textile technology. Many papers dealing with the theoretical aspects of friction in textiles have been published[12~14], but information about frictional effects which occur during yarn winding is limited. Since tension in running yarn can be reduced by decreasing the frictional forces at all points where the yarn comes in contact with parts of the machinery, it is necessary to do further investigation to see effects of friction variation on yarn winding tension variation.

Buckle and Pollitt have described methods and instruments used for measuring the coefficient of friction between running yarn and guides and give some results[15].

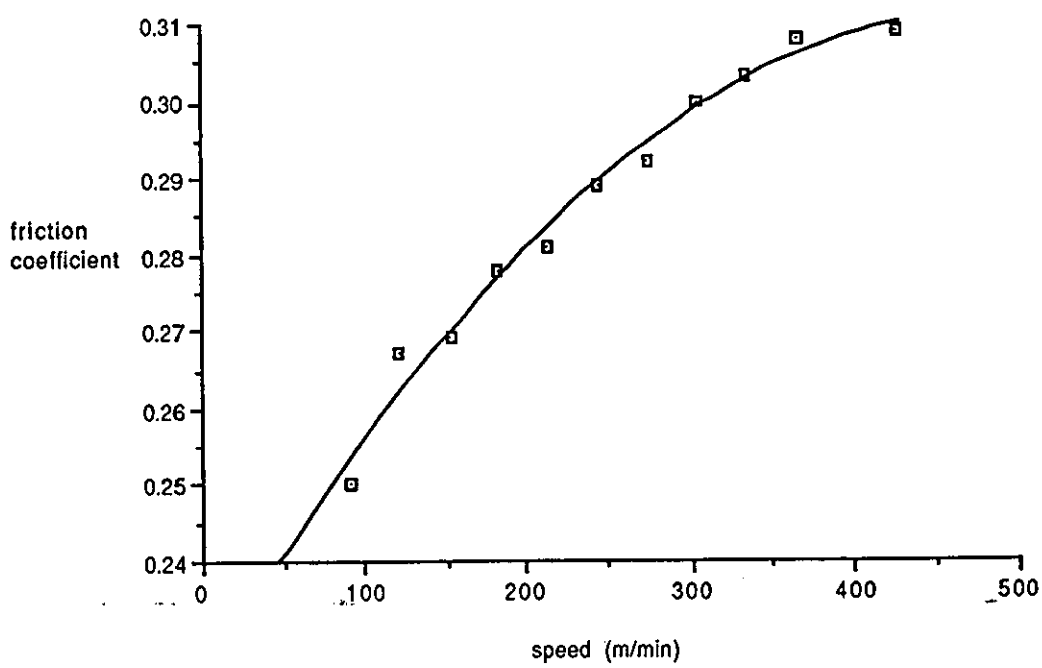


Fig 3.10 Friction Coefficient and Yarn Speed

Fig 3.10 shows the relation between the coefficient of friction and yarn speed. At first, the coefficient rises steadily. When the speed reaches 300 m/min., the coefficient rises more slowly. Increasing the yarn speed from 350 to 450 m/min., further increases the coefficient of friction by only 0.006. In the Masterspinner, the maximum yarn delivery velocity is 400 m/min., and the yarn winding velocity ranges approximately between 360m/min. and 440m/min.. So the variation of friction coefficient at this velocity range does not seem to affect the winding tension variation significantly. The shape of the curve may be explained by reference to the heating effect caused by the passage of the yarn over the guide. If the frictional behaviour at low speeds is governed by partly plastic and partly elastic deformation, then, as the speed rises, with consequent increase of temperature at the region of contact, the ratio of plastic to elastic deformation may increase owing to the elastic property of the materials.

Fig 3.11 shows the relation between friction coefficient and initial yarn tension. It can be seen that the coefficient drops rapidly at first , as initial yarn tension increases up to 7g. After that point, the curve levels off. When the initial yarn tension increases from 7g to 14g, the final friction coefficient decreases by only 0.015.

During cone winding, the average tension is over 20 gram and the tension should not change significantly, if an efficient tension compensator is fitted. Therefore, the frictional force variation should not strongly influnence the winding tension variation during cone winding.

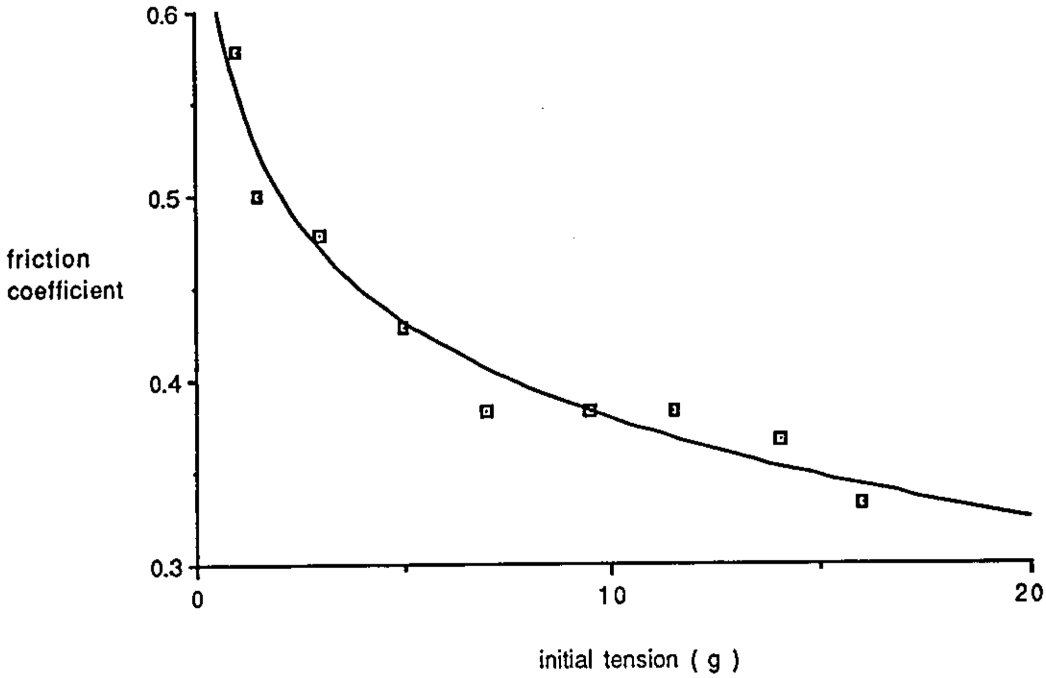


Fig 3.11 Friction Coefficient and Initial Tension

3.5 Conclusion

Through the investigation of tension variation in cone winding, the following conclusion can be drawn:

The amplitude of winding tension variation largely depends on the position where yarn is wound onto the package. This position can be expressed either by cam angular displacement ϕ_c or by traverse guide displacement S. The amplitude of winding tension variation also depends on the package size, that is yarn layer thickness z. The bigger the package size is, the smaller the amplitude is. The inevitable variation of ϕ_c and z during cone winding promotes winding tension variation.

The winding tension is also affected by the yarn path length variation, this variation demands extra yarn which leads to additional yarn tension variation at the both ends of the package.

Compared with tension variation due to winding velocity variation and yarn path length variation, frictional forces have only a very limited effect on yarn winding tension variation. Neglecting the friction variation will therefore not substantially affect the research work on the winding tension variation.

CHAPTER 4 TENSION COMPENSATOR SURVEY AND INVESTIGATION

According to the analysis described in the previous chapter, the yarn winding tension variation is due to two main factors. One is the winding velocity variation. The other is the yarn path length variation. These two factors often mix together and result in irregular tension variation in the yarn. A lot of effort has been made by researchers from universities and companies to design yarn tension compensators. The common purpose is to reduce yarn winding tension variation.

An extensive patent survey has already been carried out within the Mechanical Engineering Department at Loughborough University of Technology.[16] Their contribution helps the research work on this subject greatly. Several literature searches have been conducted by the author at various stages of the work. Initial surveys reveal that patents on the subject date from the middle 1970s and include most patents which have been filed or amended during the subsequent years.

4.1 Tension Compensator Patents

There are a lot of UK, US and German patents on the tension compensator including electrical, electronic, mechanical and pneumatic methods. Here only a few examples are given and classified into different categories.

4.1.1 Spring Tensioner

(1) A patent (UK patent 2026558) relating to a spring compensator is held by Appalachian Electronic Instruments Inc. which claims a tensioning device comprising an electromagnetic coil and core structure alongside a pair of wear surface members. These members in the form of discs are supported in parallel vertical planes. With the yarn running between and engaging the confronting surfaces of the wear surface members, the magnetic attraction forces vary the tension of the yarn when the yarn leaves the yarn tension devices.[17]

(2) UK patent 2024269 claims a tension compensator for use on a package winder. The compensator has two pins, projecting from a rotatable member, which is biased to rotate in a predetermined direction to a stop, so that a strand passed between the pins in a zig-zag fashion moves the pins against the bias to straighten the strand under high tension, and permits the bias to move the pins to store the strand under the lower tension. This compensator is made self-threading.[18]

4.1.2 Pneumatic Tensioner

Phillips Electronic & Associated Industries Ltd. claim a tension device (UK patent 1550424), in which wire is drawn into a nozzle by the flow of air. The thickness of the nozzle is slightly greater than the thickness of the wire. A pressure differential therefore exists between the two sides of the wire loop and this causes the wire to be forced against the side of the nozzle. The frictional effects between the wire and the wall of nozzle also tension the wire.[19]

4.1.3 Mechanical Tensioner

A German company W. Schlafhorst & Co. hold a patent (US patent 4113193) which claims a method of winding conical cross-wound coils at constant thread-feeding velocity, where the varying winding speeds between the largest and the smallest periphery of the coils are compensated by filling and emptying a thread storage device which is controlled by a mechanical device.[20]

4.1.4 Electromagnetic Tensioner

McCoy-Ellison Inc. claims a tension device, in which tension is added to the yarn by an adjustable electromagnetic field controlled by a central unit. The tension can be controlled to the exact degree required and can be adjusted manually at the control unit.[21]

Of the methods described above, some inventions have been applied commercially, but the research on these projects is still proceeding to seek more satisfactory solutions. The following sections describe some methods which the author has considered to reduce the winding tension variation.

4.2 Using a Servo Motor to Keep Constant Winding Velocity

Since the difference between the yarn winding velocity and the yarn delivery velocity contributes a large proportion of winding tension variation, it is suggested that keeping the winding velocity constant may be an ideal solution to the problem. Some papers have already described various mechanical ways of keeping constant winding velocity[22], but here an alternative is given. It is suggested that an electrical servo motor is employed to drive the conical package directly. The servo motor should be so programmed that it drives the conical package at a varying speed. At each winding-on point, the yarn winding velocity should keep constant, therefore minimising the winding tension variation.

Obviously, the electrical motor must have a high rated speed to cope with the high yarn delivery velocity, and also a high rated torque to cope with the high angular acceleration and the large inertia of the package, especially when the package size is big.

Based on the analysis of the winding velocity in Chapter 2, the angular velocity, angular acceleration and driving torque of the package are calculated at different package sizes, the results of which can help to select a suitable servo motor.

4.2.1 The Inertia of the Yarn Package

During cone winding, in the short term, the winding velocity changes significantly from one end to the other. This means a high acceleration is required. In the long term, the package size also increases considerably, which means a large increase in the inertia of the package. Both factors play important roles in the selection of a servo motor.

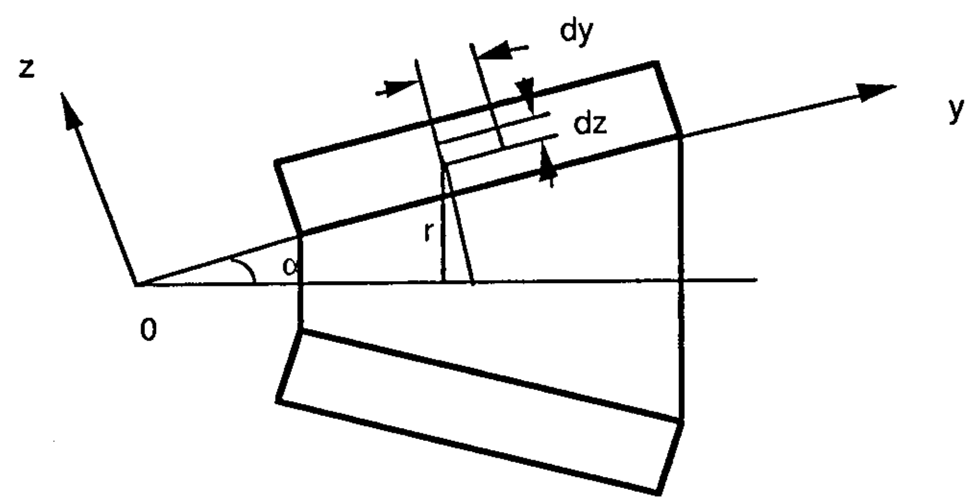


Fig 4.1 Calculation of Yarn Package Inertia

The calculation of the package inertia is given as follows:

$$I_0 = \int r^2 \, dm = \int r^2 \rho \, dv$$

where ρ is the density of the yarn package

r is the radius of rotation

$$I_0 = \int [(z+y^* \tan\alpha)^* \cos\alpha]^2 {}^* \rho \, {}^*dz^* dy \, {}^*(z+y^* \tan\alpha) \, {}^*\cos\alpha \, {}^*d\varnothing$$

$$= \int [(z+y^* \tan\alpha) {}^*\cos\alpha]^3 {}^* \rho \, {}^*dz^* dy \, {}^*d\varnothing$$

$$= \int_0^{2\pi} \int_{y1}^{y2} \int_{z1}^{z2} (z^* \cos\alpha + y \, {}^*\sin\alpha)^3 {}^* \rho \, {}^*dy^* dz {}^* d\varnothing$$

$$= 2\pi\rho \Big[\cos^3\alpha^* \int_{y1}^{y2} \int_{z1}^{z2} z^3 {}^*dz^* dy + 3\cos^2\alpha^* \sin\alpha^* \int_{y1}^{y2} \int_{z1}^{z2} z^2 {}^* y \, {}^*dz \, {}^*dy$$

$$+ 3\cos\alpha^* \sin^2\alpha^* \int_{y1}^{y2} \int_{z1}^{z2} z^* y^2 {}^* dz \, {}^*dy + \sin^3\alpha \int_{y1}^{y2} \int_{z1}^{z2} y^3 {}^* dz^* dy \Big]$$

$$= 2\pi\rho \Big[\cos^3\alpha^* \frac{z_2^4 - z_1^4}{4} {}^* (y_2 - y_1) + 3\cos^2\alpha^* \sin\alpha^* \frac{z_2^3 - z_1^3}{3} {}^* \frac{y_2^2 - y_1^2}{2} +$$

$$3\cos\alpha^* \sin^2\alpha^* \frac{z_2^2 - z_1^2}{2} {}^* \frac{y_2^3 - y_1^3}{3} + \sin^3\alpha^* \frac{y_2^4 - y_1^4}{4} {}^* (z_2 - z_1) \Big] \quad (4.1)$$

4.2.2 The Angular Acceleration and the Driving Torque

If we neglect the variation of guide velocity, in order to keep constant winding velocity, the peripheral velocity should be constant. The only way to do this is to keep changing the package angular velocity continuously. The package angular velocity and acceleration can be easily obtained from what has been described in Chapter 2. Here, only a brief description is given.

(1) Angular velocity of package can be expressed:

$$\omega = \frac{V}{r} \qquad (4.2)$$

where V is the peripheral velocity of the winding-on point

r is the radius of the winding-on point in the package

(2) Angular acceleration of package is:

$$\varepsilon = \frac{\omega_2 - \omega_1}{t_1} \qquad (4.3)$$

where t_1 is the time interval between ω_1 and ω_2

(3) Driving torque:

$$T = \varepsilon * I_0 \qquad (4.4)$$

where I_0 is the inertia of the package

A computer program has been written to calculate the inertia, angular velocity, angular acceleration and driving torque of the package for the conditions where the yarn delivery speed is 500m/min. and 300m/min.

The results of calculation are listed in table 4.1

yarn delivery velocity = 500 m/min.				
package size z mm	inertia of package m^2*kg*10^{-4}	angular velocity 1/s	angular acceleration 1/(s*s)	driving torque N*m
0	0	322	2609	0
10	1.229	233	1310	0.16
20	4.047	182	788	0.31
30	9.463	149	526	0.50
40	18.730	126	376	0.70
50	33.360	110	283	0.94
60	55.010	97	220	1.21
70	85.960	87	176	1.51

yarn delivery velocity = 300 m/min.				
package size z mm	inertia of package m^2*kg*10^{-4}	angular velocity 1/s	angular acceleration 1/(s*s)	driving torque N*m
0	0	194	939	0
10	1.229	140	472	0.058
20	4.047	109	284	0.114
30	9.463	90	189	0.179
40	18.790	76	136	0.253
50	33.360	66	102	0.339
60	55.100	58	79	0.436
70	85.960	52	63	0.545

Table 4.1 Angular Velocity and Driving Torque of Package

4.2.3 Assessment of Constant Winding Velocity Method

From the results listed above, it can be seen that when the yarn delivery velocity is 300m/min.., the highest angular velocity of the package is 194 /s (i.e. 1850 r/min.) and the largest driving torque is 0.545 Nm (i.e. 77 oz.in.). Whereas, when the yarn delivery velocity is 500m/min.., the highest angular velocity of the package is 322 /s (i.e. 3080 r/min.) and the largest driving torque is 1.51 Nm (i.e. 213 oz. in.). Compared with the electrical data (listed in Table4.2) of printed armature servo motors which are commercially available, only the first condition can be met, no printed servo motor can meet the second condition(i.e. v=500 m/min.) at the present time.

In spite of the technical reason mentioned above, the economic consideration of using a servo motor to control each winding head should also be taken into account. For a winding machine which has 144 winding heads, a substantial amount would be added to its total cost, if a servo motor were to fit on each winding head. The machine would also need to be redesigned to make more space for these servo motors, since the package are close together to keep the machine as short as possible.

Although an electrical package driver is more adaptable than a mechanical one, the idea of using a servo motor to keep constant winding velocity during high speed winding cannot yet be put into practice .

PRINTED MOTOR PRODUCT RANGE

Type	Rated Torque oz.in.	Rated Speed rpm	Rated Current A.	Nominal Voltage V.
General Purpose				
GPM9M4-395	11	4000	4.5	15
GPM12M4-347	17	2100	4.5	12
GPM12M4-321	28	3600	5.0	24
GPM16M4-338	54	2500	5.7	28
GPM16M4-340	85	2600	6.0	37
Servo				
G9M2	17	3600	7.1	12
G9M4	25	3650	4.7	24
G9M4T*	20	2700	4.9	18
G9M4H	40	3250	4.6	30
G12M4	55	3650	4.4	48
G12M4T*	66	2270	5.8	30
G12M4H	142	2660	5.9	60
G12M4HA	95	2600	4.3	60
G16M4	140	2700	5.5	60

* Includes integral tachometer.

Type	Rated Torque oz.in.	Rated Speed rpm	Rated Current A.	Nominal Voltage V.
Heavyweights				
MC13	180	3000	7.7	70
MC17N**	160	3000	6.0	83
MC17H**	220	3000	6.5	105
MC17B**	180	3000	24	24
G19M4-P**	450	3000	14.4	83
G19M4-S**	450	3000	7.2	164
G19M4-K**	340	3200	17.0	60
G19M4-B**	450	3000	51.5	24
MC-23S**	1000	3000	14.8	172
MC-24**	1350	3000	25.0	140
MC-27**	2025	3000	33.0	150

** Power outputs may be increased by forced cooling.

Tachometers	Output volts/1000 rpm	Ripple % P-P @ 1000 rpm
G6T	1.0	3
G9T	3.0	5
G12T	6.0	3

Alternative mechanical and electrical specifications are available on request.

Table 4.2 Electrical Data of Servo Motor

4.3 Using a Curved Bar to Keep Constant Yarn Path Length

As mentioned in Chapter 3, the other major factor which affects yarn winding tension variation is the yarn path length variation. According to the calculation in Chapter 3, the maximum yarn path length difference between different winding-on points on the package is 7 mm. This certainly causes yarn tension variation , even if the winding velocity is constant and equal to the delivery velocity. A curved bar which hopefully eliminates this difference is suggested to replace the straight distribution bar which is currently used in the spinning-winding machine.

4.3.1 Design Strategy

The proposed curved bar should have the advantage of keeping yarn path length constant between traverse guide and a fixed guide which could be the delivery roller nip-point.

Fig 4.2 can be used to illustrate the calculation of the profile of the curved bar.

The proposed curved bar is assumed to fit within the plane y=0. A represents the traverse guide which moves back and forth along a straight line *g*. The coordinate of the point A is (a, b_0, c_0), where a is a parameter, which varies from -75 mm to 75 mm, whereas b_0 and c_0 are constant, both b_0 and c_0 depend on the winding machine structure. B is the moving point on the curved bar *f*. The point B moves back and forth corresponding to point A. The coordinate of point B is (x, 0, z) , the point B is confined within plane y=0. C is a fixed guide, the coordinates of point C is (0, e, 0), where e also depends on the structure of the machine.

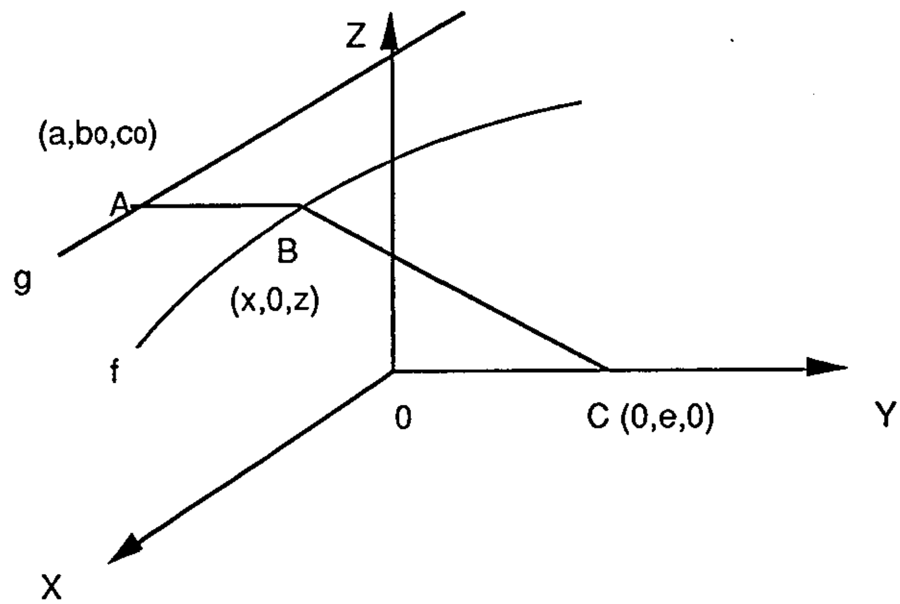


Fig 4.2 Calculation of the Curved Bar Profile

During winding, the yarn passes through the fixed guide C, over the curved bar *f* and then comes to the traverse guide A, where yarn is wound on to the package. The desired curved bar should have a profile such that the length of the yarn in the path AB and BC is constant, no matter where the traverse guide A is. That means:

AB + BC = L= constant.

It has already been proved that the locus of a moving point from which the sum of the distances to the other two fixed points is constant is an ellipsoid. In this case, point A moves along the line *g* periodically, point B moves within plane y=0, point C is a fixed point, and AB+BC=L=constant. For each displacement of point A, there is a ellipsoid corresponding to it, and the point B must lie on the intersection between the ellipsoid and the plane y=0. This intersection is an ellipse. As point A moves along the line *g*, a group of ellipsoids which satisfy the condition AB+BC=L will be generated and the intersections of the ellipsoids and the plane y=0 comprise a family of ellipses. Since B is a moving point on each of the ellipses, then the point B should be on the envelope of this family of ellipses. The locus of point B is part of the envelope.

4.3.2. Envelope Theory

Before the envelope equation is set up, the theory of envelopes is briefly introduced here:

Assume a plane curve is defined in the form:

$$f_c\,(x, y) = 0 \tag{4.5}$$

As the curve moves in a two-dimensional space, it generates a series of curves with the same shape at the different positions. If the position of the moving curve is a function of a single parameter θ, the series of curves can be written in the form:

$$f\,(x, y, \theta) = 0 \tag{4.6}$$

The envelope of the moving curve is itself a curve in the two-dimensional space, which is tangential to the moving curve at all positions. At an arbitrary position defined by θ, the contact point at which the envelope is a tangent to the moving curve must satisfy the equation of the line tangent to the moving curve:

$$\frac{\partial f}{\partial x} dx + \frac{\partial f}{\partial y} dy = 0 \qquad (4.7)$$

Since every point on the envelope is also a point on the moving curve, the tangent point between the envelope and the moving curve automatically satisfies equation(4.6). Therefore, a total differential expression of equation(4.6) vanishes at any point on the envelope:

$$\frac{\partial f}{\partial x} dx + \frac{\partial f}{\partial y} dy + \frac{\partial f}{\partial \theta} d\theta = 0 \qquad (4.8)$$

Combining equations (4.7) and (4.8), and using the fact that $d\theta$ is arbitrary, the following equation is obtained:

$$\frac{\partial f}{\partial \theta} = 0 \qquad (4.9)$$

Hence, the coordinates of a point on the envelope at a specified θ can be determined by equations (4.6) and (4.9).[23]

4.3.3 Envelope Equation

For the curved distribution bar problem, the equation of the family of ellipses is defined in the form:

$$f(a, x, z) = 0 \qquad (4.10)$$

where a is a parameter and $a1 < a < a2$

This time the shape of ellipse changes corresponding to the position of the focus point A, but the nature of the problem does not change.

If the envelope of the equation exists and p(X, Z) is a point on the envelope, then , there exists a 'A' and $f(A, X, Z) = 0$

s.t. $a1 < A < a2$

for other a $f(a, X, Z) >= 0$

s.t. $a1 < a < a2$

the envelope equation can be obtained by minimising the function f(a,x,z). That is:

$$\frac{\partial f(a,x,z)}{\partial a} = 0 \qquad (4.11)$$

subject to $a1 < a < a2$.

The relative function in the form:

$$a = a(x, z) \qquad (4.12)$$

can be obtained from the above equation(4.11).

The envelope equation of the family of ellipses can then be expressed in a form:

$$f(a(x,z)\ x, z) = 0 \qquad (4.13)$$

According to the definition, the equation of an ellipsoid can be expressed as follows:

$$\sqrt{(x-a)^2+(y-b_0)^2+(z-c_0)^2}+\sqrt{(x-0)^2+(y-e)^2+(z-0)^2} = L \qquad (4.14)$$

where a is a parameter

L is the constant path length

Then, the equation of the cross line of the ellipsoids and the plane y=0 is:

$$\sqrt{(x-a)^2+b_0^2+(z-c_0)^2}+\sqrt{x^2+e^2+z^2} = L$$

This is an equation of ellipses and can be rewritten as:

$$(a^2-L^2)*x^2+(c_0^2-L^2)*z^2+2*a*c_0*x*z+a*[3e^2+L^2-(a^2+b_0^2+c_0^2)]*x+c_0*$$
$$[3e^2+L^2-(a^2+b_0^2+c_0^2)]*z+\frac{(a^2+b_0^2+c_0^2+e^2-L^2)}{4}-(a^2+b_0^2+c_0^2)*e^2=0$$

(4.15)

The relative equation between parameter 'a' and variables 'x' and 'z' is:

$$-3xa^2+2a(x^2-c_0z-e^2-\frac{1}{4})+x(2c_0z+3e^2+L^2-b_0^2+c_0^2)=0 \quad (4.16)$$

For different values of guide displacement 'a', the above equation(4.15) represents a family of ellipses. The equation(4.16) indicates the relation among a, x and z. The x and z are confined on the envelope. Because the locus of point B should be continuous and the point B should be on each of the ellipses, the point B must be on the envelope of these ellipses.

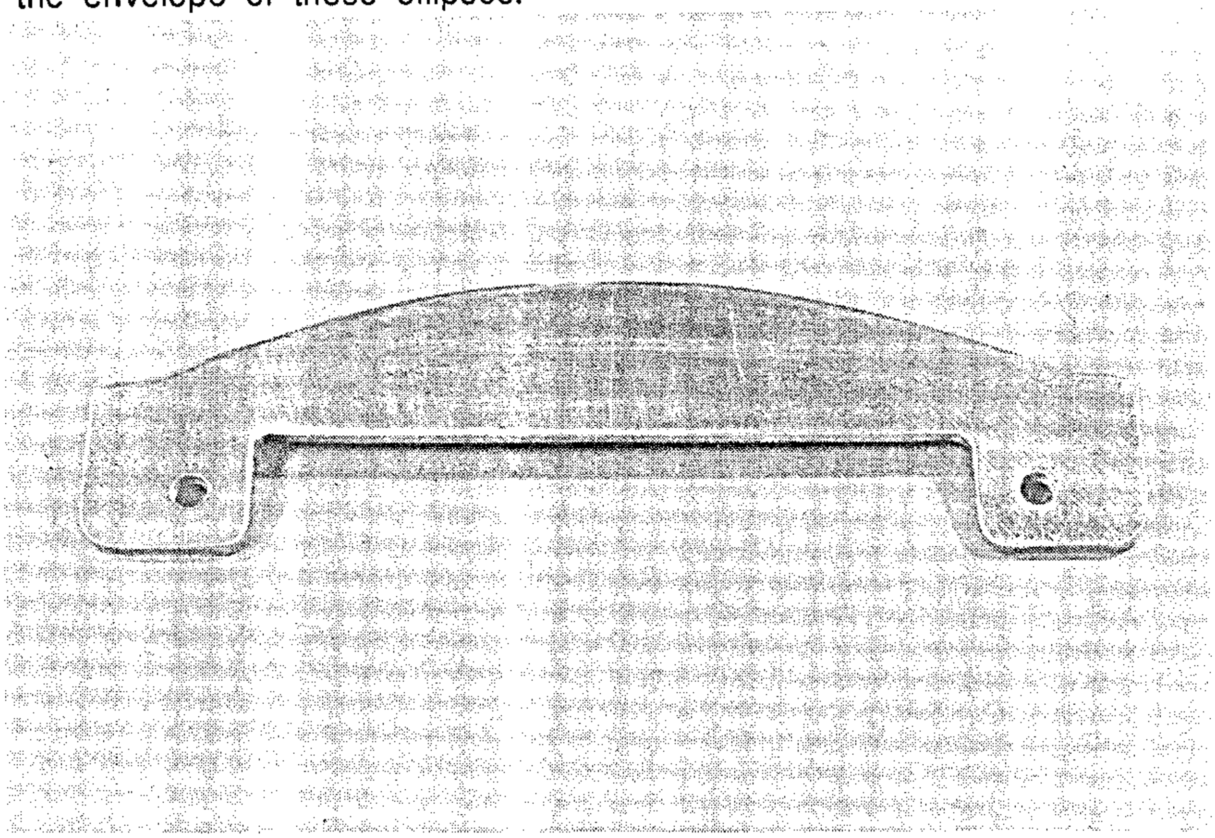


Fig 4.3 The Shape of Curved Bar

4.3.4 Profile of the Curved Bar

It is quite complicated to establish the equation $f(a(x,z), x, z)=0$. However, a computer program has been written employing a numerical method to obtain an approximate solution of the locus of the moving point B, which should be on the envelope of ellipses.

Fig 4.3 shows the shape of the curved bar.

The calculation of curved bar profile are based on the current machine structure. That is b_0=125 mm, c_0=105 mm, e=5 mm.

4.3.5 Assessment of the Constant Path Length Method

A computer program has been written to check the path length variation, where the curved bar is employed. The results show that by using a curved distribution bar, the path length variation is limited to 0.02 mm, which is a significant decrease. Fig 3.9b and Fig 3.9a in previous chapters indicate the experimental results of yarn winding tension variation. The former , in which a curved bar is employed, is much smoother than the latter, where a straight distribution bar is employed. The two peaks (in Fig3.9a) of tension curve which correspond to the winding positions at the two ends of the package, where yarn path length is longer, disappear in Fig3.9b, so using a curved bar instead of the straight distribution bar can reduce the yarn winding tension variation.

4.4 Conclusion

A lot of patents regarding the yarn tension compensator have been claimed, and a lot of literature has been published. As a result, some innovations have come into the commercial market. Generally speaking, some yarn tension compensators are quite adequate to a degree, especially when the yarn winding speed is low. But the tendency of industry is to promote high productivity, and one of the results is to increase machine speed. Therefore, some compensators become inadequate and need to be redesigned to cope with the ever increasing demands.

Two alternative methods for reducing winding tension variation have been investigated in this chapter. One is using a sevo motor to keep winding velocity constant. This method seems less attractive, not only because of the lack of a suitable servo motor, but also the potential high cost. The other is using a curved distribution bar to keep constant yarn path length. this has been proved a simple, reliable way to reduce tension variation which is due to the path length variation.

The yarn tension compensators listed in this section can be divided into two categories according to their different working principles.

(1) passive compensator

This kind of compensator, such as spring compensator, is forced to work by the varying yarn tension and normally has difficulty in coping with high speed winding.

(2) positive compensator

This kind of compensator, such as a mechanical or a electronic compensator, works actively and its response increases as the machine speed increases.

That is why some leading textile machine companies prefer a positive compensator.

According to the winding velocity analysis in chapter 2 and the yarn winding tension analysis in Chapter 3, the short term winding tension variation corresponds with the traverse guide position, the long term winding tension variation with the package size. In order to reduce the winding tension variation , any further investigation of the yarn winding compensator should consider both factors.

Since a mechanical device can be made simple and reliable, and an electronic device can be flexible and adaptable, both devices will be investigated to work as winding tension compensators.

CHAPTER 5 SYSTEMATIC GENERATION OF COMPENSATION MECHANISM

5.1 The Criteria of the Compensator Design

In the spinning-winding machine, if a curved bar is fitted to eliminate the variation in the yarn path length between the traverse guide and the delivery rollers, then the main cause of variation in the yarn winding tension is the variation in the yarn winding velocity. That has been proved by both the mathematical analysis and the yarn tension experiments. As is stated in the preceding chapters, there are two main factors which affect the yarn winding velocity. One is the guide velocity, which depends on guide position (if the yarn delivery velocity is set); the other is the package peripheral velocity which depends on the winding-on position and the package size. In brief, the yarn winding velocity variation is mainly due to the variation of the guide displacement and the package size.

The variation in the yarn winding velocity leads either to a yarn surplus if winding velocity is lower than the delivery velocity, or to a yarn shortage if the winding velocity is higher. Both contribute to the yarn tension variation.

During winding the variation of the guide displacement and the increasing of package size is inevitable; a mechanism is suggested to fit between the delivery rollers and the curved bar to work as a tension compensator. When the yarn is in surplus, the mechanism takes up some yarn. When the yarn is in shortage, the mechanism releases some yarn. By delicately adjusting the yarn supply, to some extent, the mechanical compensator can considerably minimise the variation in yarn winding error, and therefore reduce the variation in yarn winding tension.

The aim of this chapter is to search for feasible mechanisms which can be used as a desired mechanical yarn compensator. The first step to achieve this aim is the kinematic synthesis of the compensation mechanism.

5.2 Introduction and Specification

Kinematic synthesis has been defined as a combination of type synthesis, number synthesis and dimensional synthesis.[24]

Type synthesis refers to the kind of mechanism selected; it might be a linkage, a gear system, belt and pulley or a cam system. For this case, considering the manufacturing processes, safety, space, reliability and economics, a linkage is selected as the compensation mechanism.

Number synthesis deals with the number of links and the number of joints or pairs that are required to obtain a certain mobility. Number synthesis will be applied in this chapter to search for feasible mechanisms.

Dimensional synthesis is the step in design to determine the dimensions of the individual links. This step will be carried out in the next chapter by using optimisation technology.

It would be helpful to undertake design methodologies first so that all feasible mechanisms can be generated systematically. There are many publications describing work with this aim.[25~28] Among them, Freudenstein and Malci developed a method to separate kinematic structures from functional considerations for the creation of mechanisms.[29] They enumerated the kinematic structures in an essentially systematic way according to the degree of freedom, number of moving members, complexity and nature of motion (planar or spatial). Each enumerated structure can then be sketched and evaluated with respect to the requirements of the mechanism. Potentially acceptable mechanisms can then be evaluated in depth until a final design is achieved.

Based on the design requirement listed in the last section and the machine arrangement, some restrictions and evaluation criteria for enumerating the feasible compensator mechanisms can be listed. The specifications which are going to affect the choice between one mechanism structure and the other are listed under two headings. There are some requirements which must be met, and these are called mandatory characteristics. There are others that are desirable, but probably cannot be attained exactly, and these are called desirable characteristics. These desirable characteristics can be used to evaluate alternatives. Such lists aid the elimination of the unsatisfactory alternatives. Generally speaking, the more items on the lists, the better the result that can be obtained. Otherwise the enumeration could be ended with a lengthy number of alternatives to select from, but without any criteria for choice. Considering the compensation mechanism requirements, the following specifications are listed.

mandatory characteristics:

1. The mechanism is limited to two degrees of freedom.
2. All joints are turning pairs or straight-line sliding pairs.
3. The input elements are adjacent to the frame.

Desirable characteristics:

1. The number of elements should be as few as possible due to limited machine fitting space.
2. The mechanism should be able to be analysed easily.
3. The transmission angle should be within the acceptable limitation.
4. The number of sliding pairs should be as few as possible due to the difficulty of lubrication.
5. The inertia force and moment of the mechanism should be small.

According to the mandatory characteristics, those planar mechanisms which have only two degrees of freedom are considered. Obviously the two inputs of the mechanism should correspond to the traverse guide displacement and the package size, whereas the single output should correspond to the requirement of yarn supply. In order to use the existing spinning-winding machine parts as input elements, the input parts should be adjacent to the frame. In addition, only those mechanisms which obey the general degree of freedom equation are considered.

5.3 Kinematic Chain Graphs

The abstract representation of kinematic chains has long been investigated with the aid of graph theory. This is a powerful method for the creation of mechanisms in a relatively simple, semi-systematic way.

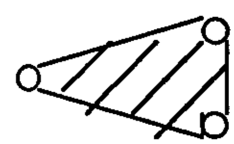
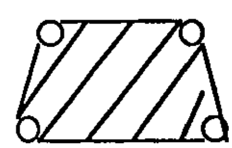
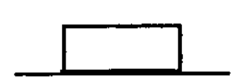
Name	Definition	Graph
binary	a link with two low pairs	
ternary	a link with three low pairs	
quaternary	a link with four low pairs	
pin joint	a turning pair between two links	
sliding joint	a prismatic pair between two links	

Table 5.1 Definition and Graph of Kinematic Symbols

A graph of a kinematic chain is defined as such a graph in which the links correspond to lines, the joints correspond to circles and the joint-connection of links to circles-connection of lines. For example, a ternary link (i.e. a link with three pairs) can be represented by a triangle, its three pair elements being represented by three circles. The sketching of this graph of a kinematic chain is straightforward. The inverse, the sketching of a mechanism from its graph, is not difficult but requires some practice.

The most frequently used symbols in mechanisms are listed in Table 5.1.

5.4 Associated Linkage Concept

The methods used to creatively discover suitable types of linkages are based on the concept of associated linkages. The associated linkage concept was developed by R. C. Johnson and K. Towligh,[30] and consists of the following procedure.

1. The determination of rules that must be satisfied for the selection of a suitable "associated linkage".

2. The application of suitable associated linkages to the synthesis of different types of devices.

These concepts are used to generate the suitable associated linkages for the desired mechanical yarn compensator.

Firstly, a planar mechanism must obey the equation of the general degree of freedom. That is:

$$F=3(n-1)-2P_L \qquad (5.1)$$

where F= the degree of freedom of the linkage

n= number of the links

P_L= number of low pair joints

Secondly, the mechanism is normally composed of binary, ternary, quaternary, etc. That is:

$$B+T+Q=n \qquad (5.2)$$

where B= number of binary in the linkage

T= number of ternary in the linkage

Q= number of quaternary in the linkage

Thirdly, the total low-pair-element number in the mechanism is twice the number of low pair numbers. That is:

$$2B+3T+4Q=2P_L \qquad (5.3)$$

This is because a binary has two low pair elements, a ternary has three, a quaternary has four. So every feasible associated linkage should follow the above three equations.

5.5 Generation of a Feasible Compensation Mechanism

In order to generate compensation mechanisms systematically, Hall's theory is applied. He suggested a kind of modelling which might lead to a systematic exploration of different types of mechanisms.[31]

According to the mandatory requirement, only planar mechanisms which have two degrees of freedom are considered.

By substituting F=2 into equation (5.1), and then substituting the new equation into equation (5.3), we get :

$$2B+3T+4Q=3n-5 \quad (5.4)$$

Combining equation (5.2) and (5.4), a group of equations can be obtained. That is:

$$B+T+Q=n$$

$$2B+3T+4Q=3n-5 \quad (5.5)$$

Equation (5.5) can be used to decide the number and the type of the elements of the kinematic chain. The kinematic chain can then be converted into a mechanism which has two degrees of freedom, if one of the elements is fixed.

Before using equation (5.5) to evaluate all kinds of possible mechanisms, the following conditions must be met.

1. B, T, Q and n must be integral.
2. n must be greater than three.
3. n must be an odd number.

This is because the total number of kinematic pair elements should be an even number, therefore, according to equation (5.4), n must be an odd number.

5.5.1 Associated Linkage Assortment

An assortment of links consists of a specific number of binary, ternary, quaternary, etc. In equation (5.5), by choosing different n, the different assortments of the mechanism can be obtained. They are given in the Table 5.2.

Assortment No.	n	B	T	Q	joint number
1	5	5	0	0	5
2	7	6	0	1	8
3	7	5	2	0	8
4	9	7	0	2	11
5	9	6	2	1	11
6	9	5	4	0	11

Table 5.2 Assortment of the Associated Linkage

For each of the above assortments, several combinations can be derived. Considering the limited space available in the machine, those mechanisms which have more than seven members are considered too cumbersome. In the next section, only the mechanisms which have five members and seven members will be considered.

5.5.2 Graphs of Associated Linkages

Based on the assortment of the mechanism listed in Table 5.2, the graph of the associated linkages can be drawn. It can start with the least complicated associated linkage chain since simplicity is an obvious design advantage. A particular assortment may be assembled in more than one way. The results are listed in the Table 5.3.

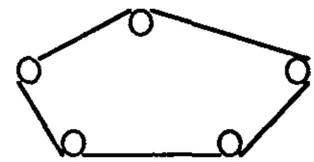
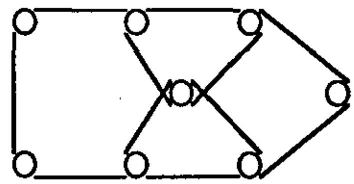
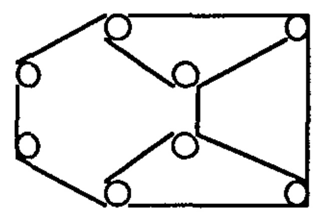
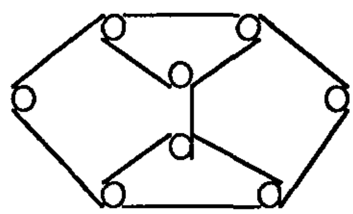
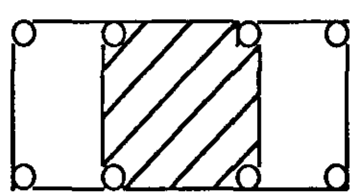
Assortment No.	Linkage Structure	Graph
1	B=5 T=0 Q=0	
2	B=5 T=2 Q=0	
3	B=5 T=2 Q=0	
4	B=5 T=2 Q=0	
5	B=6 T=0 Q=1	

Table 5.3 Associated Linkages

5.5.3 Assessment of the Associated Linkages

The first mandatory characteristic of the desired compensation mechanism is the degree of freedom of the linkage. However, the output motion of the mechanism which is of primery concern to the designer also largely depends on what kind of the degree of freedom the mechanism has.

An output motion is affected by both input parts of the mechanism, only if the mechanism has a "total degree of freedom". Hain has stated that a kinematic chain can be said to have a total degree of freedom when it can fulfil the following condition: " If we select any number of the chain to be the frame of a mechanism, and choose any F member of it to be the independent driving elements (or motor elements) then the position of every remaining member (the driven members) is dependent on the position of all motor elements."[32]

A chain which cannot fulfil the above condition may possess a partial degree of freedom F. In all or some of the mechanisms derived from such a chain, the movement of certain of the driven members depends on the movements of only a number F_p of the motor elements, where $F_p < F$.

Alternatively, a chain which does not satisfy this condition for a total degree of freedom will be fractionated. Such a chain possesses at least one link (which must have at least four joints) which may be cut into two or more parts, with the result that, of the severed sub-chains, each forms a closed kinematic chain.

In this section, different assortments are assessed respectively according to their degree of freedom.

Assortment 1

This is a typical five-bar linkage and has a total degree of freedom. Any link can be chosen as a frame. Both links which are adjacent to the frame can be used as input elements. Either of the other two elements can be output link.

Assortment 2 and 3

Each kinematic chain contains a four-bar loop . If one of these four members is chosen as frame and a second one as motor element, the movements of the other two members of this sub-chain are dependent only on this input member. Neither of them could be made the second motor element. That means for each of the linkages, two input links must be arranged at the different loop. If one of the four-bar loop members is chosen as output link, then the linkage has a partial degree of freedom.

Assortment 4

This kinematic chain has two five-bar loops, so it has a total degree of freedom. If a ternary is fixed as frame, any two of the links which are adjacent to the frame can be used as input links. Either coupler or rocker can be used as an output link.

Assortment 5

This kinematic chain has two independent loops, If quaternary is chosen as frame, and if two input elements sit in the same loop, then the mechanism cannot work properly. If two input elements sit in different loops, the mechanism has fractionated degree of freedom. Obviously, this assortment cannot meet the need of the desired compensation mechanism.

Based on the above assessment, the associated linkage 1, 2, 3 and 4 can meet the mandatory requirements, so, they are feasible compensation mechanisms. However, they still need further evaluation and comparison to select the most feasible one.

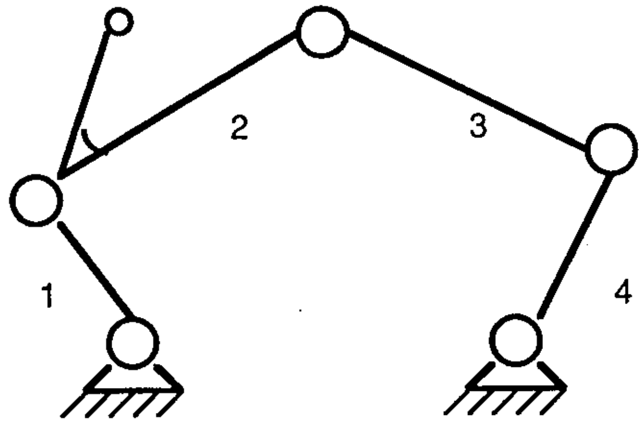
5.6 Evaluation and Recommendations

In general, evaluation of mechanisms is to decide which one contains more desirable characteristics. Evaluation of the feasible mechanisms can supply their priority orders for further studies. Based on the desired characteristics listed in the section 5.2 , four feasible mechanisms are evaluated.

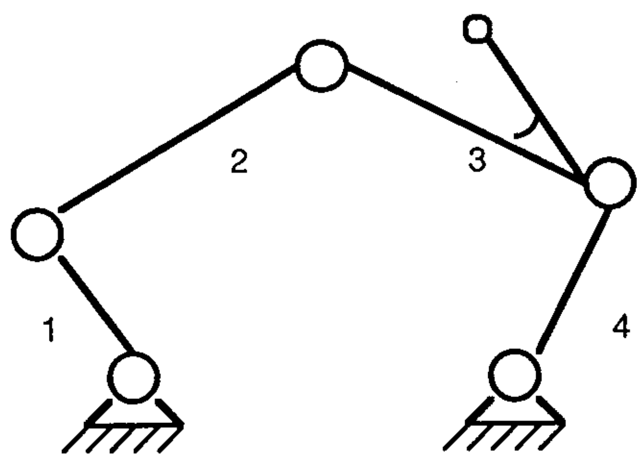
Fig 5.1-5.3 show the feasible linkages derived from the associated linkages listed in Table 5.3.

For the first desirable characteristic, linkage 1 has only five links, two links less than the other three, so it should be given a higher score than the others.

Considering the second desirable characteristic, the linkages 1, 2, 3 can be dismounted to input links and dyads. This kind of mechanism is called a grade two mechanism, and is easy to analyse mathematically. Since linkage 1 only has one dyad, one dyad less than the other two linkages, it is the simplest to analyze. For linkage 4, if the input links are given as shown in the linkage 4(a) (the arrow represents the input), it is also a grade two mechanism and it has two dyads. If the input links are given as shown in the graph 4(b), then linkage 4 is a grade three mechanism, and circular point theory should be applied to follow the movement of the ternary. This mechanism is complicated and the analysis of the kinematic characteristics of the linkage is rather difficult.

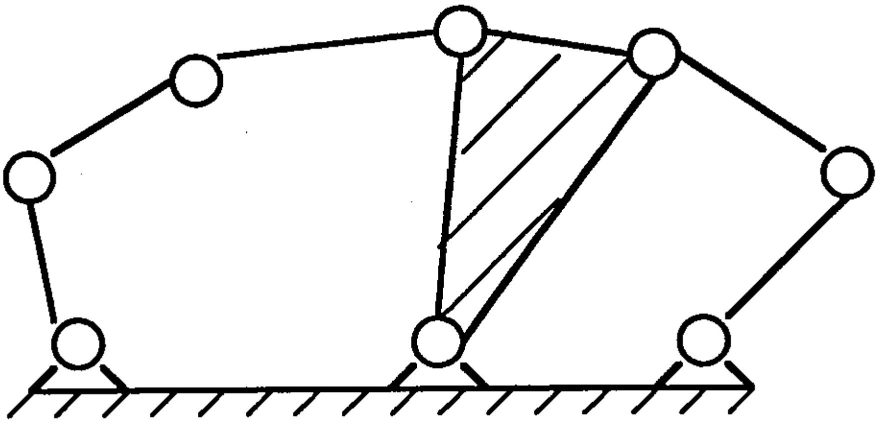


Linkage 1 (a)

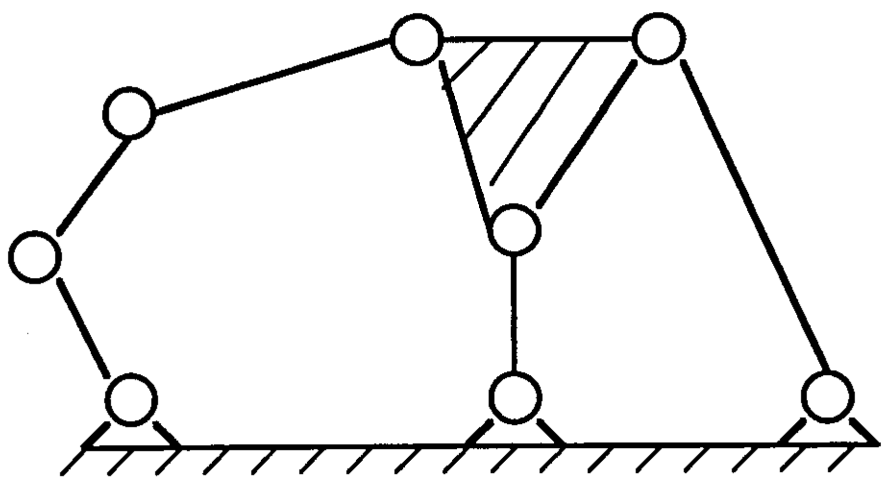


Linkage 1 (b)

Fig 5.1 Linkage 1 (five-bar linkage)

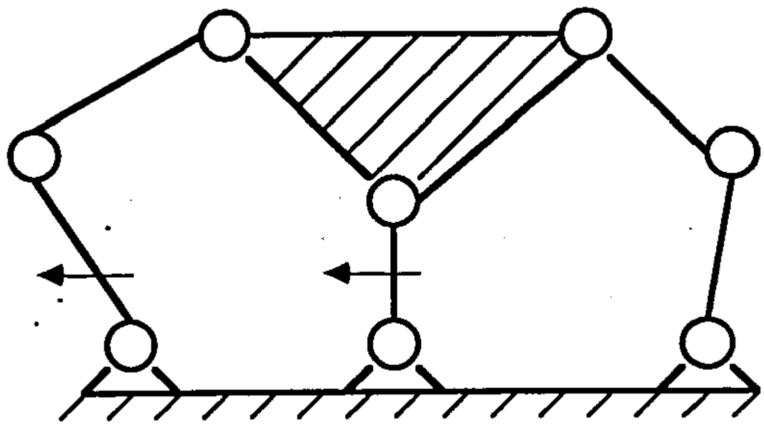


Linkage 2

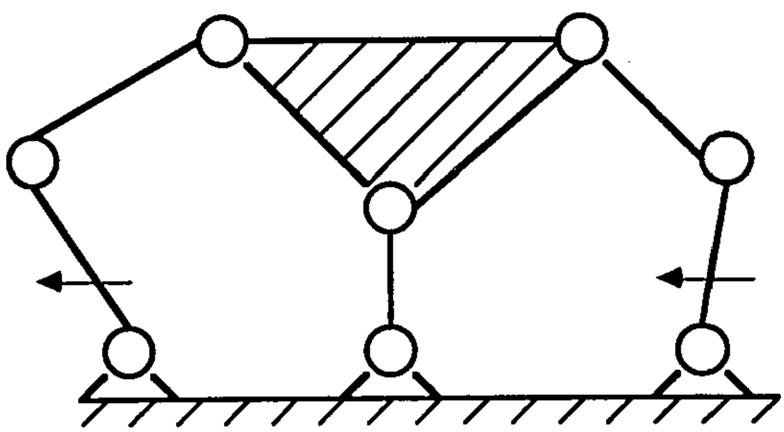


Linkage 3

Fig 5.2 Linkage 2 and Linkage 3 (seven-bar linkage I)



Linkage 4 (a)



Linkage 4 (b)

Fig 5.3 Linkage 4 (seven-bar linkage II)

As for the third desirable characteristic, because the transmission angle not only depends on the mechanical structures of linkages but also on the link dimensions, it is quite difficult to prejudge which feasible mechanism gives a preferable transmission angle. For this reason, none of them has priority for the third desirable characteristic.

For the fourth desirable characteristic, all joints in these mechanisms are turning pairs, but in linkage 1, the number of turning pairs is less than that of the other linkages.

For the last desirable characteristic, it is difficult to predict the inertia force and the inertia moment of these linkages at this stage, but it is obvious that all links in the five bar linkage are binaries, the binary is simpler than other link structures and normally can be made lighter than the ternary or quaternary. This possibly leads to less inertia force and inertia moment in a high speed winding machine.

According to the evaluation given in the previous paragraph, linkage 1 has the highest priority, therefore , linkage 1 is recommended as the mechanical yarn compensator

In linkage 1, even If link 1 and link 4 are chosen as input links, there are still two alternative linkages. Fig 5.1 gives two alternative arrangements of the linkage 1. One arrangement has link 2 as output link, the other has link 3 as output link, but at this stage, it is hard to say which is more suitable for the job. The comparison of the two linkages will be given in a later chapter.

In the linkage compensator, link 1 can be used as an input crank to provide a main motion which corresponds to the traverse guide displacement. Link 4 can be used as a rocker to input the control motion which corresponds to the package size. However, either link 2 or link 3 can be used as a coupler to output a desired coupler curve.

The working principle of the mechanical compensator is that the compensation mechanism drives the coupler point, where a yarn roller is fitted, to precisely adjust the amount of yarn held in the compensator according to the analysis of yarn winding error. In this way, the mechanical compensator changes the yarn supply rate and reduces the difference between yarn demand and supply during cone winding.

CHAPTER 6 KINEMATIC ANALYSIS OF MECHANICAL COMPENSATOR

6.1 General Layout of the Compensation Mechanism

In the previous chapter, some feasible compensation mechanisms have been worked out. By enumeration, evaluation and comparison, a five-bar linkage is finally selected as the mechanical yarn compensator.

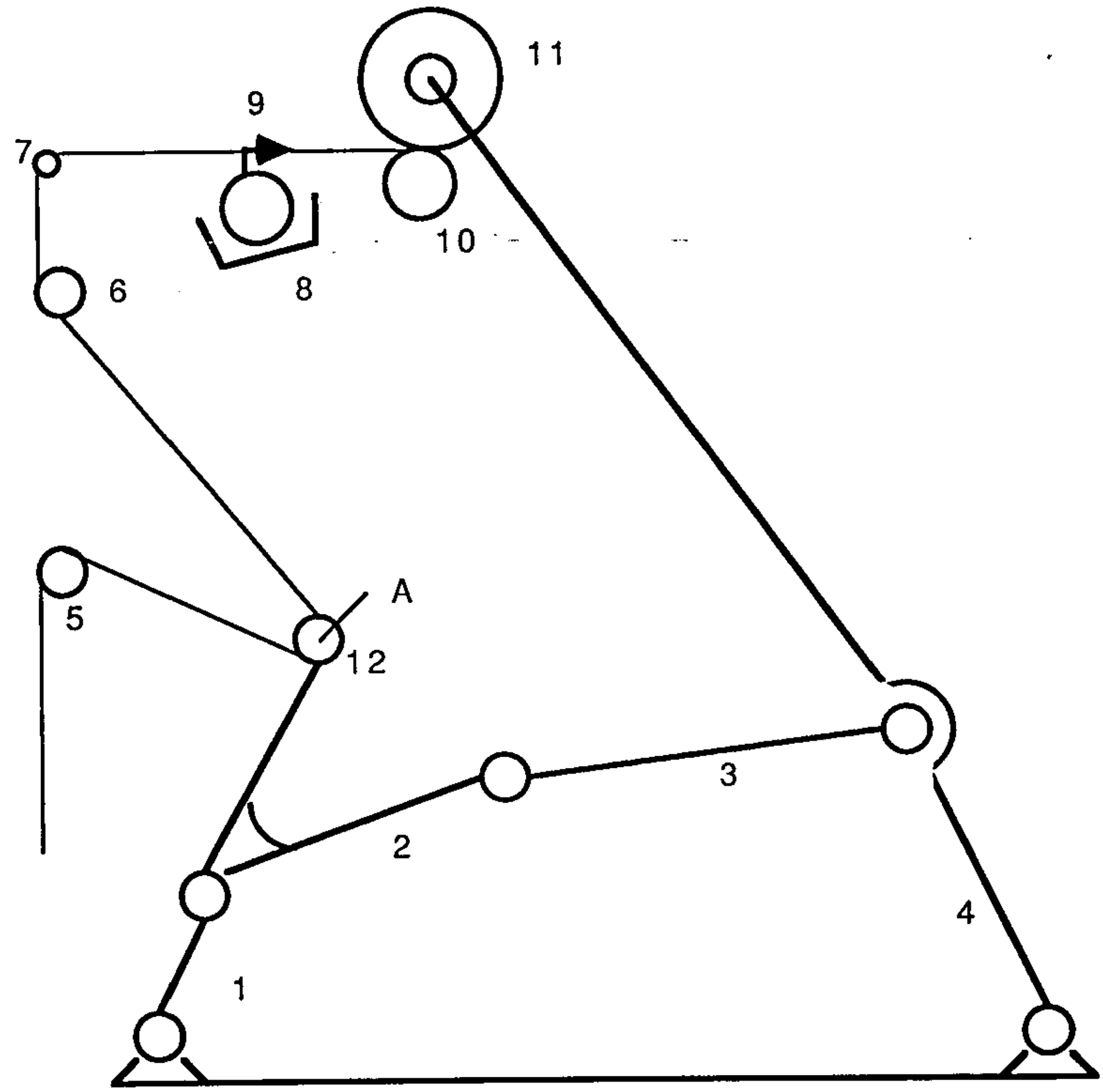
Fig 6.1 gives a layout of the mechanical compensator.

In the yarn compensator, link 1 is a crank. It is driven by the cam shaft through a pair of gears. Because a complete cycle of the guide motion corresponds thirteen turns of the cam shaft, and a complete cycle of guide motion includes two to-and-fro guide motions, the ratio of angular velocity of the cam shaft to the crank is 6.5. So one revolution of the crank corresponds to one back and forth motion of the traverse guide .

Link 2 is a coupler on which a roller is fitted. The centre of the roller is coupler point A. The roller is used to take up or release yarn during cone winding.

Link 3 is a connecting link which connects link 2 and 4. Link 3 can be used as a coupler. In that case, the link 2 becomes connecting link.

Link 4 is a rocker which is also a part of the package holder. During winding, as the package size grows up, its angular displacement decreases as well. That is the input of the rocker.



1. crank
2. coupler
3. connecting link
4. rocker (packager holder)
5. bottom guide
6. top guide
7. distribution bar
8. cam
9. traverse guide
10. driving drum
11. yarn package
12. roller

Fig. 6.1 Mechanical Tension Compensator

The working procedure of the mechanical compensator is as follows:

When the yarn comes out from the delivery shaft, the yarn passes over the fixed bottom guide 5, then, goes round a roller which is fitted on the link 2, the centre of the roller is the coupler point A, and then, the yarn passes over the fixed top guide 6. After that, the yarn is distributed by the traverse guide 8 onto the rotating package 10. The amount of yarn held in the compensator depends on the position of coupler point A. Changing the position of coupler point A can also change the amount of yarn held between the bottom guide 5 and the top guide 6, consequently changing the yarn supply to the winding unit.

According to the calculation of the yarn winding error, the yarn compensator should work in such a way that, when yarn is in surplus, it takes the extra amount of yarn, and when the yarn is in shortage, it releases the same amount of yarn, so as to keep the balance of yarn demand and supply.

Theoretically, all these can be done by planning the motion of coupler point A carefully. The motion of the coupler point A depends on the motion of input links and the dimension of the linkage. Since the crank is driven by the cam shaft through a pair of gears, if the pattern breaking is neglected, it can be assumed that the crank rotates at a constant angular velocity. The crank angular displacement corresponds to the traverse guide displacement. The angular displacement of rocker 4 depends on the building of the yarn package. Since the size of package grows slowly, it affects the motion of coupler point A only in a long run. By deliberately adjusting the dimensions of the links, the motion of coupler point A can be used to reduce, if not eliminate, the difference between the yarn supply and demand, therefore, to reduce the variation in the winding tension.

6.2 Relative Position Analysis of Five-bar Linkage

6.2.1 Introduction to the Kinematic Analysis

Kinematics is the study of the relative motion between two or more rigid bodies. Each body is considered to include all points that move with a set of coordinate axes fixed in the body, and any points in space may be considered as momentarily attached to one of the systems of a rigid body. Kinematic analysis of mechanisms normally starts with a kinematic chain which is the assembly of link/pair combinations to form one or more loops. A kinematic chain is considered as a mechanism if one of its links is fixed as frame and if its degree of freedom equals the number of the input elements in the kinematic chain.

Kinematic analysis includes three stages: position analysis, velocity analysis, and acceleration analysis. Of the three stages, position analysis is the most complex one, because it requires the solution of nonlinear equations. Once the solution of position analysis is obtained, velocity analysis and acceleration analysis can be solved by differentiating position equations with respect to time. The expression of velocity and acceleration then can be obtained.

Many plane mechanisms can be assembled from one or more of the basic combinations of links and pairs, such as two-link dyad, oscillating slider, rotating guide etc. If a computer programme of a complete position-velocity-acceleration analysis for each of the basic combinations is made, then, the kinematic analysis of the mechanism is straightforward with the help of a properly prepared set of computer subroutines.

A lot of publications have discussed the kinematic analysis of mechanisms. Burton, Hartenberg, Suh and Radcliffe have described various methods to analyse the motion of mechanisms graphically, analytically or in some cases, numerically.[33~35]

In order to simplify the mathematical modelling for the kinematic analysis of a mechanism, some assumptions are made:

1. All moving parts are rigid bodies.
2. No clearance exists between joint elements.
3. Any deviation in the spherical geometry of the elements is negligible.

In the next section, some basic concepts about mechanisms is introduced to help the motion analysis of the mechanical compensator.

6.2.2 Rigid Body Motion

In kinematic analysis, all parts of a mechanism are considered as rigid body. That means there can be no relative motion between two arbitrarily chosen points on the same body. It is often convenient to describe the geometrical shape of a rigid body as if it were dimensioned as an engineering drawing .

Fig 6.2 shows a rigid body, it is described by parameters: r, s, ø.

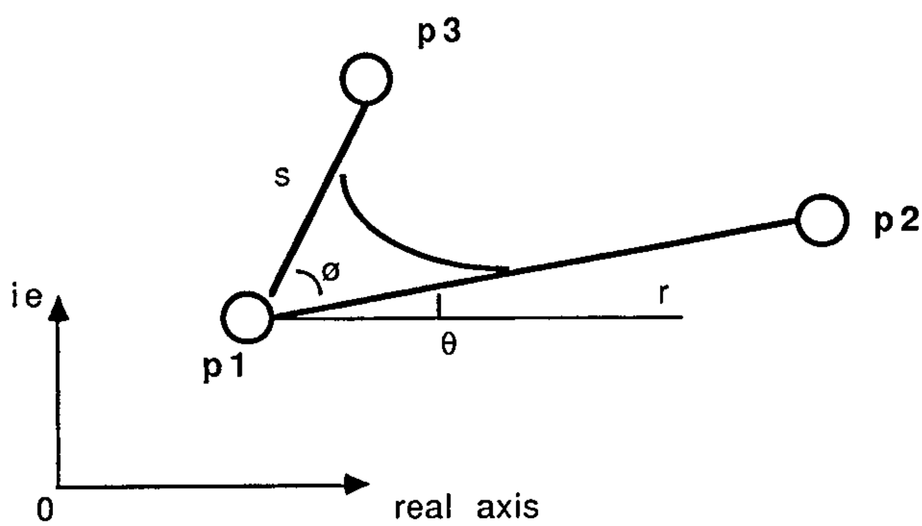


Fig 6.2 Rigid Body

Once the actual position angle θ is known along with the position of the reference point p_1, the position of any other point such as p_3 can be calculated.

$$p_3 = p_1 + (p_3 - p_1) = p_1 + s * e^{i(ø + \theta)} \quad (6.1)$$

$$p_{3x} = p_{1x} + s * \cos(ø + \theta) \quad (6.2)$$

$$p_{3y} = p_{1y} + s * \sin(ø + \theta) \quad (6.3)$$

By using a rotation matrix, the velocity and acceleration of any point of interest can be described in terms of the known motion of the reference point.

$$\dot{p}_3 = \dot{p}_1 + \dot{\theta} \times (p_3 - p_1) = \dot{p}_1 + \dot{\theta} * s * ie^{i(\phi+\theta)} \tag{6.4}$$

$$\begin{aligned} \ddot{p}_3 &= \ddot{p}_1 + \ddot{\theta} \times (p_3 - p_1) + \dot{\theta} \times (\dot{\theta} \times (p_3 - p_1)) \\ &= \ddot{p}_1 + \ddot{\theta} * s * ie^{i(\phi+\theta)} - \dot{\theta}^2 * s * e^{i(\phi+\theta)} \end{aligned} \tag{6.5}$$

6.2.3 Two-link Dyad Motion

The two-link dyad is the most common combinations of link/pair. Fig 6.3 shows a two-link dyad. The analysis of the two-link dyad must account for the possibility that either or both reference point 1 and 2 may be in motion.

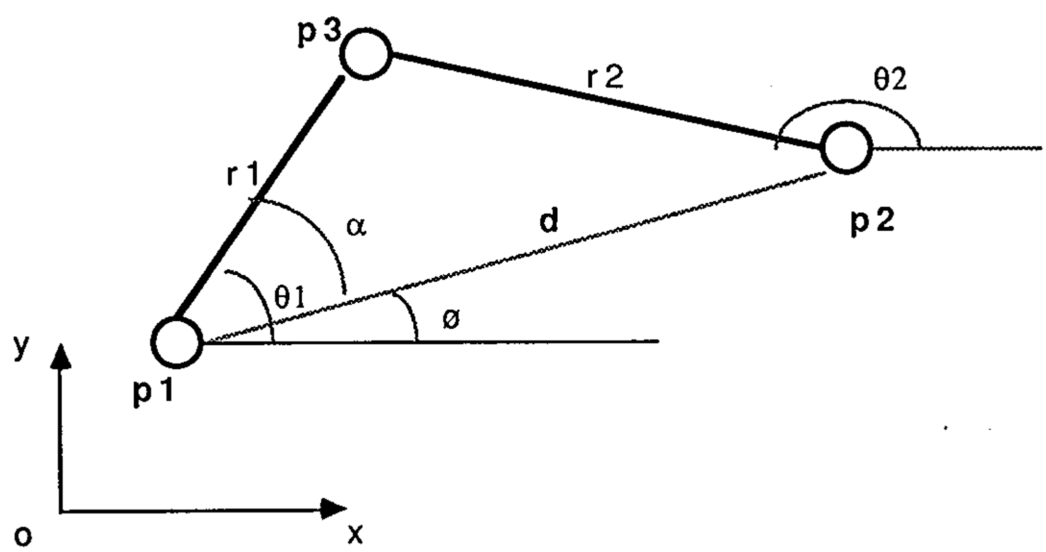


Fig. 6.3 Two-Link Dyad

The position analysis is accomplished in the following steps:

1. Calculate the distance d between reference points p1 and p2 from

$$d^2 = (p_{2x} - p_{1x})^2 + (p_{2y} - p_{1y})^2 \qquad (6.6)$$

2. Check mobility. If

$$d > (r_1 + r_2) \qquad \text{or} \qquad d < | (r_1 - r_2) |$$

the dyad cannot be assembled.

3. Calculate the ø, the angle for vector **d**.

$$ø = \tan^{-1}\left(\frac{p_{2y} - p_{1y}}{p_{2x} - p_{1x}} \right) \qquad (6.7)$$

4. Calculate the α, the angle between **r1** and **d**.

$$\alpha = \cos^{-1}\left(\frac{r_1^2 + d^2 - r_2^2}{2r_1 d} \right) \qquad (6.8)$$

5. The sign of α is determined from the specified mode of assembly in the first position. With continuous mobility the mode of assembly does not change as a result of a change in position.

$$\theta_1 = ø \pm \alpha \qquad (6.9)$$

6. The coordinates of the point of interest p3 becomes

$$p_{3x} = p_{1x} + r_1 \cos \theta_1$$
$$p_{3y} = p_{1y} + r_1 \sin \theta_1 \qquad (6.10)$$

7. Finally,

$$\theta_2 = \tan^{-1}\left(\frac{p_{3y} - p_{2y}}{p_{3x} - p_{2x}}\right) \tag{6.11}$$

A computer subroutine has been made to analyse the moving position of the two-link dyad.

If the position and the velocity of the reference points are known. The velocity analysis of the point of interest can be made on the bases of the equivalence of the two expressions.

$$\dot{p}_3 = \dot{p}_1 + \dot{\theta}_1 \times (p_3 - p_1) = \dot{p}_2 + \dot{\theta}_2 \times (p_3 - p_2) \tag{6.12}$$

The acceleration analysis also can be made in a similar way.

$$\begin{aligned} \ddot{p}_3 &= \ddot{p}_1 + \ddot{\theta}_1 \times (p_3 - p_1) + \dot{\theta}_1 \times (\dot{\theta}_1 \times (p_3 - p_1)) \\ &= \ddot{p}_2 + \ddot{\theta}_2 \times (p_3 - p_2) + \dot{\theta}_2 \times (\dot{\theta}_2 \times (p_3 - p_2)) \end{aligned} \tag{6.13}$$

6.2.4 The Five-bar Linkage Position Analysis

In order to analyse the motion of the five-bar linkage, especially to trace the position of coupler point A, the five-bar linkage can be disassembled into two input links and one dyad. As is shown in Fig 6.4.

The input link 1, whose position is defined by the specified angle θ1 and whose angular velocity is proportional to the cam shaft angular velocity, is a rigid body. According to the analysis of the rigid body motion, then, the motion of point p2 is known.

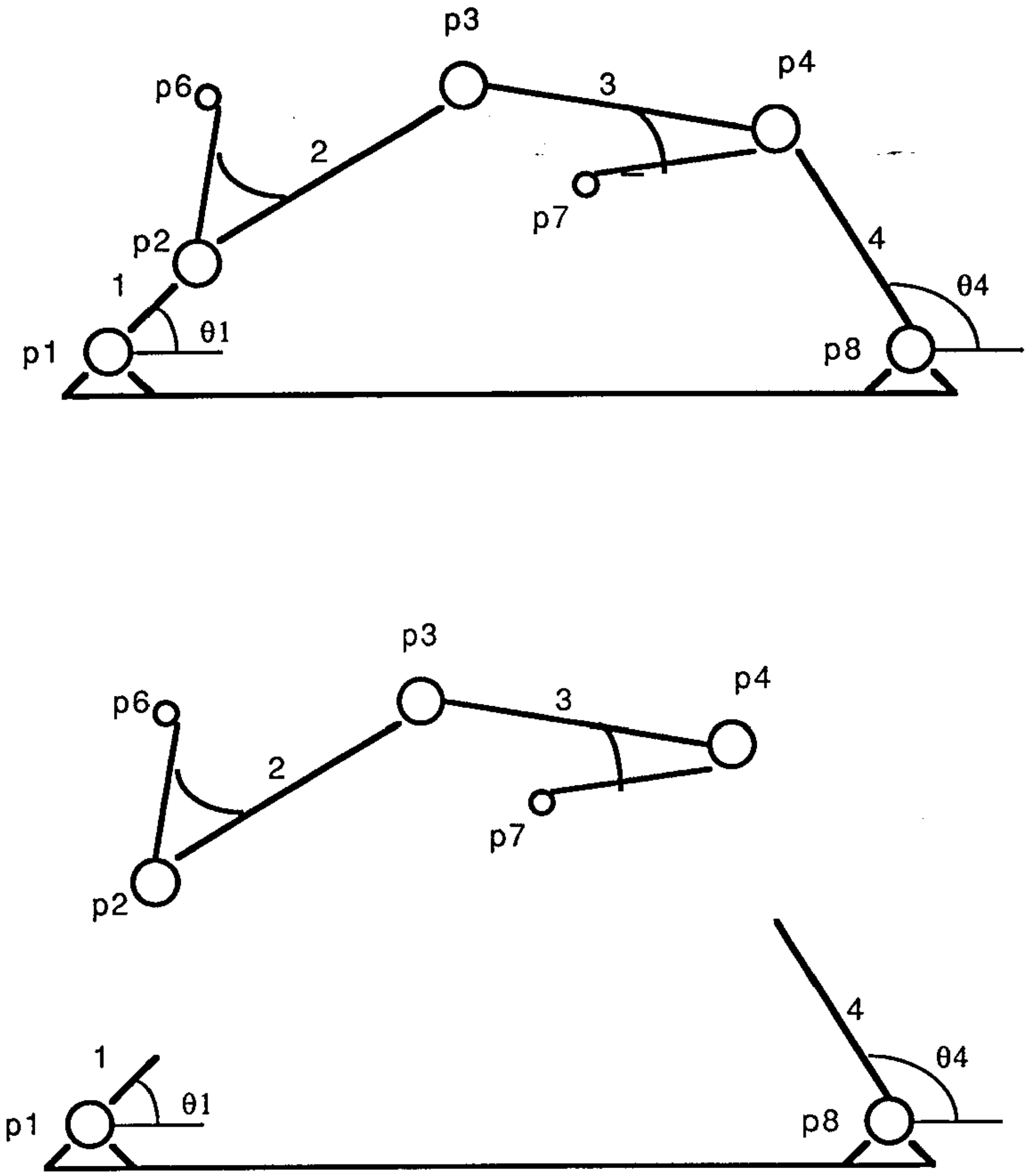


Fig 6.4 Disassembled Five-Bar Linkage

The input link 4, whose position is defined by the specified angle θ4 and whose angular velocity is a function of the package size, is also a rigid body, the motion of the point p4 is known as well.

Link 2 and 3 form a two-link dyad. Since the motion of the point p2 and p4 can be obtained in the previous analysis, the motion of point p3 can be analysed in a manner similar to that described in the section 6.2.3.

In link 2, point p2, p3 and p6 constitute a rigid body. The motion of the coupler point p6 depends on the motion of point p2 and p3, both of which are known in the kinematic analysis of the rigid body and the two-link dyad. The same analysis also applies to link 3. Therefore, the motion of coupler point A can be easily analysed.

So far, the position analysis for the five-bar linkage has been accomplished.

If the dimensions of the five-bar linkage and the motion of the two input links are known, then the position analysis of the coupler point p6 or p7 can be done only by running a computer programme, which includes the subroutines for the rigid body motion and the two-link dyad motion.

6.3 Yarn Length Held in the Compensator

As stated before, the yarn length held in the mechanical compensator depends on the position of the coupler point A. According to the five-bar linkage position analysis, the position of the coupler point A can be decided, thereby the yarn length held in the compensator can be calculated by using the results obtained in the five-bar linkage position analysis.

The yarn length calculation is accomplished in the following steps where the notation is as shown in the Fig 6.5.

In the notation, the coordinate of o1, o2, o are:

o1 (x1, y1), o2 (x2, y2), o (x, y)

r1 is the radius of top guide

r2 is the radius of bottom guide

r is the radius of the roller

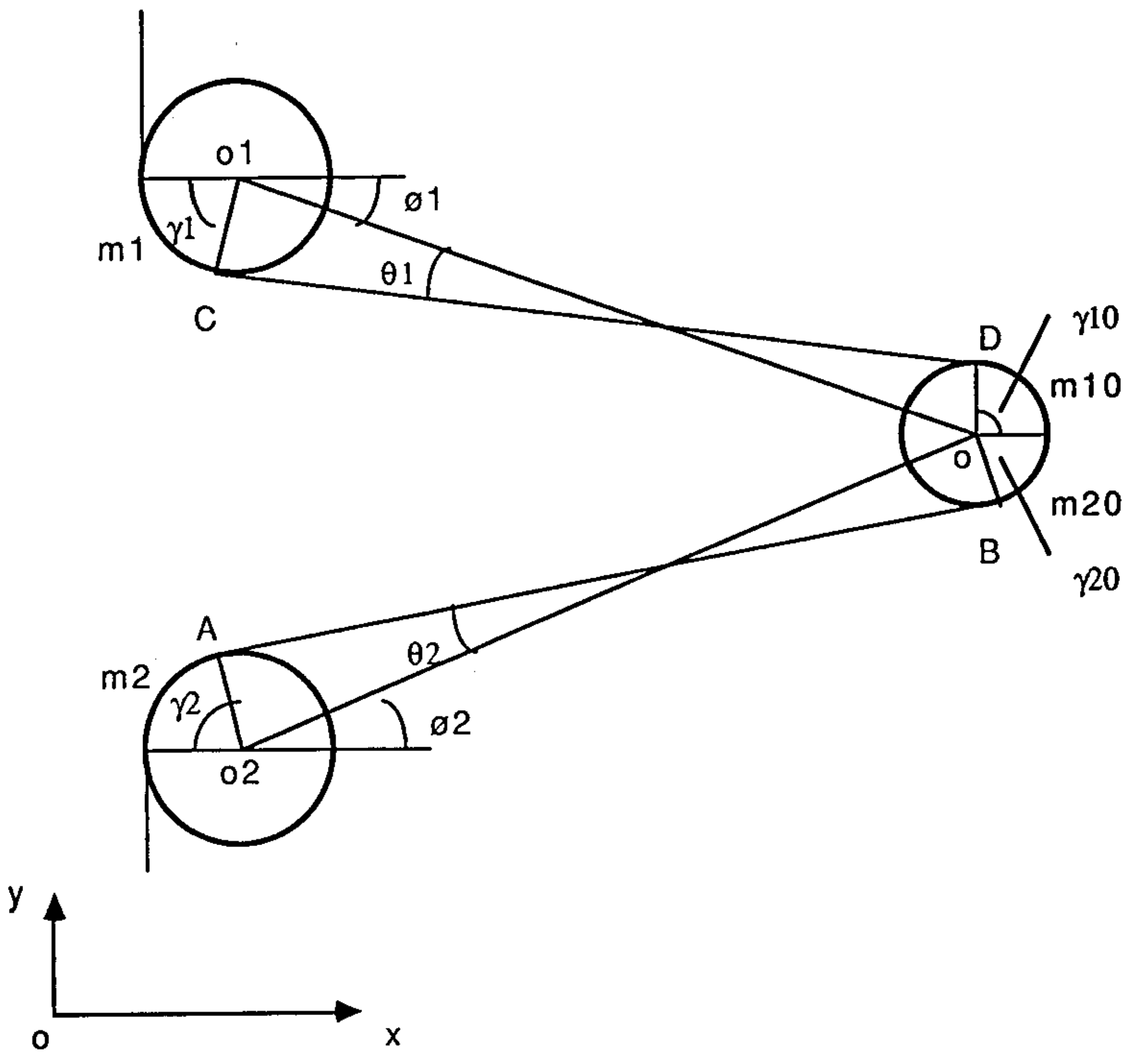


Fig 6.5 Notation for Yarn Length Calulation

1. Calculate the distance AB, CD between the guide and the roller.

$$(AB)^2 = (oo_2)^2 - (r + r_2)^2$$
$$= (x - x_2)^2 + (y - y_2)^2 - (r + r_2)^2 \qquad (6.14)$$

$$(CD)^2 = (oo_1)^2 - (r + r_1)^2$$
$$= (x - x_1)^2 + (y - y_1)^2 - (r + r_1)^2 \qquad (6.15)$$

2. Calculate the angle ø1, ø2, θ1, θ2.

$$\theta_1 = \tan^{-1}\left(\frac{r + r_1}{oo_1}\right) \qquad (6.16)$$

$$\theta_2 = \tan^{-1}\left(\frac{r + r_2}{oo_2}\right) \qquad (6.17)$$

$$\varnothing_1 = \tan^{-1}\left(\frac{y - y_1}{x - x_1}\right)$$

$$\varnothing_2 = \tan^{-1}\left(\frac{y - y_2}{x - x_2}\right) \qquad (6.19)$$

3 Calculate the contact angle γ1, γ2, γ10, γ20.

$$\gamma_1 = \frac{\pi}{2} + \theta_1 - \left|\varnothing_1\right| \qquad (6.20)$$

$$\gamma_2 = \frac{\pi}{2} + \theta_2 - \left|\varnothing_2\right| \qquad (6.21)$$

4. Calculate the contact length m1, m2, m10, m20.

$$m_1 = \gamma_1 * r_1 \qquad (6.22)$$

$$m_2 = \gamma_2 * r_2 \qquad (6.23)$$

$$m_{10} = \gamma_1 * r$$

$$m_{20} = \gamma_2 * r \qquad (6.25)$$

5. Calculate the yarn length held in the compensator

$$L = AB + CD + m_1 + m_2 + m_{10} + m_{20} \qquad (6.26)$$

A computer subroutine has been written to calculate the length of yarn held in the mechanical compensator.

6.4 Flowcharts for the Computer programme

In this section several flowcharts are given for the computer subprograms which are used for the kinematic analysis of the mechanical yarn compensator. These programmes are also prepared to optimise the dimensions of the mechanical yarn compensator.

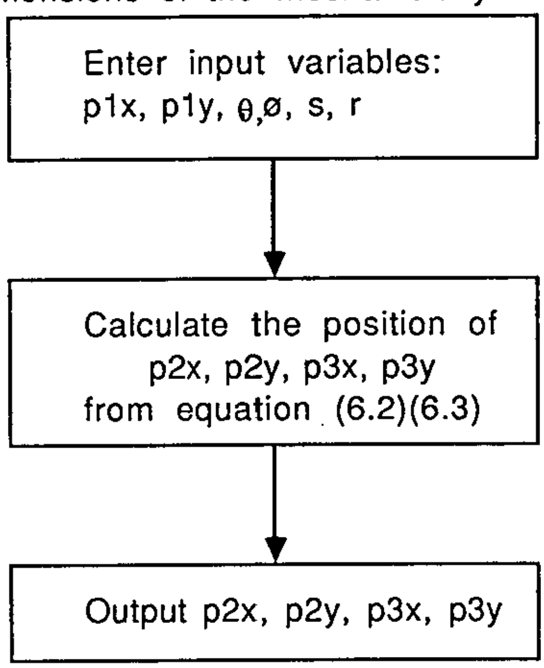


Fig 6.6 Rigid Body Position Flowchart

1. Flowchart for the rigid body position analysis

Based on the equation (6.2),(6,3), the flowchart (fig6.6) describes a computer programme to calculate the coordinates of the points p2 and p3 in fig6.2.

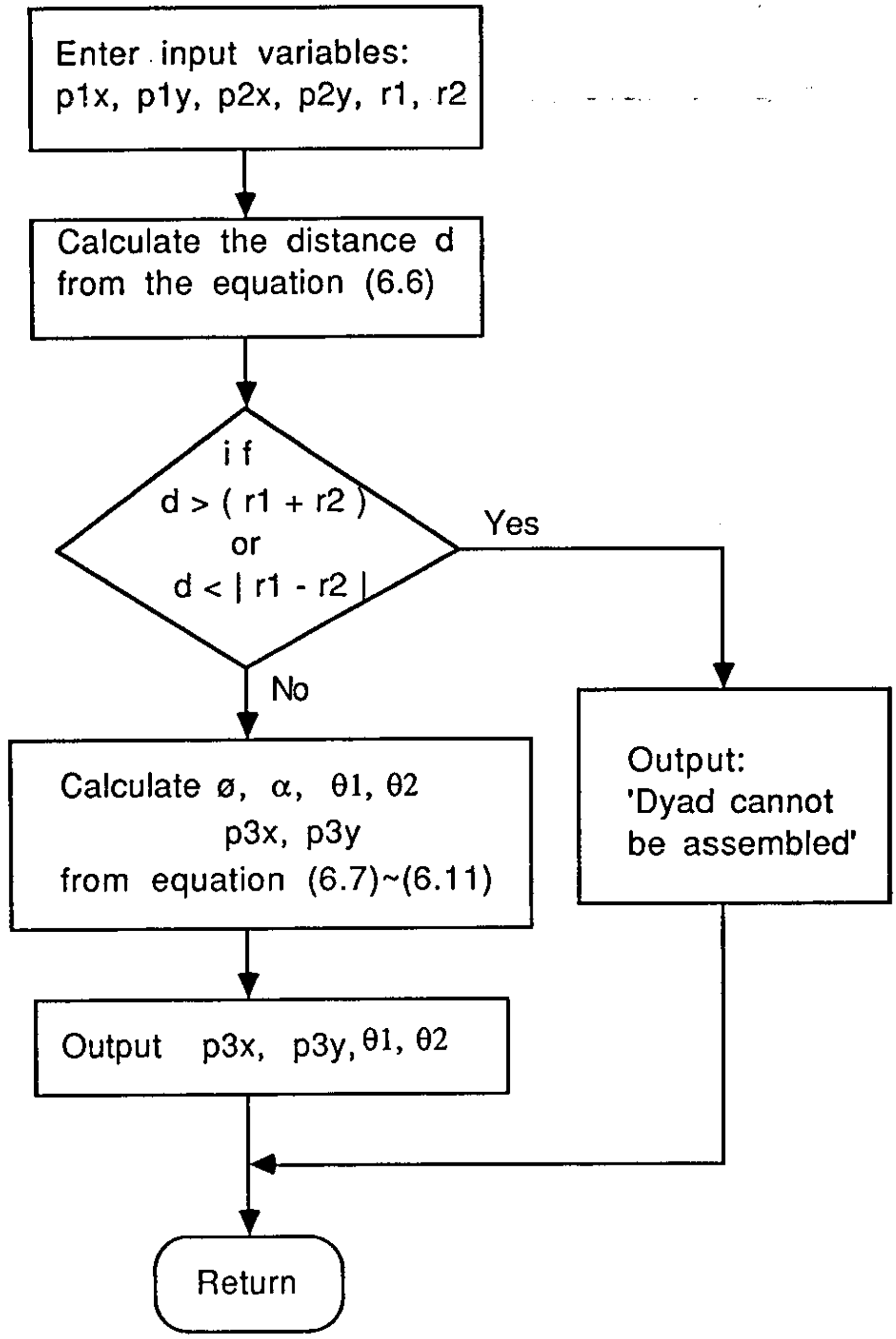


Fig 6.7 Two-link Dyad Position Flowchart

2. Flowchart for the two-link dyad position analysis

Based on the equations (6.6) ~ (6.11), the flowchart(fig6.7) describes a computer subroutine to calculate the coordinates of point p3 and the link position angles $\theta 1$and $\theta 2$ in fig6.3.

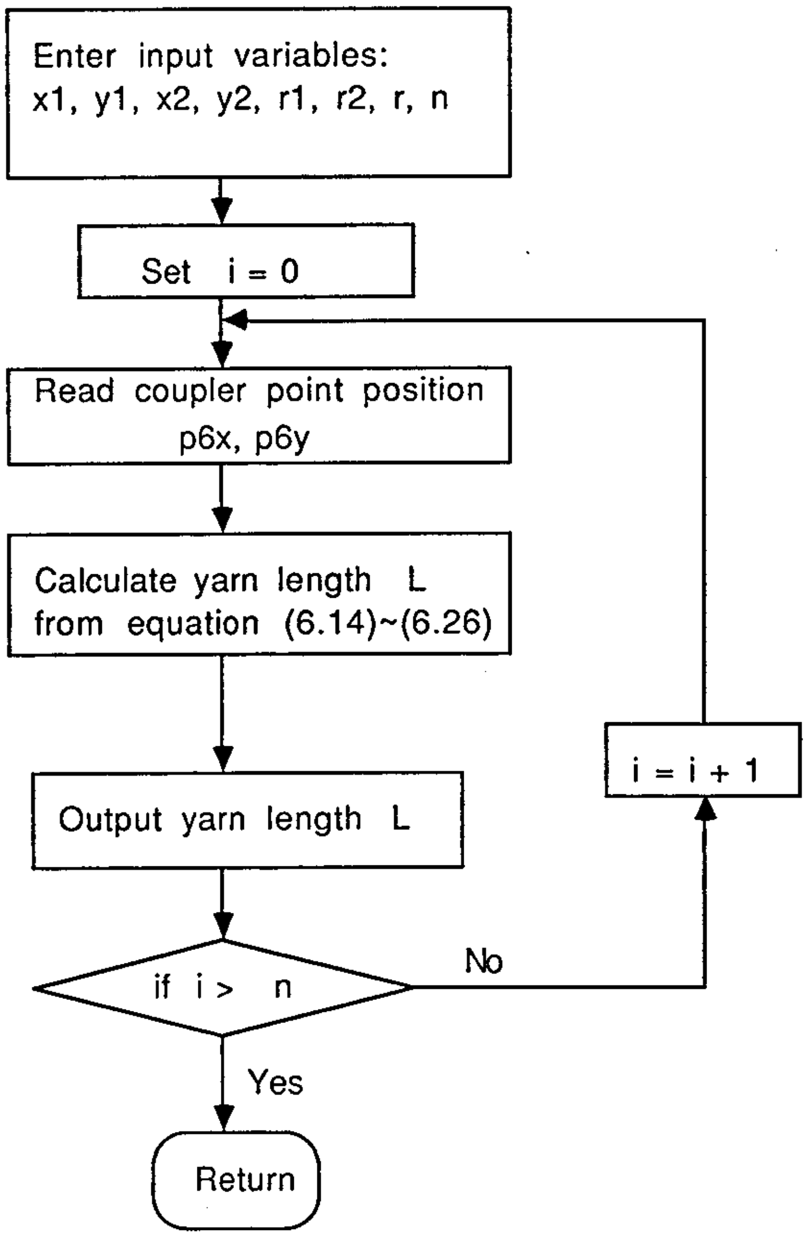


Fig 6.8 Yarn Length Calculation Flowchart

3. Flowchart for the yarn length calculation

Based on the equations (6.14) ~ (6.26), the flowchart(fig6.8) describes a computer subroutine to calculate the yarn length L held in the mechanical tension compensator in fig6.5.

4. Flowchart for the five-bar linkage position analysis

Based on the rigid body and two dyad motion analysis, this flowchart(fig6.9) describes the programme for five-bar linkage position analysis.

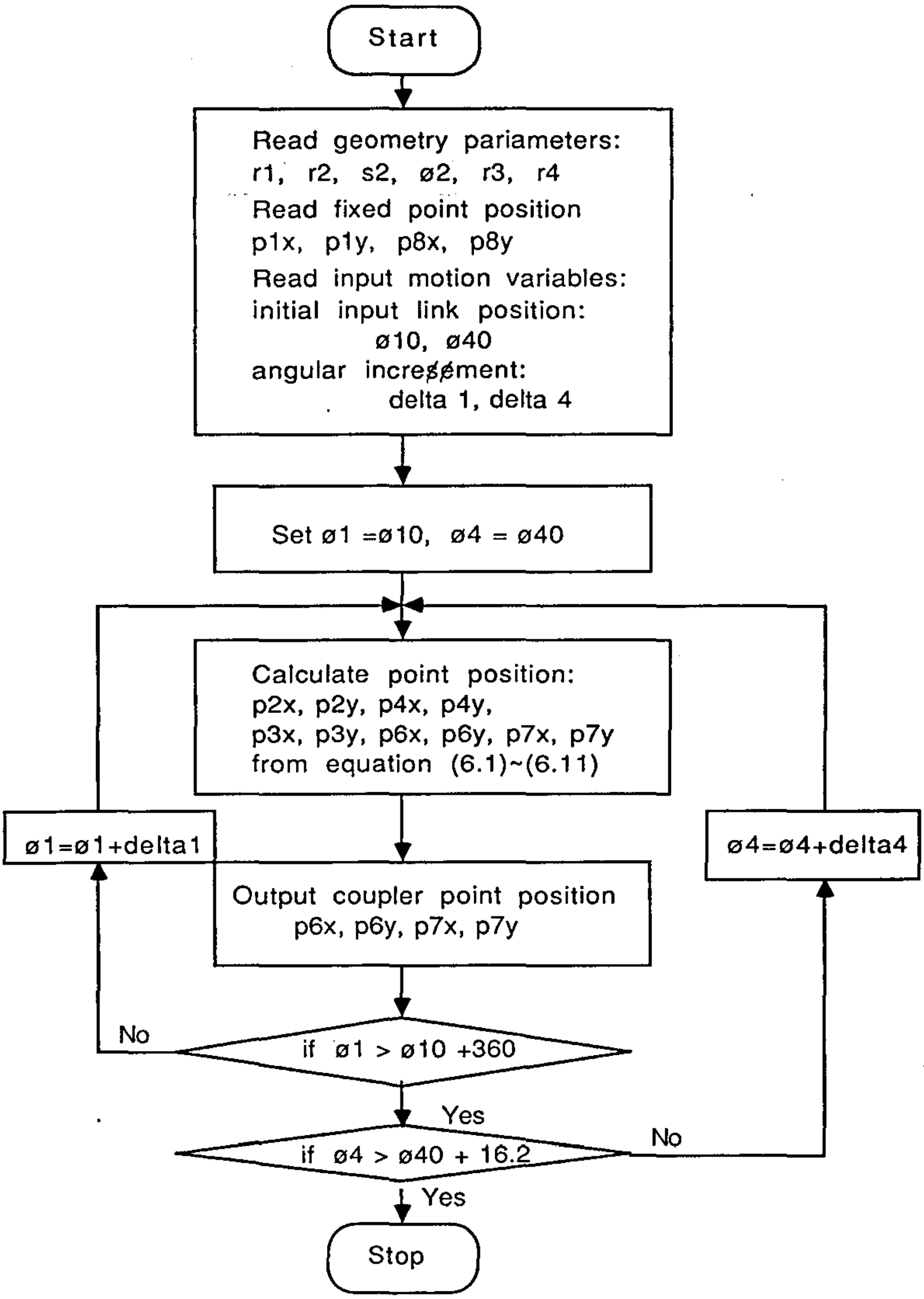


Fig 6.9 Linkage Position Flowchart

6.5 Mechanical Error

Kinematic analysis of the mechanical compensator provides a systematic way to trace the motion of the coupler point, where a yarn roller is fitted. The motion of the coupler point is important, simply because it not only decides the yarn length held in the compensation mechanism, but also decides the yarn supply rate to the yarn winding unit. As mentioned in the previous chapter, in the spinning unit, yarn production rate is constant, whereas in the winding unit yarn winding rate is not constant. The supply rate of a compensation mechanism is determined in such a way that the difference between yarn supply rate and yarn winding rate should be constant in order to keep constant yarn winding tension. The difference between yarn production rate and the supply rate should be taken by the compensation mechanism itself.

A linkage, with constant parameters, which transforms a motion defined by two input variables into another motion defined by an output variable, can be designed to generate a given function in a given interval. Such a given function is the difference between the yarn supply rate and the yarn production rate. Here the integration of the given function over the time interval is defined as mechanical error. The traditional definition of mechanical error which is due to the mathematical or structural error has not be considered. An additional error due to deflection of links, play in the joints, and manufacturing tolerances is also neglected. The kinematic analysis of the motion of the coupler point A and the calculation of yarn length held in the compensator have provided a method to calculate the mechanical error. An optimisation technology is applied to optimise the dimensions of the mechanical compensator in order to produce the desired mechanical error to counteract the winding error. The process is described in the next chapter.

CHAPTER 7 OPTIMISATION OF THE MECHANICAL COMPENSATOR

7.1 Purpose of Optimisation

A linkage can be synthesised in three successive steps as mentioned in the chapter 5. These steps are type synthesis, number synthesis and dimension synthesis. For the mechanical yarn compensator, by using type synthesis and number synthesis, a five-bar linkage has been selected to serve as the yarn tension compensator. It has two input links, these two input links are designed to input two variables, i.e. traverse guide displacement s (winding-on position) and yarn layer thickness z (package size). The output link is designed to output the desired function according to the yarn winding error. However, an adequate compensator is required to work not only properly but also precisely.

In order to make the compensator work at a reasonable level of precision, both the traverse guide and the crank motion are divided into equal numbers of steps. It is desired that in each step, the supply of the yarn matches the demand of yarn in the winding unit. Because of the difference between the winding velocity and the yarn delivery velocity, in some segments, the yarn is in surplus and in others, in shortage. The mechanical compensator should take up or release the extra amount of yarn. If the exact amount of yarn is provided as it is demanded in the winding unit, the yarn tension variation due to variation of yarn winding error will diminish.

The taking-up rate and release rate of the compensation mechanism depend on the motion of the coupler point; however, the motion of the coupler point largely depends on the dimension of the linkage. In this chapter, an optimisation programme called DOT is introduced to determine the linkage dimensions and their initial positions which are critical to the performance of the compensator.

7.2 A Brief Introduction to Numerical Optimisation

Numerical Optimisation offers a number of improvements over the traditional approach to decision making in engineering design. Since optimisation techniques have become available and micro-computers have come into popular use, more and more designers use them in the synthesis of mechanisms.

In the most general terms, numerical optimisation covers the body of numerical methods for finding and identifying the best candidate from a collection of alternatives without having to explicitly enumerate and evaluate all possible alternatives. The technique is very useful in almost every field of engineering.

In order to apply the numerical techniques of optimisation to solve engineering problems, it is necessary to clearly delineate the boundaries of the engineering system to be optimised, to define the quantitative criterion on the basis of which candidates will be ranked to determine the optimum, to select the system variables that will be used to characterise or identify candidates, and to define a model that will express the manner in which variables are related. This composite activity constitutes the process of formulating the engineering optimisation problem. Good problem formulation is the key to the success of an optimisation study.

The general nonlinear constrained optimisation problem can be formulated as follows:

$$\text{Minimise: } F(X) \qquad \text{(objective function)} \qquad (7.1)$$

Subject to:

$$g_j(X) < 0 \quad j=1\sim m \qquad \text{(inequality constraints)} \qquad (7.2)$$

$$h_k(X) = 0 \quad k=1\sim L \qquad \text{(equality constraints)} \qquad (7.3)$$

where

$$X_i^l < X_i < X_i^u \qquad \text{(side constrants)} \qquad (7.4)$$

$$X = \{X_1, X_2, \ldots\ldots, X_n\} \qquad \text{(design variables)} \qquad (7.5)$$

for the problems in which there are no constraints, that is :

$$m = L = 0 \qquad (7.6)$$

and

$$X_i^u = -X_i^l = \infty \qquad i = 1,2,\ldots\ldots,n \qquad (7.7)$$

is called the unconstrained optimisation problem.

Optimisation problems can be classified further, based on the structure of the functions F, g_j and h_k and on the dimensions of x. Unconstrained problems in which x is a one-component vector are called one-dimensional problems and form the simplest, but nevertheless a very important subclass. The objective function F(x), as well as the constraint functions g_j and h_k, may be linear or nonlinear functions of the design variables x. The functions may be explicit or implicit in x and may be evaluated by many analytical or numerical techniques. It is important that these functions be continuous and have continuous first derivatives in x. If equality constraints are explicit in x, they can often be used to reduce the number of design variables considered, consequently the computing time can often be reduced as well.[36]

Basically, using optimisation techniques to solve a problem, an initial estimate for each design variable should be supplied, and then it is necessary to decide the search direction and the step length along this direction in every iteration. There are many methods which can be used to calculate either the search direction or the step length. Detailed information on these methods can be found in many papers and text books.[37~38]

7.3 <u>Optimum Design for the Mechanical Compensator</u>

7.3.1 Introduction to DOT Programme

DOT(Design Optimisation Technology) is a computer Programme for optimisation. Specifically, it is used to automatically adjust parameters to maximise or minimise a calculated quantity, while satisfying a multiple number of constraints. Functions considered by DOT may be linear or nonlinear and may be implicit functions of the design variables. DOT is designed to be a robust numerical optimizer for problems of up to 30 design variables and of no maximum numbers of constraints.[39]

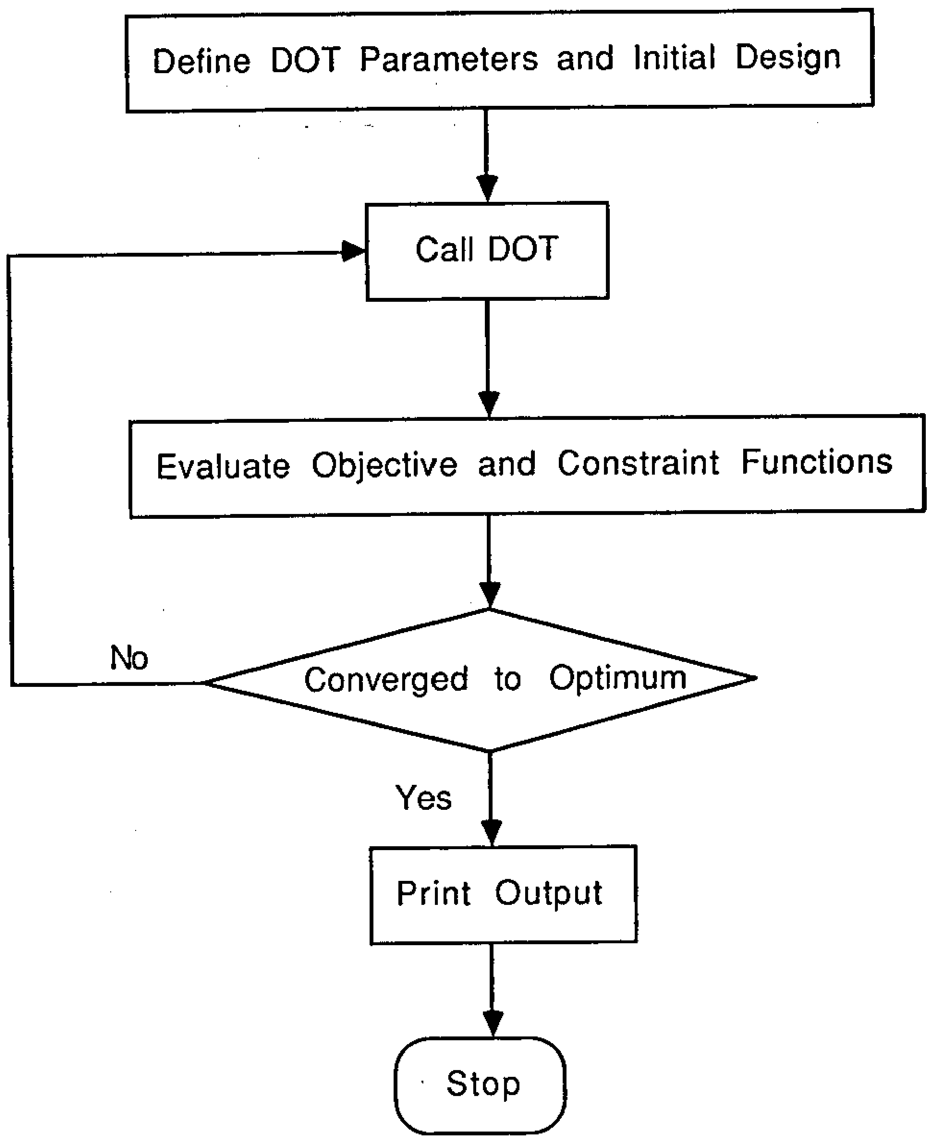


Fig 7. 1 DOT Program Flow Chart

The search algorithm used by DOT is based upon two basic steps. The first is to determine a search direction, which defines how the design variables will be changed. The key idea is that all variables will be changed simultaneously in a fashion that will improve the design. The second part is to determine how far to move in this direction, and this is called a "one-dimensional search". This process of finding a search direction and then searching is repeated until it converges to the optimum.

In this thesis , the Modified Method of Feasible Directions is used to search direction. Having obtained the search direction, the step length along the direction is found by using the Golden Section Method.

Fig 7.1 is a DOT Programme logical flowchart:

7.3.2 Objective function

Based on the specification, the objective of the optimisation of the mechanical compensator is to minimise the difference between yarn demand and yarn supply which is controlled by the mechanical compensator in both short term and long term winding.

Before establishing the objective function , it will be useful to consider some concepts relevant to this Optimisation problem.

* error of winding

ew(i,j) represents the length of yarn surplus (or shortage) corresponding to the ith segment of traverse guide displacement and jth layer of the package thickness. ew(i,j) is the yarn winding error in the specified time period.

* error of mechanism

em(i,j) represents the length of yarn taken up (or released) corresponding to the ith increment of crank rotation and jth position of rocker angular displacement. em(i,j) is the difference between the yarn lengths stored in the compensator in a successive increment of the crank rotation and the rocker angular displacement.

The optimisation of the mechanical compensator is done by minimising the value of a function F(x) which is the sum of the squares of the sum of em(i,j) and ew(i,j). The basic objective function is:

$$F(x) = \sum_{j=1}^{n} \left((8-n)^2 * \left(\sum_{i=1}^{m} (ew(i,j) + em(i,j)) \right)^2 \right) \qquad (7.8)$$

where m=40, both the half cycle of traverse guide displacement and one revolution of crank rotation are equally divided into 40 steps. Each increment of the crank rotation corresponds to a guide displacement segment.

n=7, Both the thickness of a full size package and the full range of rocker angular displacement are equally divided into 7 steps. Each step of the rocker corresponds to a layer of the package.

The sum of ew(i,j) and em(i,j) can be positive or negative, this sum reflects the remaining winding error in each step after the compensation. The sum of the squares of the above sum quantitatively reflects the remaining winding error in the whole cone winding process.

$(8-n)^2$ is the weighting of the objective function. Since the winding tension variation is very much more violent when the package size is small than when it is large. The compensation of the tension variation for a small size package winding should have high priority.

The objective function has considered the yarn winding error in both the short term and long term of cone winding. So, the value of objective function should reflect the remaining yarn winding error which also implies the remaining winding tension variation.

7.3.3 Constraints of the Design

During winding, the size of the package is building up gradually . For a short period, the angle of the rocker to the reference coordinate system hardly changes and neither does the position of point D which is the connecting point of the link CD and rocker DE. So, the five-bar linkage essentially works as a four-bar linkage in a short term. As a four-bar crank-rocker linkage, it must follow Grashof's Law.[40] That is:

> For a planar four-bar linkage, the sum of the shortest and the longest link lengths cannot be greater than the sum of the remaining two link lengths, if there is to be continuous relative rotation between two members.

Fig 7.2 shows the geometry constraints of the linkage compensator.

The following constraints must be observed in Optimisation computing.

If AD is longer than CD then

$$AB + AD < BC + CD \qquad (7.9)$$

If CD is longer than AD then

$$AB + CD < BC + AD \qquad (7.10)$$

assume AB, the crank, is the shortest link.

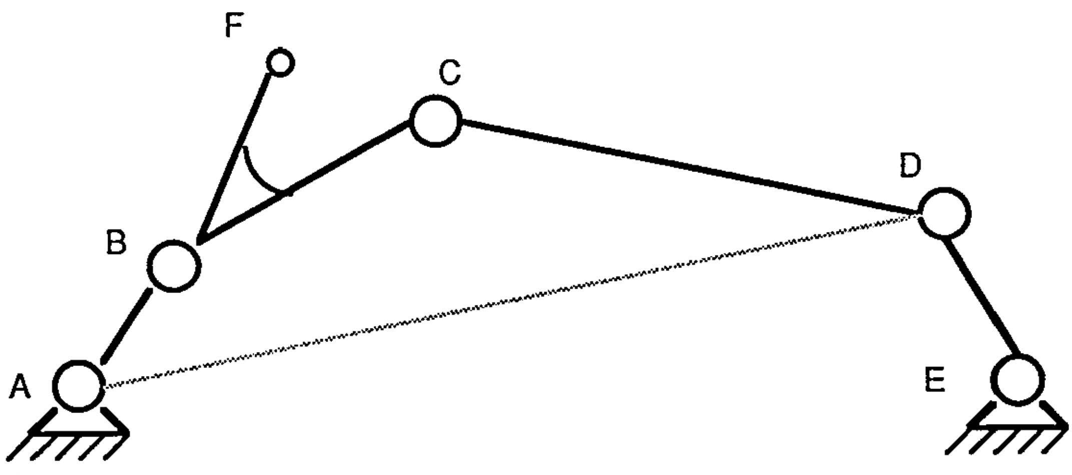


Fig 7.2 Inequality Constraint

As stated before, DE is a part of the package holder. Point D moves away from A along with the building of the package size . This will substantially change the dimension of AD. Therefore, this change needs to be taken into account in the consideration of the design constraints.

In the design of the five bar mechanical compensator, no matter where the positions of the crank and the rocker are, the dyad BCD should always remain assembled. That is

$$BD < BC + CD \qquad (7.11)$$

$$CD - BC < BD \qquad (7.12)$$

These become other inequality constraints of the Optimisation.

7.3.4 Design Variables and Side Constraints

There are eight design variables, x1,x2,......,x8 shown in fig7.3, They are chosen as independent variables, among them,x1, x2, x3, x4, x5 are the dimensions of links, x6 is the angle between x2 and x5, whilst x7, x8 are initial angles of crank x1 and rocker x4.

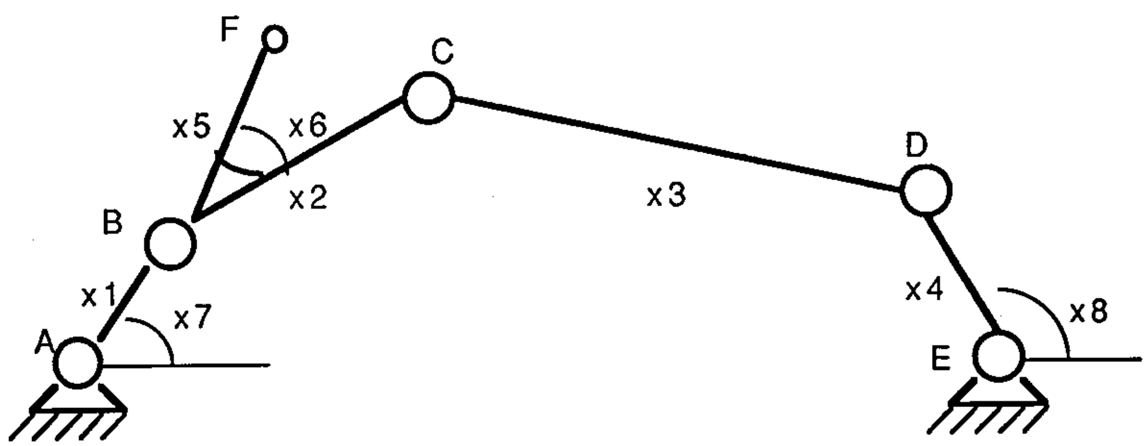


Fig 7.3a Design Variables of Compensator I

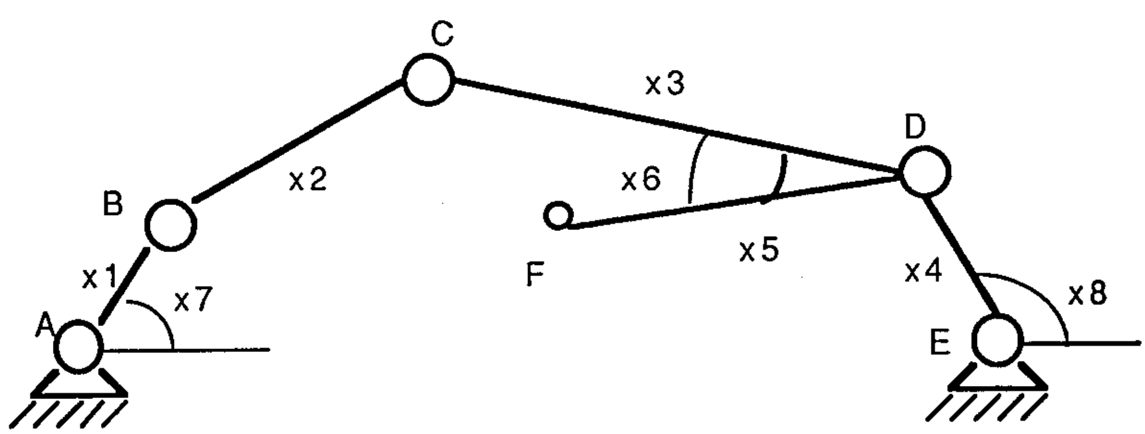


Fig 7.3b Design variables of Compensator II

F is the desired coupler point, x5 and x6 indicate its relative position to the coupler BC(x2) shown in Fig 7.3a. As mentioned in the chapter 5, there is an alternative. F could also be on the coupler CD. In that case, x5 and x6 are indicated in Fig 7.3b.

x7 and x8 are initial angles of the link AB and DE respectively. x7 corresponds to the traverse guide initial position, i.e., at the small end of the package. x8 corresponds to the initial volume of the package, i.e., an empty package.

The other dimensions, such as the location of the fixed bearings A and E are kept the same during Optimisation. The position of A cannot be chosen arbitrarily. The reason for this is that point A is the pivot of the crank , the motion of the crank is transferred from the cam shaft by a train of gears to keep a constant ratio of rotation between the crank and the cam shaft. The position of A should concern the shafts arrangment in the machine. As for point E, which is the pivot of the package holder, any change of the position E will require amendment of the current machine frame.

The side constraints listed above are results of measuring the space limitation and calculating the initial designs.

For every independent variable, the lower and upper bounds of searching are chosen as follows:

Linkage I	Linkage II
3 mm < x1 < 15 mm	3 mm < x1 < 15 mm
15 mm < x2 < 80 mm	15 mm < x2 < 80 mm
270 mm < x3 < 450 mm	270 mm < x3 < 450 mm
10 mm < x4 < 100 mm	10 mm < x4 < 100 mm
10 mm < x5 < 60 mm	10 mm < x5 < 380 mm
0 < x6 < 360	0 < x6 < 360
0 < x7 < 360	0 < x7 < 360
30 < x8 < 220	30 < x8 < 220

Table 7.1 Side constraints of Design variables

7.3.5 Some Design Parameters

The mechanical compensator is fitted in a cabinet under the winding unit. Considering the limitation of space, the compensator should be arranged carefully to make full use of space available, to avoid mechanical interference and to leave enough operating space. In the Masterspinner, the mechanical compensator is arranged in a way shown in Fig 7.4. The origin of the coordinate system is at the crank pivot point A.

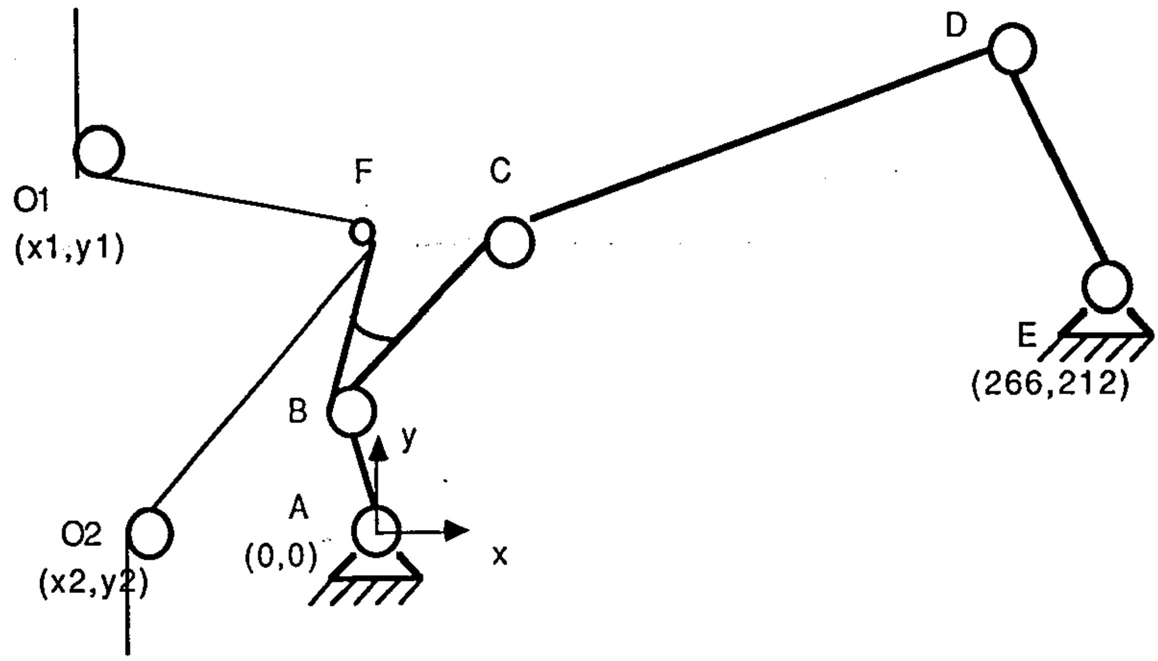


Fig 7.4 Arrangement of Compensator

* top guide position O1

The top guide position O1(-52,44) depends on the centre of the curved bar. The curved bar is designed to eliminate the yarn path length variation and is fitted on the wall of cam cabinet . In this compensator, the centre of the curved bar is used as top guide.

* bottom guide position O2

The bottom guide position O2(-45,0) is on the tangent line of the delivery roller. The position of the bottom guide leaves enough operating space for machine operators.

* angular displacement of the package holder

The package holder has two extreme positions, the low position is for working (winding package), the high position for servicing (changing package etc.). The maximum angle of displacement of the package holder is 56 degrees, whereas, the package holder angular displacement increases by only 18.9 degree during a full size package winding. Since the rocker of the compensator is part of the package holder, this angular displacement requirement should also be satisfied in the design of the mechanical compensator.

7.4 Optimisation Results

After setting up the formulae for the optimisation of the mechanical compensator in the last section, the optimisation still requires initial estimates for each independent design variable. Choosing a good initial design estimate can improve the result if there are several local minima and can also help hasten convergence. Since there is no original design of the five-bar compensator, the initial estimates could be chosen arbitrarily but within the design boundaries.

design model / value of obj. function	1	2	3	4	5	6
initial value	11781	4708	6724	20875	13669	3474
optimum value	3391	3200	3184	6372	3678	2995

Table 7.2 Value of Objective Function

design \ variable		x1	x2	x3	x4	x5	x6	x7	x8
1	initial design	6	40	360	20	30	20	220	150
	optimum design	6.41	40.87	357.16	20.15	29.77	20.06	255.87	149.41
2	initial design	8	35	350	20	25	30	250	150
	optimum design	8.03	29.63	340.46	21.38	28.49	28.95	254.13	147.53
3	initial design	8	40	360	20	30	20	250	150
	optimum design	7.66	37.31	350.43	20.9	32.27	19.64	254.63	147.54
4	initial design	8	45	350	40	55	30	150	150
	optimum design	15	68.45	298.46	11.83	80	58.41	170.71	88.38
5	initial design	8	45	350	40	255	30	200	150
	optimum design	9.59	51.86	341.03	36.54	267.28	30.82	232.94	152.88
6	initial design	6.5	45	350	55	300	15	265	160
	optimum design	6.63	45.48	349.13	54.78	295	15.02	265.5	159.36

Table 7.3 Design Models

Several collections of initial value of design variables have been chosen for optimisation. The optimisation results are listed in Table

7.2 and details of each design model are given in Table 7.3 for comparison.

Note: model 1~3 chose link BC as output link, model 4~6 chose link CD as output link.

From the optimisation results listed in the table7.2 and 7.3, model 3 and 6 are of small optimum value and therefore are selected as the mechanical compensators for further investigation.

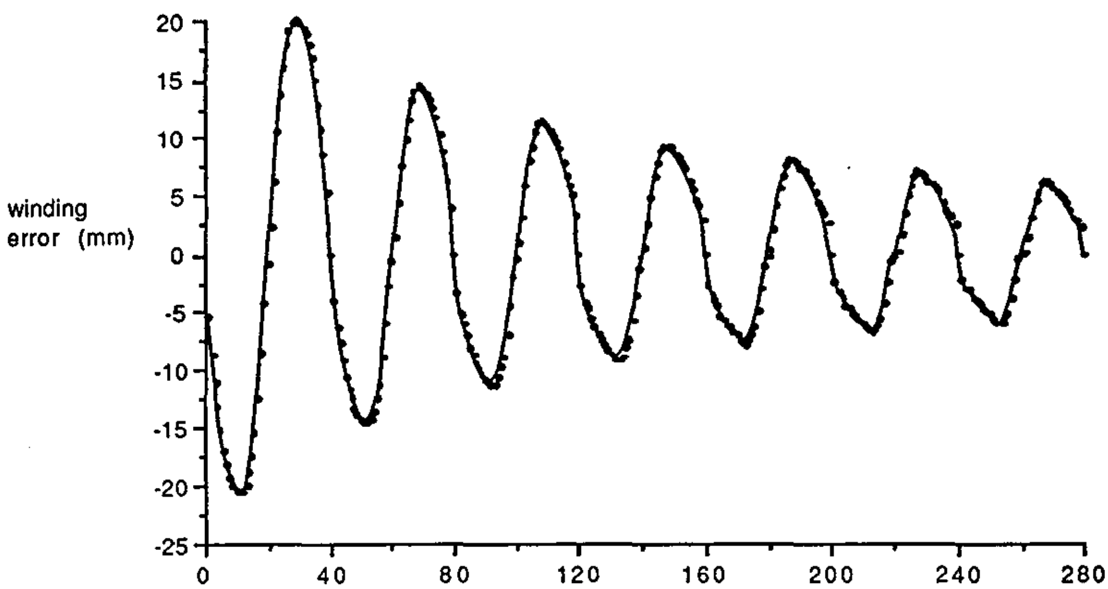


Fig 7.5 Winding Error at Different Package Size

7. 5 Discussion of the Optimisation Results

Computer programmes have been written to calculate the yarn winding error and the mechanical error of the compensators. Fig 7.5, 7.6a and 7.6b are the computer simulation of yarn winding error and the mechanical error in both short term and long term cone winding.

In the calculation, the guide displacement is divided into 40 segments and the rocker angular displacement is divided into 7 steps. The 280 points on the horizontal axis represent these segments and steps. The first 40 points represent the 40 guide displacement segments at the package size z=0 mm. The next 40 points represent another 40 segments at package size z=10 mm, and so on.

Fig 7.5 shows the yarn winding error at different package size, the negative error indicates that amount of yarn which needs to be taken by the compensator to keep constant winding tension in the yarn, the positive error indicates that amount of yarn which needs to be provided. It can be seen that when package size is small, the amplitude of the yarn winding error is about 20 mm. That is why the winding tension variation is very much more violent during small cone winding, but this amplitude of winding error decreases quickly as the package size increases. When package size reaches 60 mm, the amplitude reduces to only 6mm. According to the winding tension experiment, when the package size reaches 50 mm or more, the winding tension variation becomes quite gentle, using or not using a tension compensator does not make much difference.

Figs 7.6a and 7.6b show the mechanical error generated by the mechanical compensators I and II respectively. The positive error indicates the amount of yarn is taken up by the compensator. The negative error is the yarn being released. The amplitude of the mechanical error reduced slowly as the rocker angular displacement decreases in accordance with increase of package size.

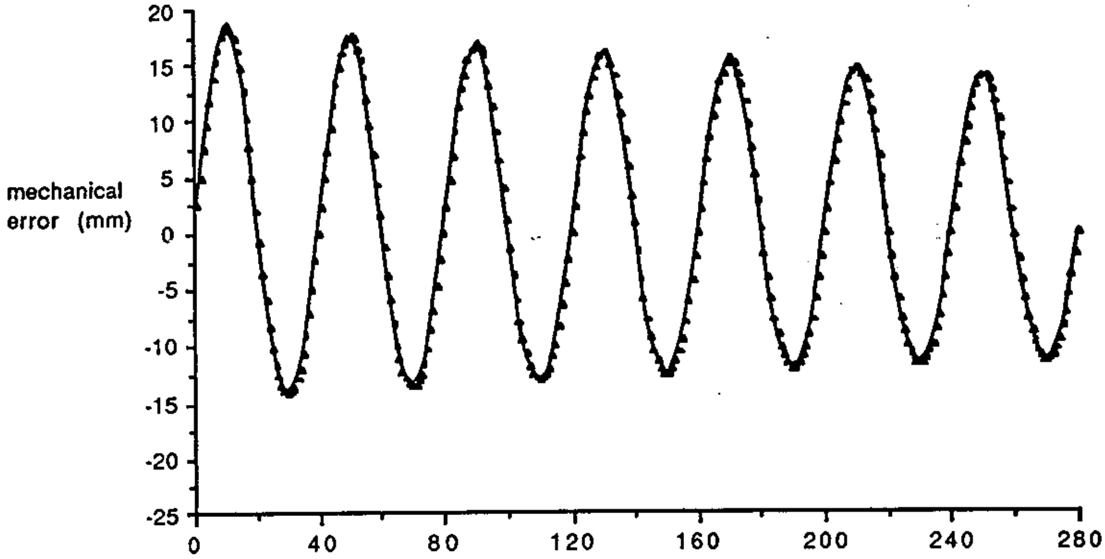


Fig 7.6a Mechanical Error of Compensator I

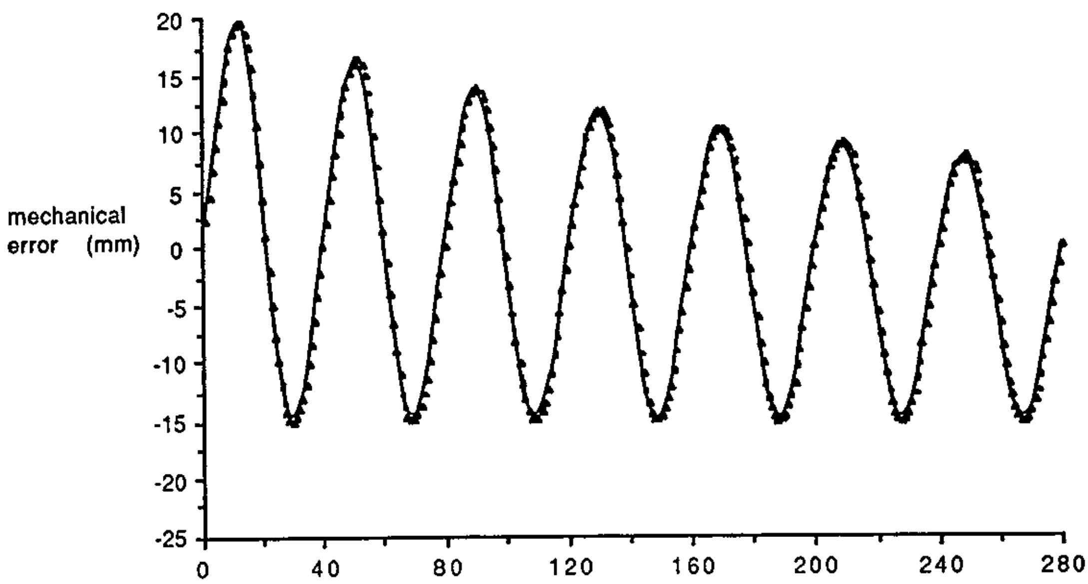


Fig 7.6b Mechanical Error of Compensator II

Both curves are opposite in phase with the yarn winding error curve. That is why the mechanical compensators can make compensation during cone winding: When yarn slack starts to produce, the compensator begins to take up; when yarn is further stretched, the compensator begins to release some yarn.

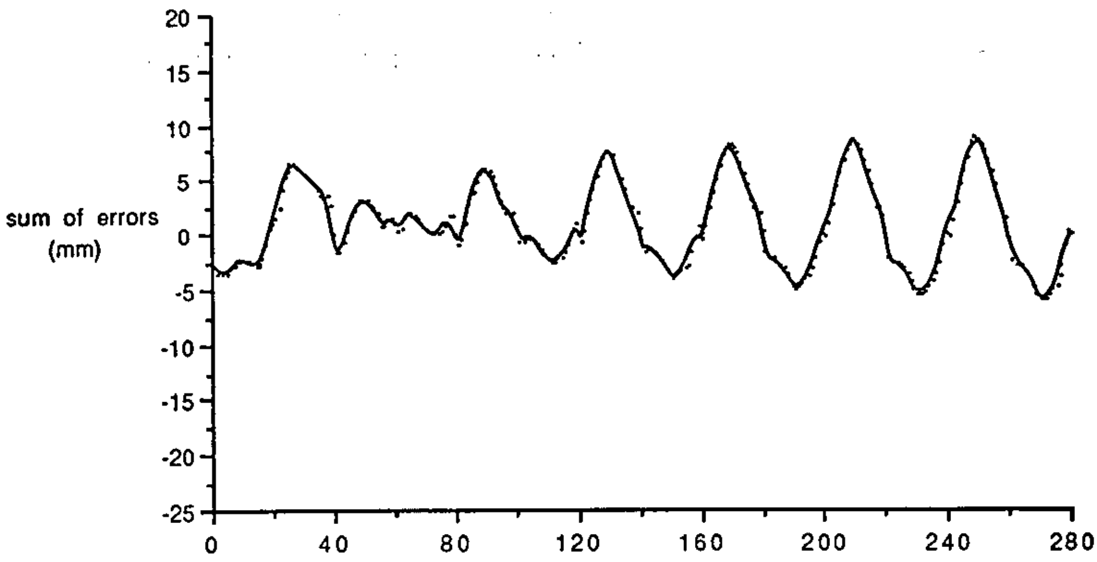


Fig 7.7a Remaining Yarn Winding Error (with compensator I)

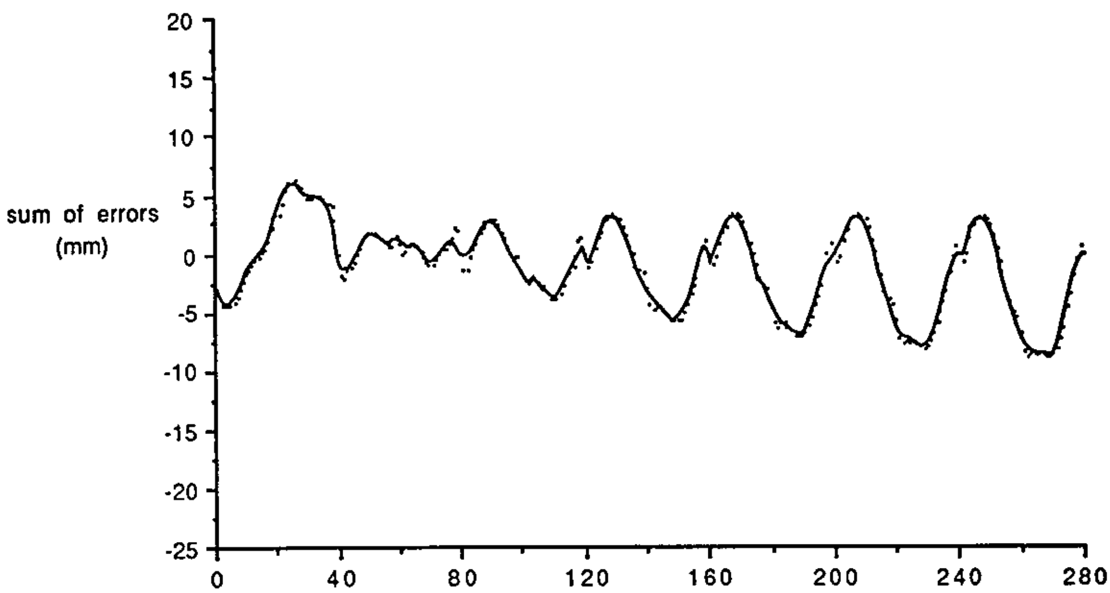


Fig 7.7b Remaining Yarn Winding Error (with compensator II)

Fig 7.7a and 7.7b are the remaining yarn winding error after the mechanical compensator is applied. Because more weighting is added to the compensation of the small package winding (z< 30 mm), this part of the remaining winding error is quite small. However, when the package size gets bigger(z > 40 mm), the remaining winding error also tends to get a bit bigger, this is because the mechanical compensator has overdone its compensation at this stage.

winding error \ package size / model	0 mm	10	20	30	40	50	60
without compensator	40.59	28.85	22.25	18.51	15.77	13.84	12.36
with compensator I	10.05	4.92	8.51	11.47	13.24	14.27	14.89
with compensator II	10.54	4.3	6.98	9.33	10.67	11.55	11.89

Table 7.4 Amplitude of Yarn Winding Error

Table 7.4 lists the amplitude of yarn winding error and the remaining yarn winding error at different package size. It can be seen that after using a mechanical compensator, the remaining yarn winding error can be reduced to one-fourth or even one-sixth of the original yarn winding error when the package size is small. When package size increases (z>=50 mm), the amplitude of remaining yarn error is still smaller than that of the original one if the mechanical compensator II is applied.

From the above computer simulation and discussion, It can be estimated that if these optimized five-bar mechanical compensators are applied in the spinning-winding machine, the winding tension variation can be significently reduced, especially when package size is small. When the package size increases to some stage (eg. z > 50 mm), the winding tension variation probably will increase a bit, but will still be under the original level.

7.6 Conclusion

In this chapter, by optimizing the dimensions of the five-bar compensator, the problem of minimizing the difference between the yarn demand and supply is formulated as a constrained optimization problem, and then the modified method of feasible directions is used to find search direction, and golden section method is used to find the optimum point in this direction. The optimisation programmes are listed in the appendix.

Based on the optimization results, the model 6 has the minimum value of the objective function, can satisfy all the constrants, has a compact dimension and can be well fitted in the cabnet. The most important thing is that it can significently reduce the variation of the yarn winding error in the whole process of cone winding. It is therefore chosen for the proposed mechanical yarn tension compensator. Fig 7.8 gives the optimal layout of the mechanical compensator.

During the optimization, it is found that the initial angle of the crank has a great effect on the initial value of the objective function. This is because the initial angle of the crank determines the phase of the output curve. The definitions of ew(i,j) and em(i,j) indicate that both of them can be positive or negative in a cycle of motion. According to the objective function (7.8), the value of F(x) can be greatly reduced if the ew and em are kept opposite in sign. In this design, This job has been done by the optimization program which chose an optimum initial angle for the crank.

It is also found that the initial value of the design variables greatly affect the value of the objective function, since a lot of local minium points exist. Because of the lack of methods to find the global optimum point, a systematic searching for the optimum point has been carried out and some results are listed in Table 7.3.

From the optimization results and the graphs shown in the Fig 7.5~7.7 and table 7.2~7.4, it can be seen that by using an optimum designed mechanical compensator, there are substantial improvements in reducing the yarn winding error. The amplitude of yarn winding error can be reduced to one-sixth or less of its original value and the remaining winding error can always be kept less than a certain level which a biggest package might have. This certainly means a great improvement in reducing the variation of yarn winding tension and the remaining tension variation can be kept under a level which is accepted by the yarn producers. Considering a mechanical compensator is simple, reliable and has potential ability in reducing yarn winding tension variation to an accepted level, it is a possible solution for the high speed cone winding.

As a mechanical solution, this compensator has a disadvantage in that it is not adaptable. Any change of yarn winding condition will require redesigning and manufacturing of a new compensator. Therefore, it is suggested that alternative solutions, which are easy to adapt to the changing conditions, such as mechatronic one can be considered in future research on the winding tension compensator.

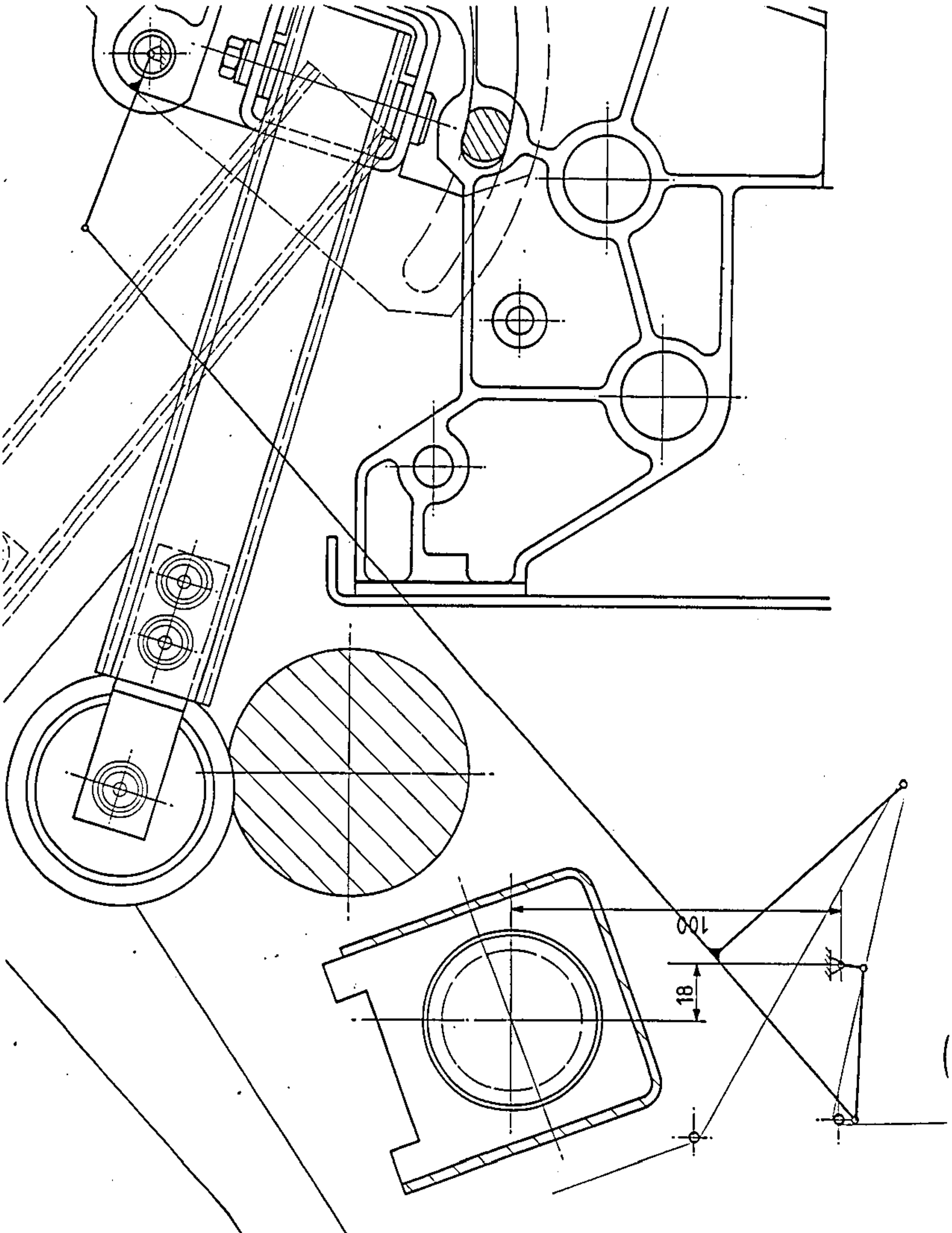


Fig 7.8 Optimal Layout of Mechanical Compensator

CHAPTER 8 MECHATRONIC TENSION COMPENSATOR

8.1 System Specification

In engineering design, electronic, mechanical, fluidic and hydraulic methods have been widely used, each with its respective advantages and disadvantages. In fact, the design techniques have involved largely the same principles, since each component in a mechanism or circuit usually has a well-defined function in the system's behaviour. However, in a design with a microprocessor, the system behaviour depends not only on the system hardware, but also on a programme of instructions to be followed by peripheral devices. To some extent, in a microprocessor based system, it is the programme which contains 'components' to perform specific functions. This character makes the system much more flexible and adaptable than before, since the system can cope with the changing work conditions by only changing its programme. This is the special character of the newly developing technology 'mechatronics'.

The name 'mechatronics' is, of course, the union of mechanics and electronics and it refers to the integrated design of products and systems which possess both a mechanical functionality and embedded electronic control. This is a direct descendant of the long established activity of 'electro-mechanical systems', but a mechatronic system differs from an electro-mechanical one in certain key ways. Firstly, the electronic controller is based upon one or more small, cheap but immensely powerful microprocessors.

Secondly, the control algorithms are essentially a multiplicity of commands and receive a multiplicity of sensor based feedback signals. Thirdly, the processing of these signals is largely by digital rather than more traditional analogue means. Finally, and perhaps most importantly, the process of design which has led to the product will have taken explicit account of the flexibility and adaptability inherent in the use of sensors and microprocessors.

In the survey of yarn tension compensators, it has been found that many types of tension compensator are in use in textile machinery since 1970s'[39], they are, in the main, mechanical, electrical and pneumatic; very few have employed mechatronic technology.[41]

It is unrealistic to spend money on microprocessors just to duplicate a function which can be performed perfectly, adequately by using existing technology. However, considering the complexity and variability of winding tension variation and the unsatisfactory nature of the present tension compensators in high speed cone winding, the newly developed mechatronic technology is introduced here to deal with this problem. Because of its flexibility and adaptability, a microcontroller and a stepper motor together with the aid of a number of additional peripheral devices are proposed to control the yarn winding tension variation.

8. 2 Microcontroller and Stepper Motor Concept

Before describing the mechatronic tension compensator, the concept of microprocessor and stepper motor are briefly introduced here. Microprocessors have now been widely applied in almost every area. Basically, a microprocessor is a programmable device,consisting of many thousands of transistors incorporated into an integrated circuit, operating at logic signal levels, carrying out arithmetic calculations and logic decisions. It is composed of a control and timing unit, an arithmetic logic unit (ALU) and registers. In most computers, it is also called the CPU (central processing unit). It forms the 'brain' of the computing system and performs most control operations as well as any arithmetic and logical calculations. Physically, a microcontroller consists of CPU, memory and ports. Normally, these elements are connected together by means of control bus, address and data bus which carry signals to and from these elements.[42]

The function of memory is that of a storage medium for programme instructions and data. There are two memory types: random access memory (RAM) and read only memory (ROM). RAM has the capability of storing new information in a specific memory location by a write operation and the information later can be transferred to another device by a read operation. ROM has the capability of leaving the information permanently stored for later retrieval so that the information is immediately available for execution when power is supplied to the microprocessor.

The ports provide means of communication with external peripheral devices. They are used for interfacing and controlling when information is to be transferred between peripheral devices and the CPU or main memory. The port associated with each unit can be arranged either to take information in or transfer information to a peripheral. Sometimes a port may be bidirectional, allowing a two-way flow of information. the input/output ports form the interface between microprocessor and peripheral devices.

In fact, a microcontroller is a microprocessor specially configured to control mechanisms and processes rather than to do general purpose calculations. The system it is embedded in is often called a real time control system. Microcontrollers incorporate some form of timing structure to allow synchronisation with the outside world. The capability of microcontrollers to interface to various devices such as stepper motors has broadened the potential for their application.

A stepper motor is an electric motor which converts an electric input into a mechanical motion. Conventional AC and DC motors operate from continuously applied input voltages and most often produce a continuous rotary motion. The basic feature of the stepper motor is that when the motor is energised it will move and come to rest at any one of a number of steps (or detent positions) in strict accordance with the digital input commands provided. Since a motor increments a precise step with each control pulse, a stepper motor provide precisely controllable speed or position. It easily converts digital information into exact incremental rotation without the need of any feedback device. So the system can be an open loop one. The problem of feedback loop phase shift and resultant instability common to analog servo drives can be minimised. A control system using a stepper motor has several significant advantages as follows:[43]

1. no feedback is normally required for either position control or speed control.

2. positional error is non-cumulative.

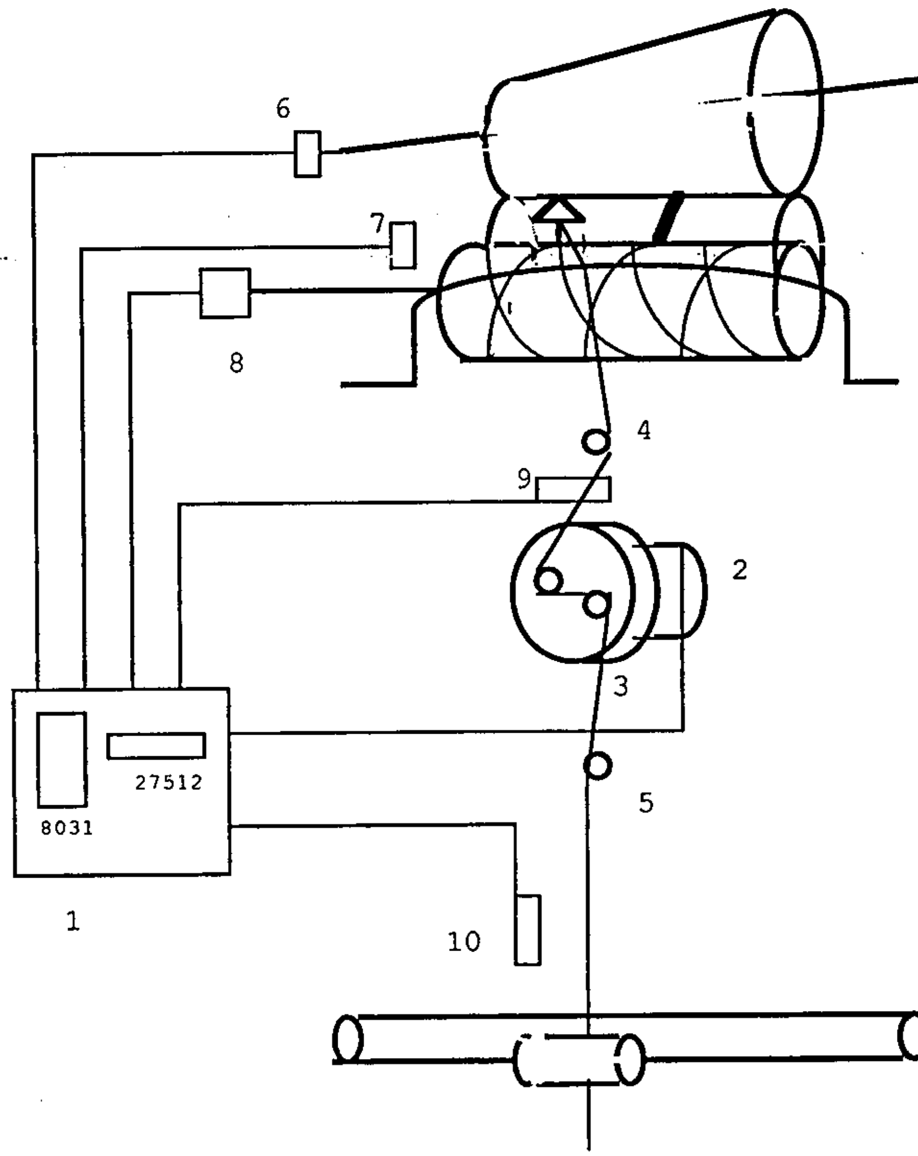
3. stepper motors are compatible with digital equipment.

The characteristics, mentioned above, of microcontroller and stepper motor make it possible to produce a new kind of yarn winding tension compensator.

8. 3 Outline of Mechatronic Compensator

In the design of the mechatronic tension compensator, a control board, a stepper motor and a few peripheral devices are formed a compensator. The compensation principle is the same as that of the mechanical ones, that is, to reduce the difference between yarn supply and demand in the winding unit. If yarn is in surplus, the compensator should take-up some yarn, if in shortage, then release some. Fig.8.1 is a block diagram which outlines the mechatronic tension compensator.

The working principle of the mechatronic compensator is that the 8031 microcontroller controls the rotation of a stepper motor 2 and its bollard disc 3, according to predetermined look-up tables stored in 27512 EPROM. As the yarn is wrapped around the bollard disc, the rotation of the bollard disc changes the yarn length held in the compensator, by taking-up or letting-out yarn, the compensator changes the yarn supply rate and reduces yarn winding error.



1 control board 2 stepper motor 3 bollard disc
4 top guide 5 bottom guide 6 cone size sensor
7 magnetic switch 8 camshaft encoder 9 optical encoder
10 yarn break switch .

Fig 8.1 Concept of Mechatronic Tension Compensator

The working procedure is: In the winding process, the cone size sensor 6 enables the microcontroller to determine current cone size. When a traverse guide leads yarn to the small end of cone, the traverse guide approaches a magnetic switch 7 and activates it, the magnetic switch then sends a signal back to the 8031 microcontroller. As soon as the microcontroller gets the signal and the signal from the cone size sensor, the microcontroller processes this information and selects a corresponding look-up table from the 27512 EPROM and then outputs driving instructions to the stepper motor according to the look-up table. The stepper motor turns the bollard disc to a series of positions at predetermined step rate either to take up or to let out yarn. This procedure repeats continuously until a yarn break signal occurs.

The encoder 8 fits on the camshaft and gives 2500 pluses at each revolution of the camshaft. Since a complete cycle of traverse guide motion corresponds to thirteen turns of the cam shaft, that makes a total of 32500 pulses for each complete cycle of traverse guide motion. This is divided by 10 to give 3250 latter on. The pulses are sent to the control board as the basis of the step rate count.

An optical encoder 9 is fitted behind the bollard disc which is half silver and half black; the function of the optical encoder is to check the bollard disc position. At its start position the bollard disc should park at the top centre position and wait to synchronise the motion of the traverse guide.

A sensor 10 is required to check for yarn breakage, but in this experiment, a switch is placed instead of the sensor to imitate the yarn break condition. If the switch is turned off, it indicates that the yarn is broken and the bollard disc will reverse to an upside-down position and park there waiting for piece-up.

The 8031 microcontroller manipulates the whole process by following an assembly language programme and look-up tables stored in the 27512 EPROM. This assembly programme is a series of commands used to instruct the microcontroller to do compensation work step by step. The look-up tables are calculated on the basis of the analysis of yarn winding error. As stated before, yarn winding error is defined as the difference between yarn supply and demand and is the function of traverse guide displacement and cone size. Each look-up table counts a total of 3250 over all its entries, corresponding to the camshaft encoder counts in one complete cycle of traverse guide motion. The look-up tables determine the stepper motor rotation direction and step rate. By following a predetermined look-up table, the microcontroller controls the step rate and position precisely, thereby controls the yarn length stored in the compensator and its supply rate to the winding unit. If the yarn supply matches the yarn winding error, then the winding tension variation can be greatly reduced.

8. 4. Control Board Circuit

In the mechatronic compensator design, the peripheral devices are connected by electric circuit. Fig8.2 is the circuit diagram. It can be divided into three functional sections, which are discussed below.

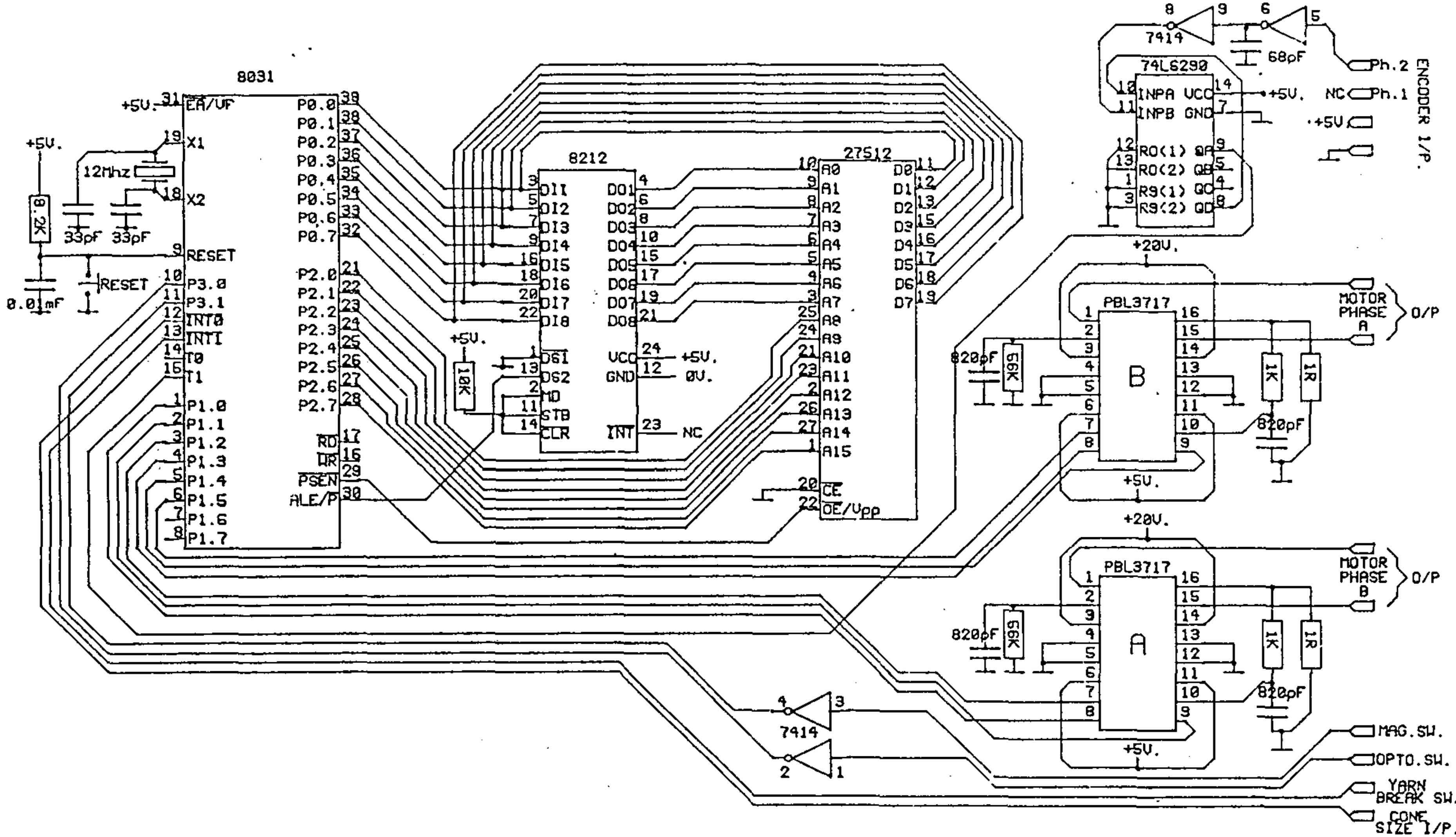


Fig 8.2 Control Board Circuit Diagram

8.4.1 Microcontroller Circuit

The 8031 microcontroller is based on the Intel MCS-51 architecture which has a 8-bit CPU, an on-chip oscillator and four 8 bit ports. It also has a five-source interrupt structure with two priority levels. However it only has 128 byte on-chip RAM and no on-chip programme ROM.[44]

In Fig.8.3 the pins X1, X2 are the input/output of a single stage on-chip inverter, which is configured with a 12 MHz quartz crystal as an external oscillator. The oscillator drives the internal clock generator. The clock generator provides the internal clocking signals to the chip. The internal clocking signals are at half of the oscillator frequency, and define the internal phase states and machine cycles.

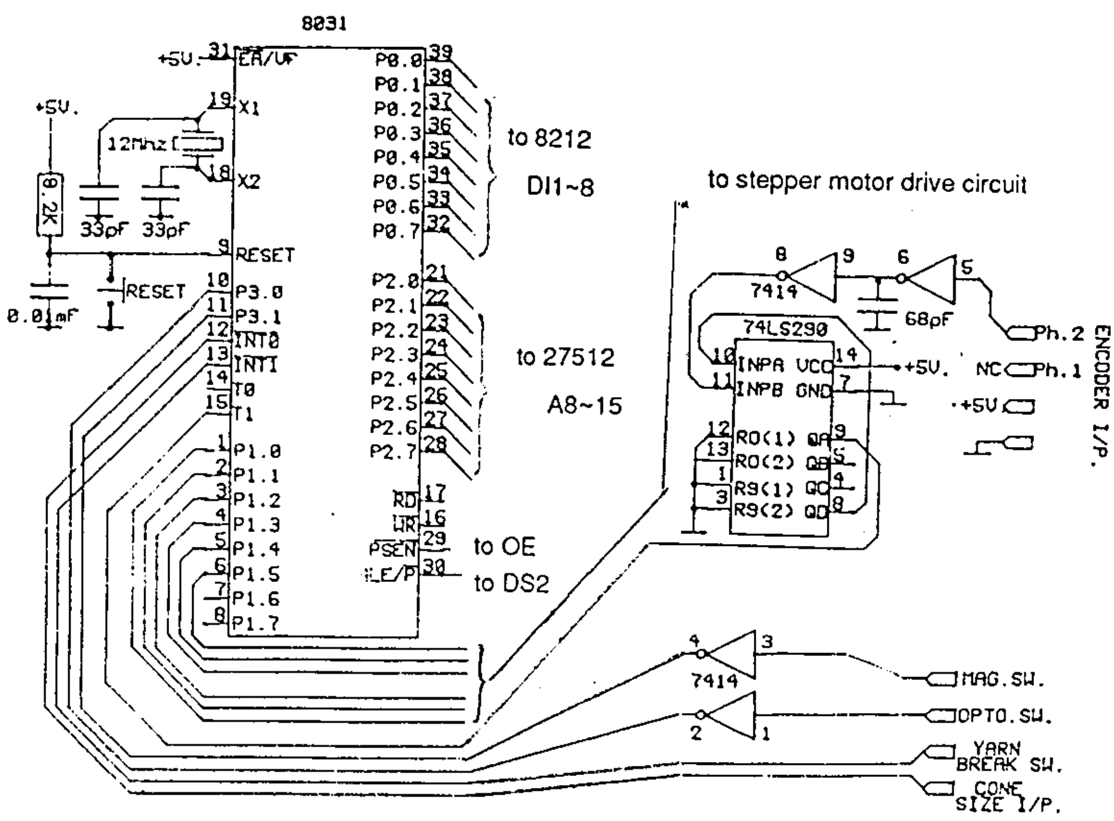


Fig 8.3 Microcontroller Circuit

The four ports in the 8031 are bidirectional, each consists of a latch an output drive, and an input buffer.

The output drives of ports 0 and 2, and the input buffer of port 0 are used to access a 27512 external memory. In this application port 0 outputs the low byte of the external memory address, which is time multiplexed with the input data byte being read. Port 2 outputs the high byte of the external memory address to the 27512.

Port 1 outputs the control instructions to the stepper motor control circuit when a step needs to be taken by the stepper motor. Pin 1.0 controls the direction of current flow, whereas Pin 1.1 and Pin 1.2 control the current level in the stepper motor winding A. Pin 1.3,1.4 and 1.5 do the same job for the stepper motor winding B.

Port 3 is used to input control signals. It is the channel by which the microcontroller communicates with the outside world. Pin 3.0 is connected to the optical switch which is used to check the initial position of the stepper motor. Pin 3.1 to the magnetic switch which is used to check the traverse guide start position and start a look-up table, and pin INT0 to a frequency generator which is used to imitate the cone size sensor to select a suitable look-up table for the appropriate cone size. Pin INT1 is connected to the yarn break switch which is used to imitate the yarn break condition and Pin T1 to the 74LS290 counter chip to get the camshaft encoder pulses which are the basis of the step rate count.

The camshaft encoder used is a high resolution encoder. It generates 2500 pluses for each revolution. That proved to be too high a frequency . So a 74LS290 counter IC is employed to proportionally reduce the pulse frequency to the microcontroller .

74LS290 is a decade counter. By connecting the Qd output to the A input and applying the input count to the B input, it gives a divide-by-ten square wave at output Qa. The square wave is then used by Timer 1 in the 8031 to count the step rate. A 68pf capacitor is used here to filter the high frequency noise from the input signals.

The RESET pin in the microcontroller 8031 is used to reset the system. It connects to the ground through a 10μf capacitor and to Vcc through an 8.2KΩ resistor; the voltage at RESET is the same as capacitor. The current drawn by RESET commences to discharge the capacitor, The larger the capacitor, the more slowly the voltage of RESET increases, the voltage of RESET will remain low long enough to effect a complete reset.

8.4.2 Memory Circuit

The Intel 27512 external memory is a 64k byte ultraviolet Erasable and Programmable Read Only Memory (EPROM) and is directly compatible with the Intel 8051 family of microcontrollers, of which the 8031 is a member. It functions as a high density software carrier; the software can be resident in the 27512 EPROM directly on a system's memory bus. and it permits immediate microprocessor access and execution of software.[45] Fig.8.4 shows the memory circuit. In this circuit, the OE pin (output enable) in 27512 connects to PSEN pin (programme store enable) of the 8031, a low level pulse from PSEN enables the 27512 EPROM to output a programme instruction or data back to the 8031 microcontroller for further processing. In the 27512 EPROM, pin A8~A15 connect to port 2 of the 8031 to input the high address byte, pins D0~D7 connect to port 1 in 8031 to output programme instruction and data, whereas, pins A0~A7 connect to an 8212 chip. The Intel 8212 is an input/output port and consists of an 8-bit latch with 3-state output buffers. The 8031 microcontroller uses a multiplexed address/data bus that contains the low order 8-bits of address information during the first part of the machine cycle, and the same bus contains data at a later time in the cycle. Because of this an address latch enable (ALE) signal provided by the 8031 can be used by the 8212 to latch the low byte of the address. In this way, the address can be available through the whole machine cycle. The 8031 microcontroller can thus read the data out from the 27512 EPROM successfully.

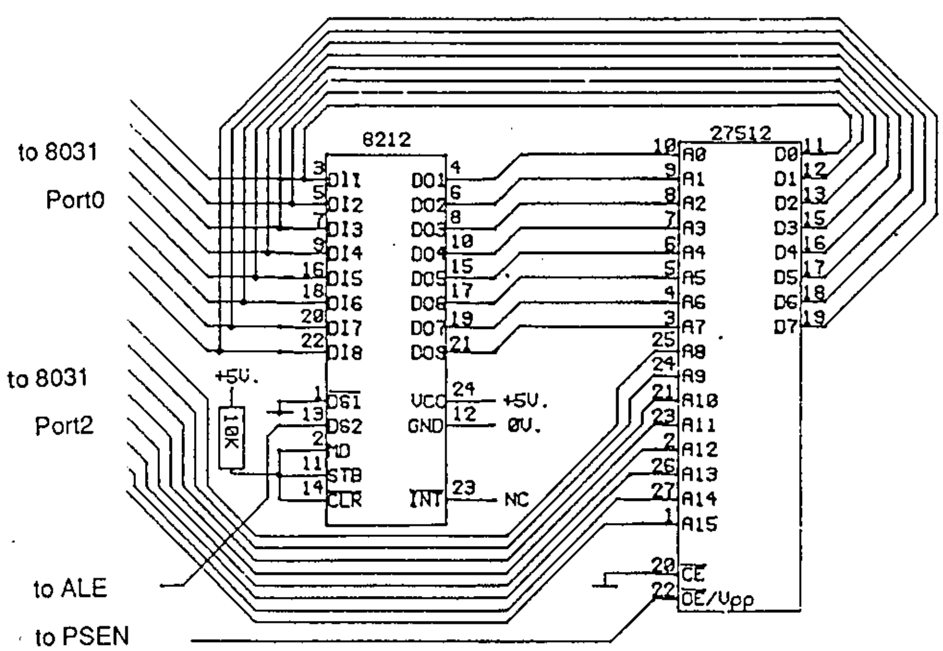


Fig 8.4 Memory Circuit

8.4.3 Stepper Motor Drive Circuit

Fig.8.5 is a typical stepper motor drive circuit. Two PBL 3717 and a few external components form a complete control and drive unit for this microcontroller control stepper motor system. The PBL 3717 is a bipolar monolithic integrated circuit used to control and drive the current in one winding of the bipolar stepper motor PQ40-200c. The circuit consists of an LS-TTL-compatible logic input, a current sensor, monostable and an output stage with built-in protection diodes.[46]

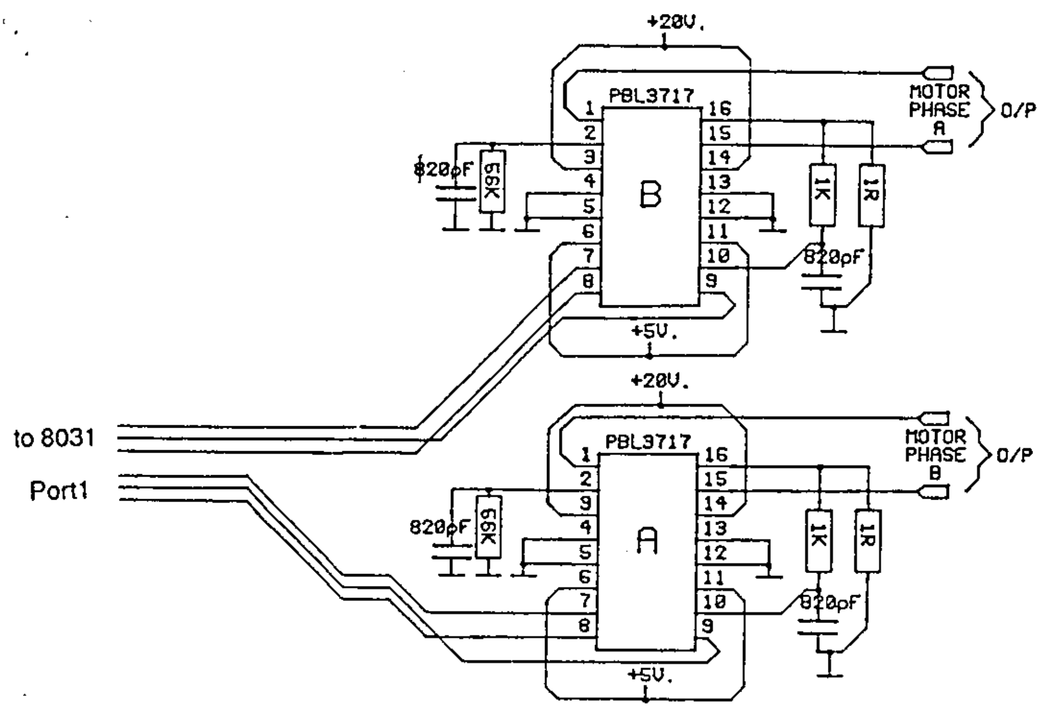


Fig 8.5 Stepper Motor Drive Circuit

In PBL 3717, pins 1 and 15 connect to one winding of the stepper motor, Phase control Pin 8 connects to port 1 of the 8031 microcontroller. The phase input determines the direction of current flow in the winding; a high level on the phase-input pin 8 causes the motor current to flow from pin 15 through the winding to pin 1.

The current level in the motor winding is selected with the logic input pins 7 and 9,which also connect to the port 1 of the 8031. The values of the different current levels are determined by the reference voltage on pins 6 and 11 together with the one ohm (1R) sensing resistor.

The 1K resistor and the 820 pF capacitor form a low pass filter for noise immunity, the 56K resistor and the 820 pF capacitor are timing components to determine the pulse time that switches off the power feed to the motor winding and cause the motor current to decrease. The chopper control uses an unregulated power supply to operate at a point well below the maximum rated supply current. The chopper drive circuit can increase the stepper motors high speed performance.

In the mechatronic tension compensator, the 8031 microcontroller is the 'brain' of the compensator, it reads and executes instructions fetched from the 27512 EPROM memory; the stepper motor and its bollard disc are the 'hands', they follow the instructions from the microcontroller and carry out compensation work. The angular displacement and speed of the bollard disc depend on the look-up tables; the control circuits are the 'nerves'; they guarantee the communication among the microcontroller, stepper motor and other devices. All these 'organs' are under the supervision of an assembly programme. If any winding condition changes, only the programme or its look-up tables need to be adjusted to meet the new condition. That certainly makes the mechatronic compensator very flexible and adaptable.

CHAPTER 9 SOFTWARE DESIGN

9. 1 Compensation Programme

9.1.1 Introduction

In a system with a microcontroller, normally the microcontroller is very much the system's supervisor. The microcontroller controls the functions performed by other components. It fetches instructions from memory, decodes their binary contents and uses them to initiate various processing actions. During the execution of instructions, the microcontroller references memory and I/O ports as necessary. It also recognises and responds to various externally generated signals. The microcontroller does this work by following a programme which is written in assembly language. In the design of the mechatronic tension compensator, an assembly programme which is executed by the 8031 microcontroller has been assembled and loaded into the 27512 EPROM. The objective of this programme is to control the motion of the stepper motor according to preset look-up tables and also according to external interrupt signals.

This yarn tension compensation programme consists of a main programme, a cone size comparison routine, a stepper motor park routine, an input routine to get the step rate count from look-up table and an output routine to drive the stepper motor in half-step mode. There are also two interrupt service routines, one for cone size interrupt, the other for step rate interrupt.

The mechatronic tension compensator is based on a microcontroller and a few interface chips, by receiving multiple sensor signals and following a preset programme, it manipulates a stepper motor as tension compensation mechanism.

9.1.2 Flow Chart of the Compensation programme

Before any programme can be written. It is necessary to plan the logical sequence of events to be carried out to achieve the desired goal. A number of techniques may be used, but the most useful for assembly language programming is the construction of a flow chart. This is simply a diagram which indicates the sequence of events and action to be taken and the points where branching is required.

The winding tension compensation programme can also be illustrated more clearly in a form of flow chart. The following flow charts give the general views of how the microcontroller obtains the variable cone size count, compares it with look-up table entry count, selects a corresponding look-up table and outputs instructions to the stepper motor and how it responds to external signals . Only the main programme and major subroutine flow charts are discussed. Followings are some symbols used in the flow chart, their functions are briefly described for reference.

Microcontroller input pins:(see fig8.3)

P3.0: connected to magnetic switch

p3.1: connected to optical switch

P3.2(INT0): connected to cone size sensor

P3.3(INT1): connected to yarn break switch

Registers:

R0: stores forward/reverse offset to half-step sequence

R1: stores common offset to 'CNTCMP' and 'DTAPTR' tables

R2: stores stepper motor turning direction command

R3: stores pre-fetched step-rate count

The compensation programme is listed in the appendix. The functions of compensation programme are briefly introduced here.

1. main programme

The main programme is to initialise the microcontroller, to select and update look-up table, to load step rate counter and check if yarn is broken.

2. input routine

The input routine is to load data pointer with command or count. If it is a 'turn' command, then change stepper motor rotation direction. If it is a 'repeat' command, start at the origin of the current look-up table. If it is a count, load it into step rate count register.

3. output routine

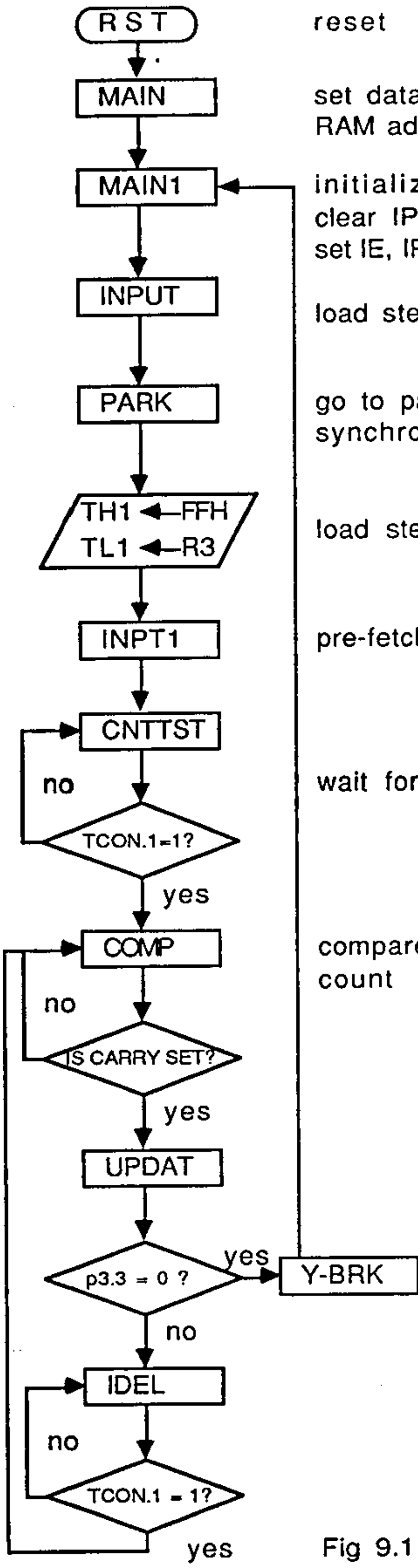
The output routine is to output a step forward or backward instruction to the port 1 according to half-step sequence table.

4. park routine

The park routine is to reverse the bollard disc to a park position and wait for synchronisation.

5. cone size comparison routine

This routine is to select proper look-up table by comparing cone size count to look-up table table entry count.



reset

set data pointer to DTAB0 and preserve its origin in RAM address 34H,35H

initialization
clear IP,SP, Timer0, Counter1, R0,R2
set IE, IP, TMOD, TCON

load step rate count in R3, direction command in R2

go to park position and wait for yarn piece-up and synchronization

load step rate counter1 with the content of R3

pre-fetch next step rate count and preserve in R3

wait for valid cone size count

compare cone size count with look-up table entry count

wait the end of the current look-up table

up date look-up table and preserve its origin in RAM address 34H, 35H

if yarn is broken, clear IE,TCON
set DPTR to current look-up table origin

if yarn is not broken, stay in idel and wait for valid cone size count

Fig 9.1 Main Program Flowchart

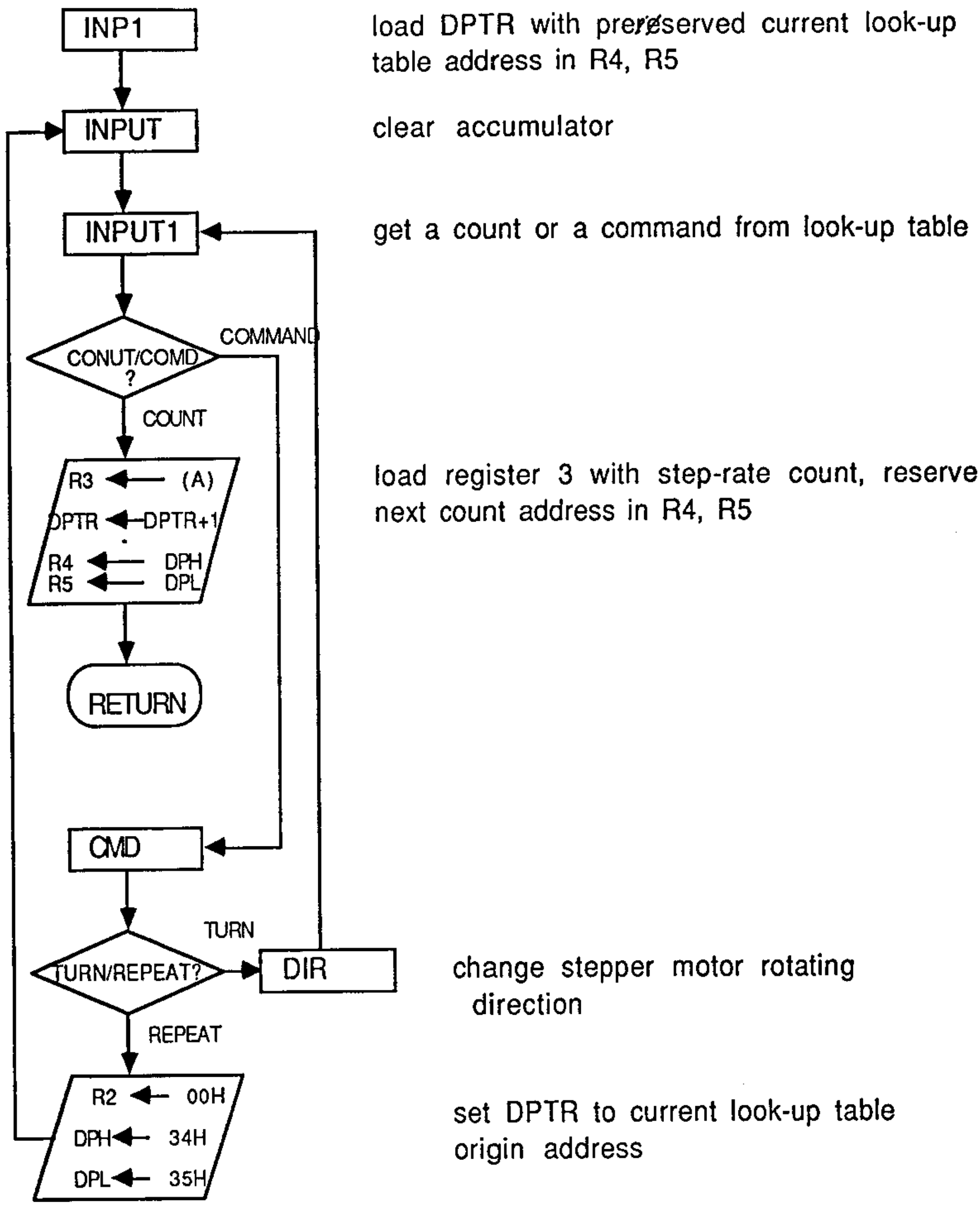
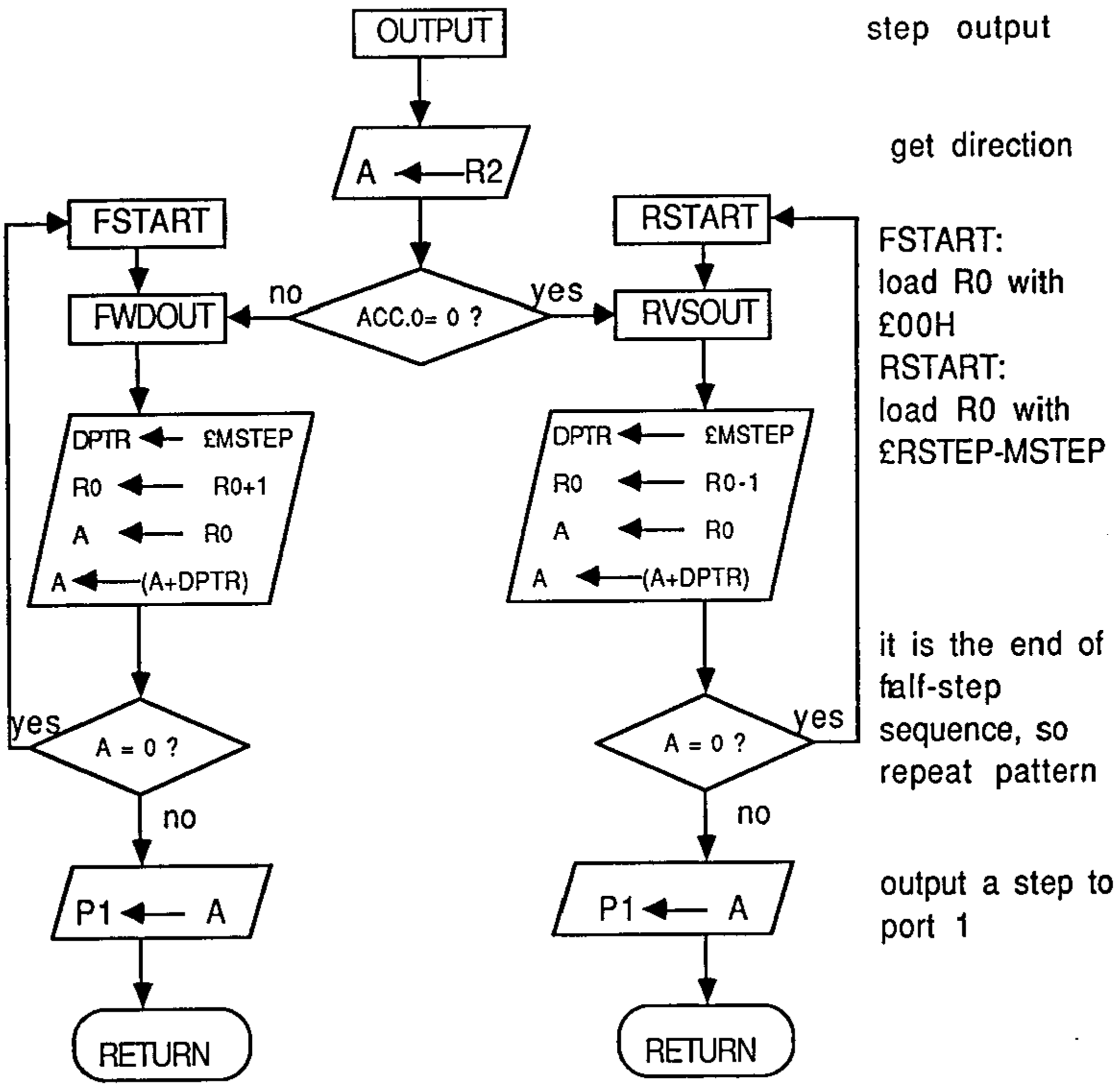
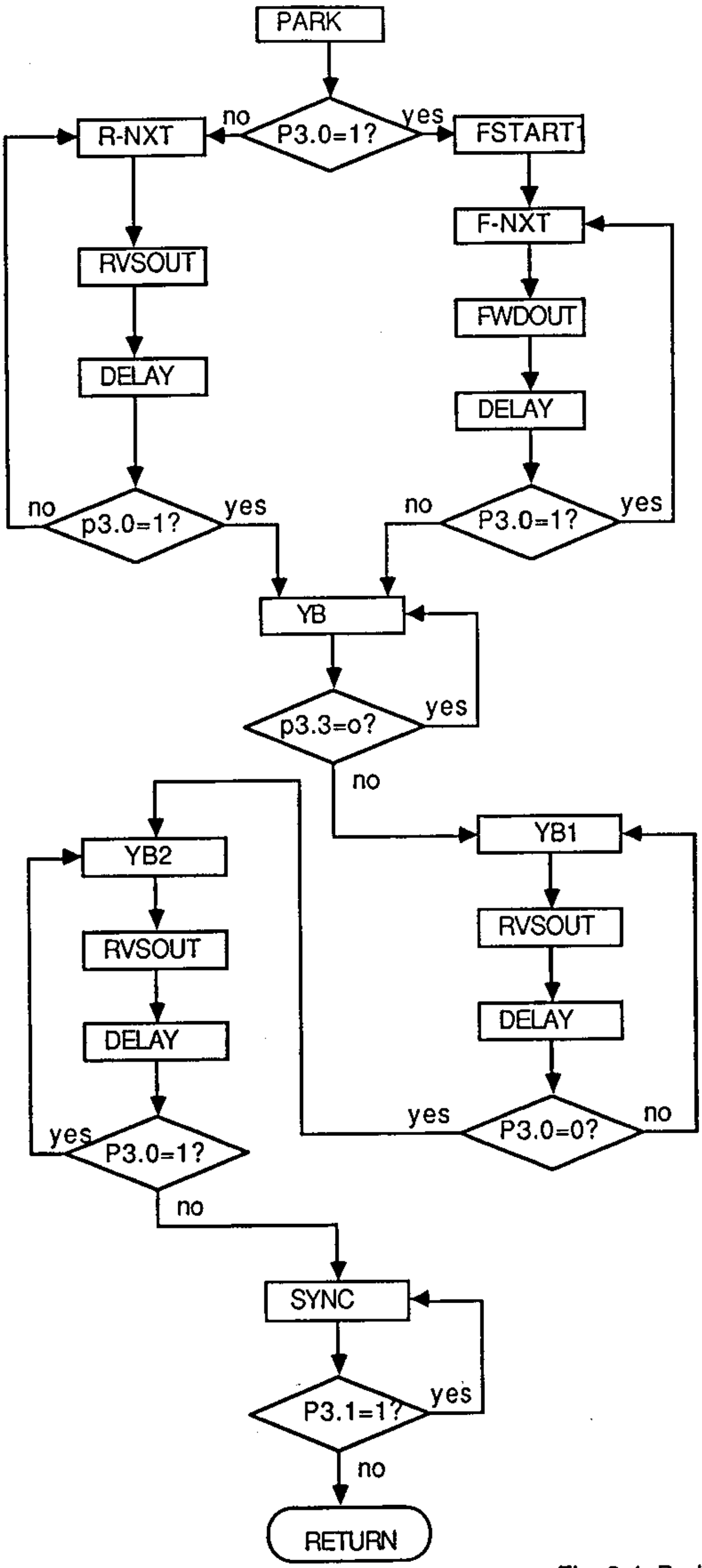


Fig 9.2 Input Routine Flowchart



This routine outputs a step forward or reverse according to the half-step sequence table

Fig 9.3 Output Routine Flowchart



step forward or reverse under software delay to park position

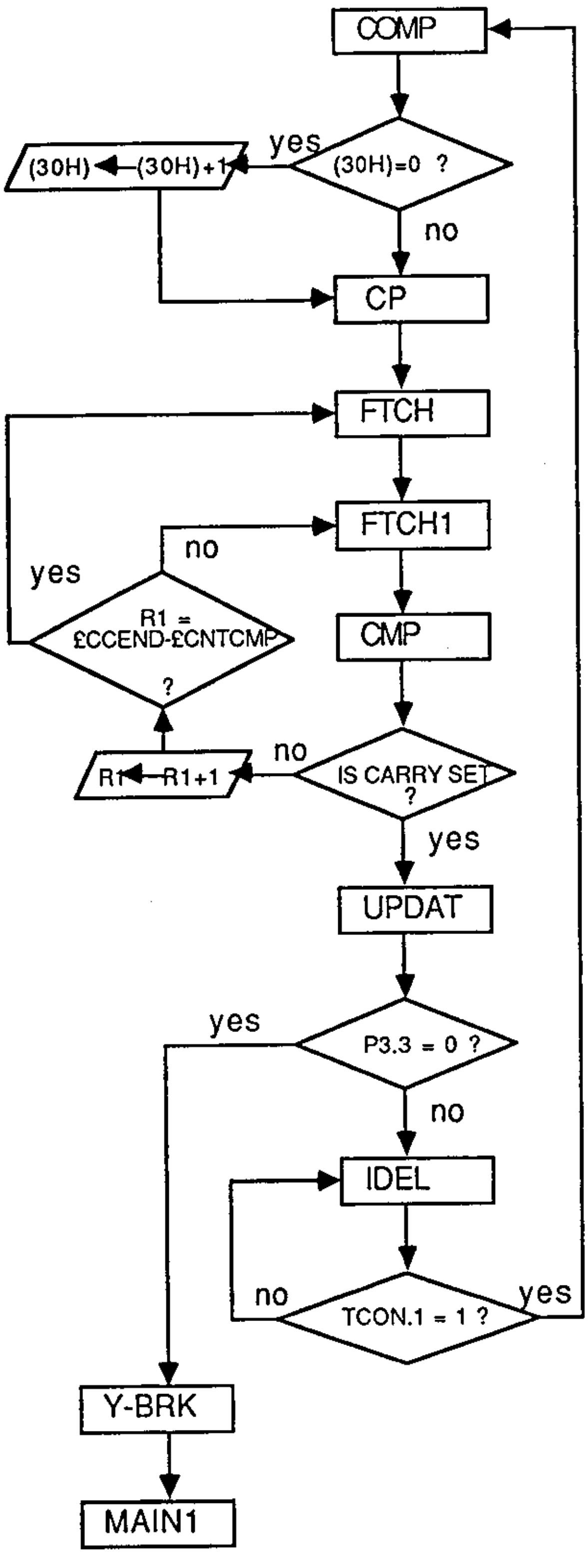
check if yarn is broken,so loop till yarn is pieced

bollard disc reverse to upside down positionand wait for synchronization

if bollard disc top point is black, then reverse to silver and keep reverse until black. if top point is silver,then reverse until black

wait for a falling edge from P3.1 to synchronize motion of traverse guide

Fig 9.4 Park Routine Flowchart



disable external interrupt 0

copy Timer 0 to R6, R7

clear R1

load look-up table entry count to RAM address 32H, 33H

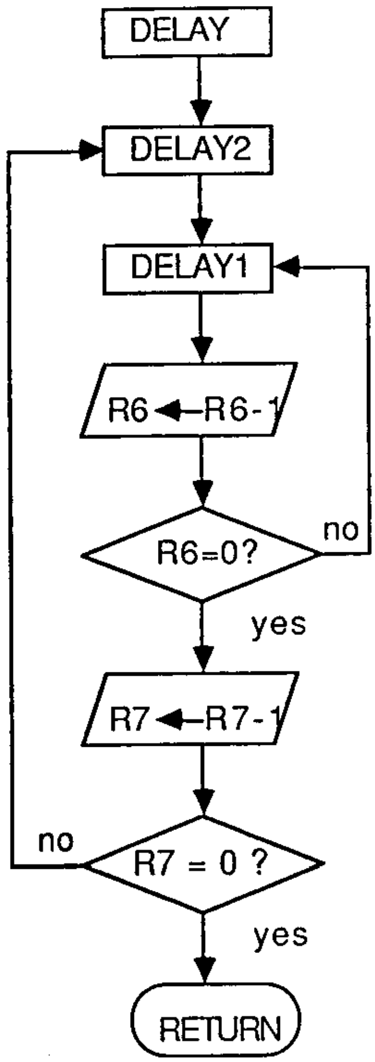
add latest cone size count to the look-up table entry count

if carry is not set, check if there is any look-up table entry count left, if carry is set, update look-up table and reserve its origin address in 34H, 35H

if yarn is broken (P3.3=0), clear interrupt enable and timer control register, set DPTR to the current look-up table origin go back to MAIN1 routine

if yarn is not broken, stay idel and wait for valid cone size count for further comparison

Fig 9.5 Cone Size Comparison Routine



set delay time for park routine

load R7 with £04H

load R6 with £FFH

Fig 9.6 Delay Routine Flowchart

9.1.3 Interrupts and Switch Interface

In this tension compensation programme there are several interrupt edge flags set or cleared by switches and two interrupt service routines which keep the compensator working properly.

During operation, an external event may signal the microprocessor to suspend the execution of its current programme and to execute instead a different set of instructions, that is called an interrupt .

IEO_SR is a cone size interrupt routine. For each revolution of the cone package, this routine loads terminal count value from the Timer 0 into the RAM address 30H, 31H for later use in the COMP routine, this count value is used to compare with the look-up table entry count and help to choose the corresponding look-up table. This interrupt occurs when interrupt edge flag TCON.1 is set by the cone size sensor.

TF1_SR is the step rate interrupt routine. This routine loads step rate count from register R3 into the low byte of Counter 1, the high byte is always FFH. when the Counter 1 carry flag is set, it outputs a step instruction, afterwards it gets the next step rate count into R3. This routine has the first priority. So, the stepper motor will carry on until the end of the current look-up table before starting a new look-up table.

The switch interface is used to reflect the external events and feedback signals to the microcontroller system. There are a few switches used in the mechatronic tension compensator system.

1. Magnetic switch

As stated before, the yarn winding error is a function of traverse guide position. When the mechatronic tension compensator is used to eliminate the winding error. A magnetic switch is used to synchronise the motion of traverse guide and the stepper motor. In PARK routine, a falling edge of the signal from magnetic switch to the pin P3.1 will 'wake' the stepper motor from the park condition and 'execute' a look-up table to start compensation.

2. Optical switch

The optical switch is fitted behind the bollard disc and used to check the stepper motor (bollard disc) park position. When the stepper motor begins to make compensation, it always starts at a top centre position with the black half on the right and the silver half on the left.

3. Yarn break switch

The yarn break switch is used to imitate yarn break condition and then let the stepper motor stay at its parked position waiting for piece-up. In the 'Y-BRK' routine the system is reinitialised except the data pointer, which still points to the address of the origin of current look-up table. So, after yarn is pieced up, a proper look-up table is ready for reference.

4. Cone size sensor

A cone size sensor is supposed to fit in the conical package holding arm and enable the Timer 0 to get the variable cone size count. Because the winding error is also a function of cone size, the count is later used in routine 'COMP' for choosing proper look-up table. In the present experimentation , a frequency generator is used to take the place of this sensor.

In executing the compensation programme, the microprocessor spends most of its time in 'COMP' routine and carries on the two interrupt routines until a yarn break signal occurs. The yarn break signal interrupts the execution of the programme and the compensator goes to the park position waiting for the operator intervention.

9.1.4 Half-stepping Mode

In order to increase the number of detent positions of the stepper motor and therefore adjust the yarn supply rate more delicately, a half-stepping mode is introduced. Half stepping produces 0.9 degree step angle as opposed to 1.8 degree step angle produced in a full step mode. So, the stepper motor in half-stepping mode can make 400 steps in one revolution.

The half-stepping mode is obtained by arranging the logical input level from the 8031 microprocessor to the PBL3717 stepper motor control circuit.

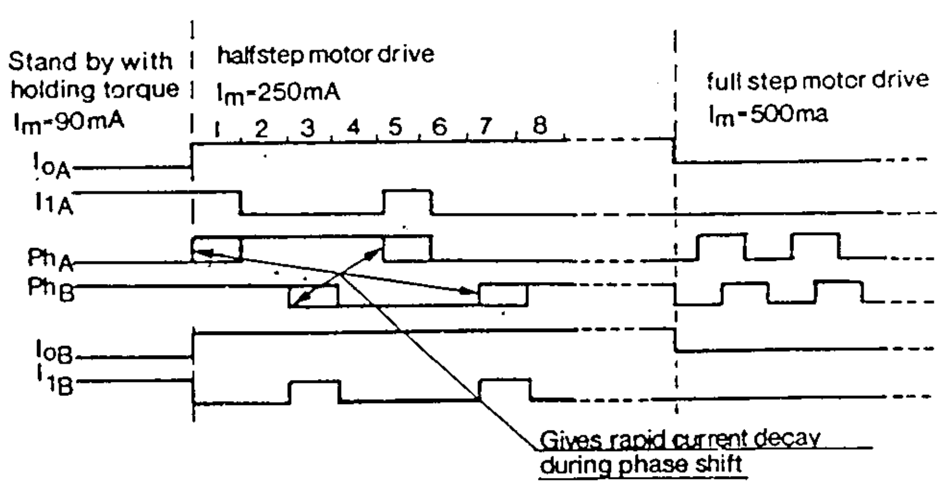


Fig 9.7 Stepper Motor Principle Operating Sequence

Fig.9.7 is the principal operating sequence for half stepping or full stepping mode. According to the pin connection of microcontroller 8031 and PBL 3717, the output logical level should be as stated in Table 9.1.

If the output routine loads port1 with a sequence of the above logical level, the stepper motor will rotate forward (CCW) in a half-stepping mode. If the reverse sequence is loaded, then the stepper motor rotates backward (CW).

PBL 3717 pin: Port 1 pin:	IB p1.5	OB p1.4	PhB p1.3	IA p1.2	OA p1.1	PhA p1.0	
Logical level:	0	1	1	1	1	0	1EH
	0	1	1	0	1	1	1BH
	1	1	1	0	1	1	3BH
	0	1	0	0	1	1	13H
	0	1	0	1	1	1	17H
	0	1	0	0	1	0	12H
	1	1	0	0	1	0	32H
	0	1	1	0	1	0	1AH

Table 9.1 Output Logical Level

9.2 Look-up Table Programme

9.2.1 Introduction

In the mechatronic tension compensator design, the stepper motor and its bollard disc work as a compensation mechanism. If in a finite time period the extra amount of yarn taken by compensator is equal to the yarn winding error over the same time period, the compensation works, if in the whole cone winding process, the yarn supply matches the yarn demand, then the compensator works successfully. In order to achieve this, there are two questions which should be considered carefully.

First, for each step, how much yarn is held by the compensator? Second, in order to match the yarn winding error what step rate should the compensator take?

The first problem is a geometric calculation and is solved by the programme 'ECLENGTH' which calculates yarn length taken by each step of the compensator. The second problem is solved by the program 'LOOKTABLE' which calculates step rate counts. The microcontroller uses these counts to control the step rate.

Since the information about the cam profile is a non-continuous, non-regular data file, and the yarn winding error is calculated based on this data file, a linear approximation programme has been written to find intermediate points on the yarn winding error curve. The time intervals of these points are set equal. The total number of the points we are concerned is 3250 which corresponds to the camshaft encoder counts for one complete cycle of guide motion. These counts are used by the microcontroller as the step rate counts to control the step rate, therefore to control the yarn supply rate to the winding unit.

9.2.2 Yarn Length Stored in the Compensator

The yarn length held in the compensator depends on the position of the bollard disc. If the bollard disc rotates in a counter-clockwise direction, more yarn will be held in the compensator. The bollard disc is centre symmetric, so only half of it need to be investigated. The amount of yarn held in the compensator can be calculated according to the Fig9.8.

where

$$o1c = r_c * \sin \o_3 \qquad (9.1)$$

$$oc = r_c * \cos \o_3 \qquad (9.2)$$

$$o1a = \mathrm{sqrt}\ (r_c * r_c - r_b * r_b) \qquad (9.3)$$

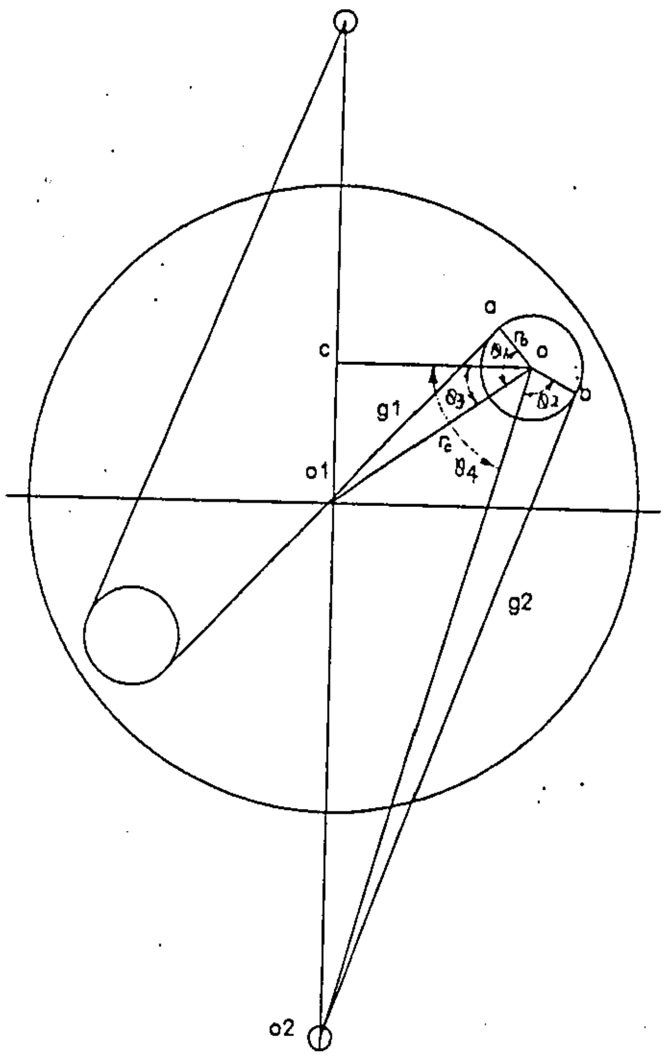


Fig 9.8 Yarn Length Held in the Compensator

$$o2c = o1o2 + o1c \quad (9.4)$$

$$o2o = \text{sqrt}\,(\,oc^{*}\,oc + (\,o1c + o1o2\,)\,^{*}\,(\,o1c + o1o2\,)) \quad (9.5)$$

$$o2b = \text{sqrt}\,(\,o2o^{*}o2o - r_b\,^{*}\,r_b) \quad (9.6)$$

$$\varnothing_1 = \arccos\,(\,r_b\,/\,r_c\,) \quad (9.7)$$

$$\varnothing_2 = \arccos\,(\,r_b\,/\,o2o\,) \quad (9.8)$$

$$\varnothing_3 = \arcsin\,(\,o1c\,/\,r_c\,) \quad (9.9)$$

$$\varnothing_4 = \arcsin\,(\,o2c\,/\,o2o\,) \quad (9.10)$$

$$\varnothing_5 = 2^{*}\,\pi - (\,\varnothing_1 + \varnothing_2 - \varnothing_3 + \varnothing_4) \quad (9.11)$$

$$g_4 = \varnothing_5\,^{*}\,r_b \quad (9.12)$$

$$g1 = o1a \quad (9.13)$$

$$g2 = o2b \quad (9.14)$$

where $\varnothing_5$ is the wrap angle of the bollard

In the right half of the tension compensator, the yarn length is

$$g8 = g1 + g4 + g2 \quad (9.15)$$

In the other half, the yarn length g9 can be calculated in a similar way. Then, the total yarn length held in the compensator is

$$g10 = g8 + g9 \quad (9.16)$$

if g0 is the yarn length held in the compensator when the bollard disc is at its start position ($\varnothing=0$), then the extra amount of yarn taken by rotation through angle $\varnothing$ is

$$g_t = g10 - g0 \quad (9.17)$$

Fig9.9 shows the yarn take-up length g_t versus the bollard disc angular displacement $\varnothing$. It can be seen that the yarn length g_t is almost linear to $\varnothing$, if the angular displacement $\varnothing$ is within 45 degrees.

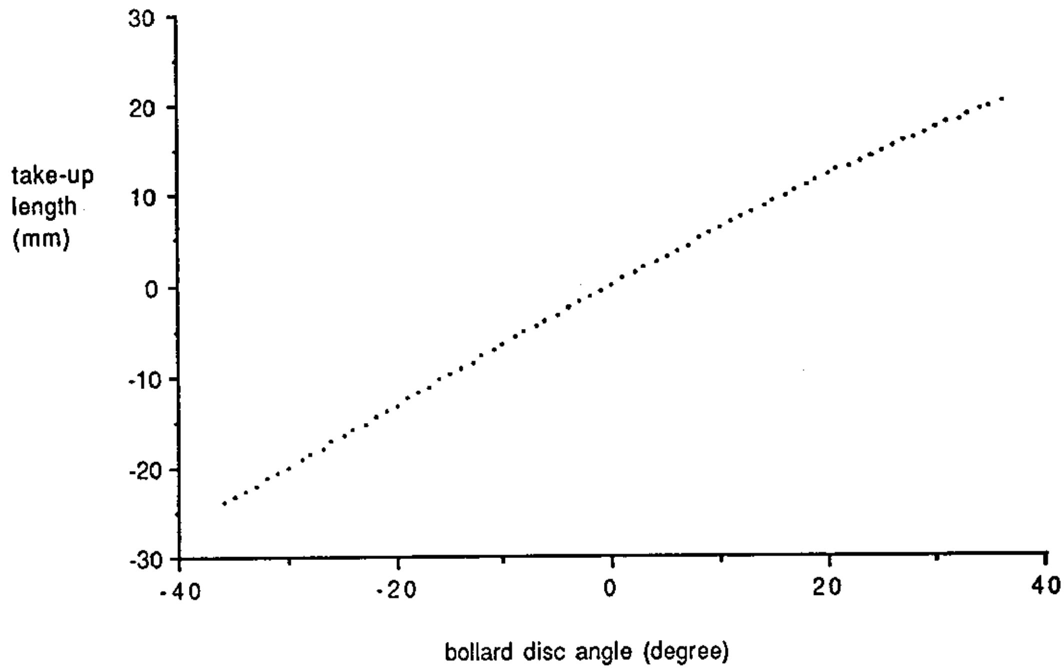


Fig 9.9 Yarn Take-up Length

9.2.3 Approximation of the Yarn Winding Error

The analysis of yarn winding error has indicated that the difference between the yarn demand and supply contributes to the yarn winding tension variation.

The look-up table programme is written to calculate the step-rate-count by matching the yarn being taken in each step to the yarn winding error. Since the yarn being taken in each step is almost linearly related to the step motor rotating angle, linear approximation can be used to obtain the step rate count. The level part of yarn winding error curve corresponds to a large step-rate-count number, and a steep part to a small count number. A large count number means a low step rate, whereas a small number a high step rate.

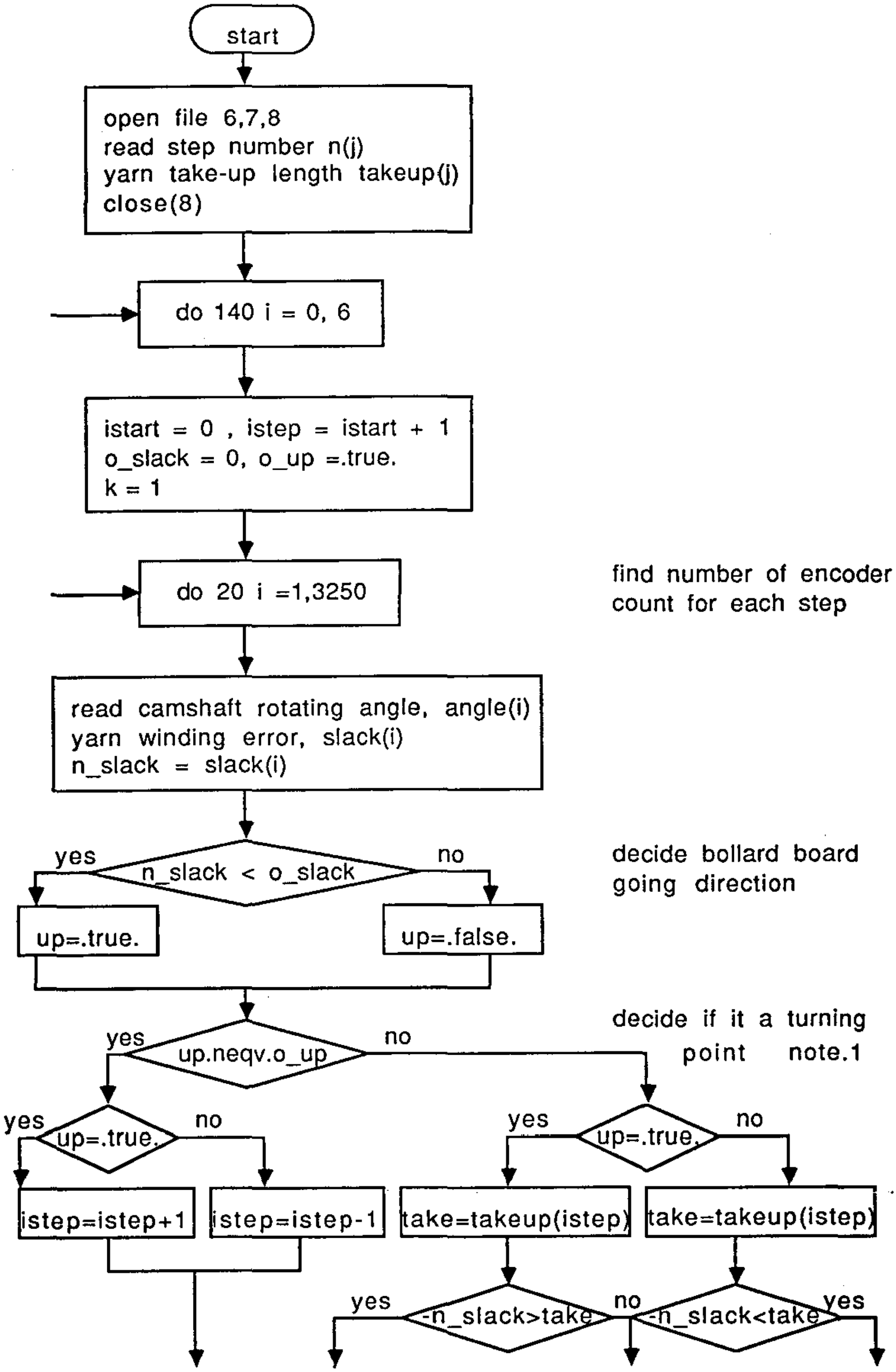
These count number are transferred to a 256-n form, since the counter in the microcontroller 8031 uses hexadecimal number system and is a count-up rather than count-down device.

9.2.4 Flowchart for the Look-up Table Programme

Fig 9.10 is a flowchart used to calculate the look-up tables for the 8031 microcontroller to control the motion of the stepper motor.

First, it decides the stepper motor rotating direction, weither to take up or release some yarn. Second, it matches the extra yarn taken by each step to the corresponding winding error. Third, it calculates the counts for each step. Fourth, it transfers the count number into 256-n form. Finally, it calculate the entries for each look-up table.

The stepper motor used in this mechatronic compensator has a high resolution, so the difference between the first half cycle and the second half cycle of yarn winding error can be reflected in the look-up tables. This certainly improves the compensation work. The look-up table programme and the look-up tables are listed in the appendix.


start
open file 6,7,8
read step number n(j)
yarn take-up length takeup(j)
close(8)
do 140 i = 0, 6
istart = 0 , istep = istart + 1
o_slack = 0, o_up =.true.
k = 1
do 20 i =1,3250
find number of encoder count for each step
read camshaft rotating angle, angle(i)
yarn winding error, slack(i)
n_slack = slack(i)
yes
n_slack < o_slack
no
decide bollard board going direction
up=.true.
up=.false.
decide if it a turning point note.1
yes
up.neqv.o_up
no
yes
up=.true.
no
yes
up=.true.
no
istep=istep+1
istep=istep-1
take=takeup(istep)
take=takeup(istep)
yes
-n_slack>take
no
-n_slack<take
yes

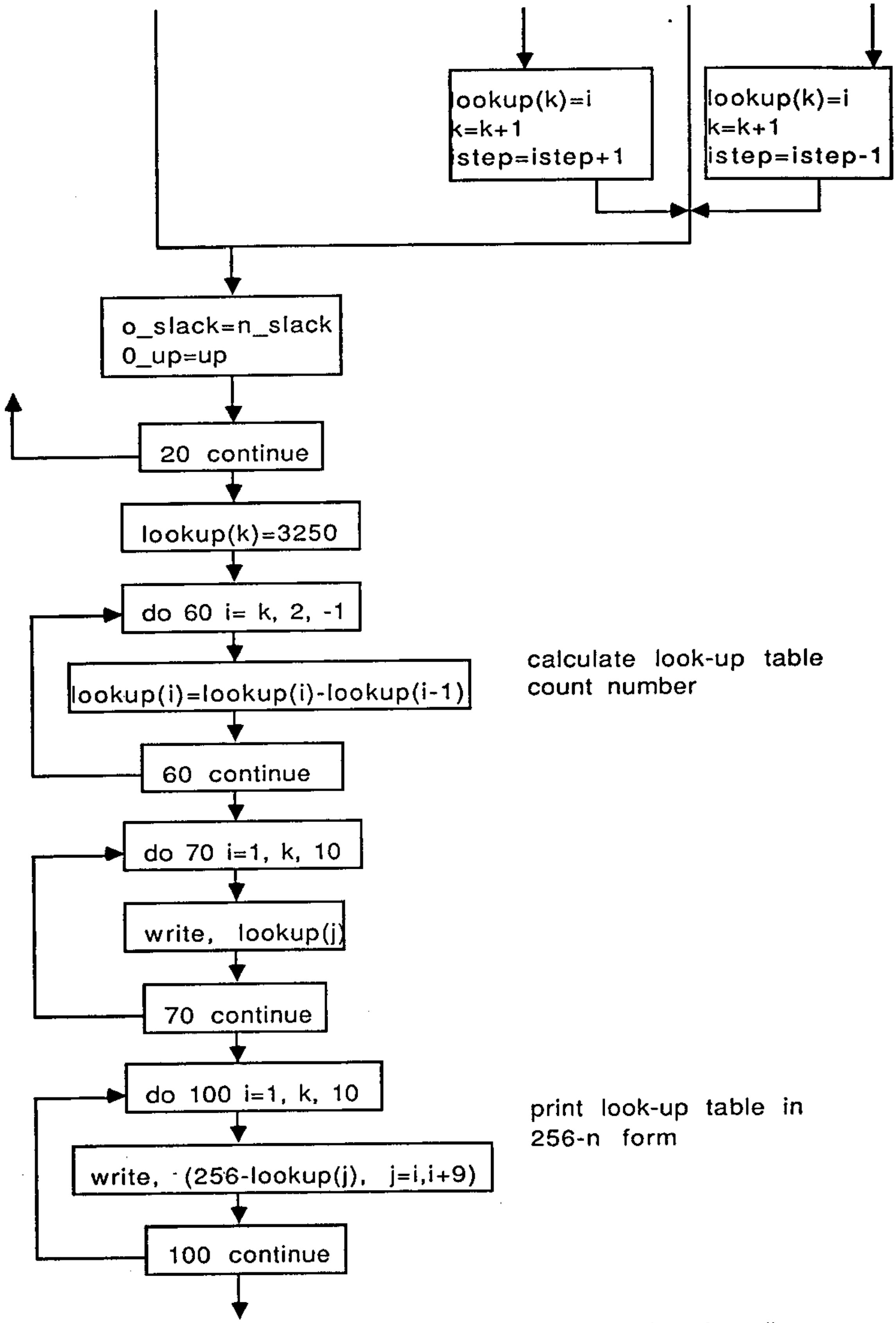
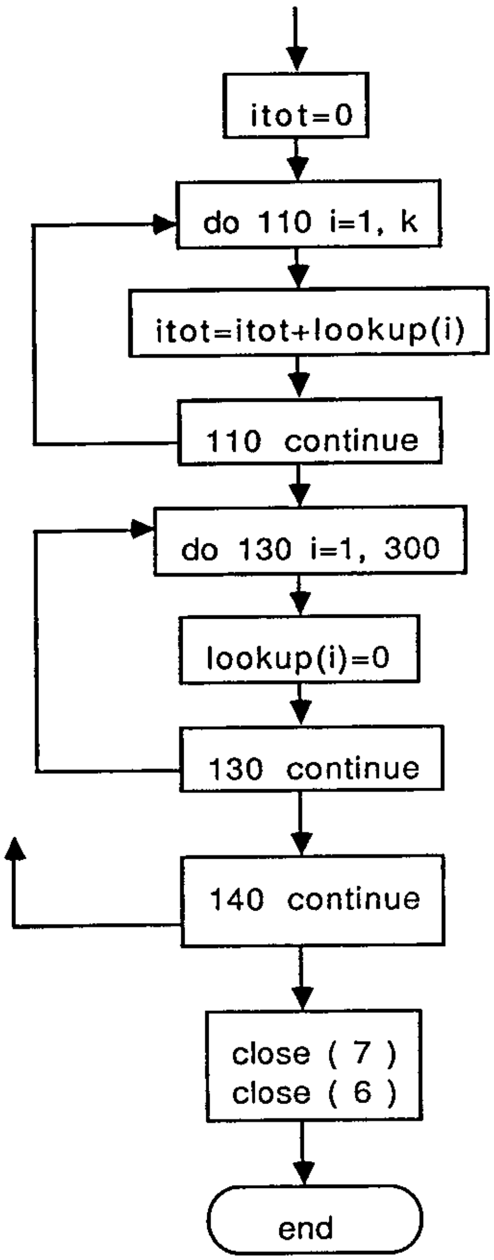


Fig 9.10 Look-up Table Program Flowchart (continued)



calculate total look-up table entries

clear lookup(i)

Note.1 If it is a turning point, decide a step go forward or backward

If it is not a turning point, decide whether it needs to take one more step or not.

Fig 9.10 Look-up Table Programme Flowchart (continued)

CHAPTER 10 WINDING TENSION EXPERIMENT

10.1 Introduction

In the previous chapters, a mathematical model for the yarn winding process has been formulated to describe the yarn winding error. On the basis of the yarn winding error analysis and the calculation of yarn length held in the compensator, look-up tables have been calculated. In order to compensate the yarn winding error which is the main cause of the variation of yarn winding tension; the mechatronic compensator follows these look-up tables to take up or release some yarn wherever is necessary.

It is of primary importance to examine how accurate this mathematical model is and how well the mechatronic tension compensator works by following these look-up tables in the high speed cone winding process. In anticipation of this investigation, a test rig was constructed. Essentially, it consists of yarn winding machine, tension compensator and the tension measuring equipments. fig 10.1 gives the general view of the test rig.

10.2 Winding Tension Test Rig

10.2.1 Spinning-winding Tester

The yarn winding machine used in the the tension experiment is provided by the Platt Saco Lowell Co. Ltd. This is a 4-position spinning-winding tester, the main function of which has been introduced in chapter 1. The maximum yarn delivery speed of this tester is 400 m/min.

There are two small modifications on the winding condition in this experiment. One is: in the Masterspinner, the yarn is produced in the spinning unit, then it is delivered into the winding unit and wound onto a package there. However, in this tension experiment, yarn comes directly from a ready-made package instead of the spinning unit. Since yarn must pass a yarn delivery roller first before it goes into the winding unit, it is the speed of yarn delivery roller that decides the yarn supply rate to the winding unit. So this modification should not affect the results of the tension experiment. The other modification is: in accordance with the assumption which has been set in the mathematical modelling of yarn winding process, the variable belt drive in this spinning-winding tester is dismounted. As described in the chapter 2, this only very slightly affects the yarn winding velocity and neglecting this should not substantially affect the experiment results.

10.2.2 Mechatronic Compensator

The mechatronic tension compensator has been constructed in the Department of Mechanical Engineering at Loughborough and has been introduced in chapter 8. In this mechatronic tension compensator (Fig 10.2), an EPROM emulator Softy-3 (1) is used in place of the system memory 27512 EPROM. The reason for this is that during the tension experiment, the compensation programme may need to be amended to deal with any unforeseeable problems or simply to improve the performance of the compensator(2). The Softy-3 can store the programme in its own memory and emulate the 27512 EPROM, so, the compensation programme and the look-up tables do not need to be 'burnt' into the 27512 EPROM before they are finally set down. This provides great convenience for the programme tuning. A IBM-PC is used here as a programme editor.

According to the original design of the mechatronic compensator, it is the package size sensor that should count the time for each revolution of package and then transfer this information to the microcontroller for selecting corresponding look-up table. Here a frequency generator(3) generates the count to imitate the package size sensor. This modification makes no difference to the microcontroller but provides the freedom of selecting different look-up tables during testing.

10.2.3 Tension Measuring Devices

Fig 10.3 is a picture of the tension measuring and recording equipment. A Rothchild A150 tension probe(1) is fitted on the frame between the top guide of compensator and the centre of the curved bar, this tension device picks up the tension variation in the yarn and converts it into an electronic signal. This signal is then magnified by the electronic Tensiometer(2) and output to a HM208 digital storage oscilloscope(3). The signal can then either be shown on the screen or be stored in the memory of the oscilloscope and output to a plotter(4) later on.

10.3 Results of Tension Experiment

During the yarn winding tension experiment, a mechatronic tension compensator and a spring compensator have been used. Due to the difficulties of fitting another shaft in the spinning-winding tester, the mechanical tension compensator has not been used. A series of tension experiments have been carried out under different winding conditions, such as different winding velocity, different package sizes and using different yarn distribution bars. These winding conditions are listed in the table 10.1.

In the tension records, the X-axis represents the traverse guide position. The square pulse indicates the guide is at the small end of the package. The Y-axis represents the tension in the yarn, 15 mm in the Y-axis equals 10 grams tensile force in the yarn. Only fig 10.8 and 10.9 are exceptions. They are obtained in a 2.5 scale. That means 15 mm in their Y-axis equals to 25 grams.

figure number	winding velocity	package size	distribution bar	tension compensator
10.4	150 m/min	20 mm	straight bar	no
10.5	150	40	straight bar	no
10.6a	150	20	curved bar	no
10.7	150	40	curved bar	no
10.8a	150	6	curved bar	no
10.8b	150	6	curved bar	spring
10.9a	250	6	curved bar	spring
10.9b	400	6	curved bar	spring
10.10a	150	15	curved bar	no
10.10b	150	15	curved bar	mechatronic
10.11a	400	15	curved bar	no
10.11b	400	15	curved bar	mechatronic
10.12	150	40	curved bar	mechatronic

Table 10.1 Winding Conditions

10.3.1 Tension Experiment without Compensator

Some results of the tension experiment have already been shown in chapters 3 and 4, and some useful conclusions have been drawn from them. These experiments and the conclusions are briefly described below:

(1) Tension experiment with different distribution bar

The Tension experiment record shown in fig10.4 indicates that the yarn path length variation contributes to the tension variation in the yarn, especially when yarn is wound to the two ends of the package. According to the calculation in chapter 4, the yarn path length varies from 280mm to 287mm, when the traverse guide moves from the middle to the end of the package. In small cone winding, because the amplitude of yarn winding error is three times bigger than that of the path length variation, the tension variation due to the yarn path length variation is not very significant. As package size increases, the winding error decreases quickly, but the yarn path length variation remains the same. At that time, it begins to play an important role in the tension variation. This can be seen by comparing fig.10.4 and 10.5.

Fig 10.6a and 10.7 are obtained under the same winding conditions as that in fig 10.4 and 10.5 but with a curved yarn distribution bar. It can be seen that a curved bar reduces yarn path length variation therefore reduces winding tension variation, especially at the two ends of package. By employing a curve bar, the analysis of winding tension experiment can be made simpler, only the winding error needs to be considered.

Compared fig 10.6a with fig 10.6b, which is a computer simulation of the yarn winding error, it is seen that the tension variation is almost the same as is predicted in the computer simulation. This proves that the mathematical model of winding process and winding error are quite accurate.

(2) Tension experiments at different winding velocity

The mathematical model for the yarn winding process and the yarn winding error in chapter 2 have predicted that the winding error is not a function of the winding velocity. Theoretically, therefore, the winding velocity should not affect the winding tension variation. The tension experiments have further confirmed this conclusion. Figs 10.10a and 10.11a show two tension curves; they are obtained under the same winding condition except at the different winding velocity. Even though the winding velocity in fig 10.11a is about three times as high as that in the fig 10.10a, the amplitude of winding tension variation is almost the same. Based on this conclusion, the winding tension experiment can be carried out at a low winding speed. This also simplifies the work of the tension experiment.

(3) Tension experiment with different package size

The mathematical model for cone winding in chapter 2 and the yarn winding error computer simulation in chapter 3 have predicted that the package size greatly affects the yarn winding tension variation. Figs10.6a, 10.7 and 10.10a are also tension experiment records, they are obtained under the same winding conditions but at different package size. The winding tension variation is much bigger in a small cone winding than that in a big one. The tension experiments show that the amplitude of the tension variation drops significantly in the whole cone winding process. When package size is bigger than 60mm, the winding tension variation is so small that actually a tension compensator is hardly needed. So, during tension compensation most attention should be given to small cone winding.

These tension experiments and their corresponding conclusions provided basis for further tension experiments.

10.3.2 Experiment with Tension Compensator

Further tension experiments have been carried out with a curved yarn distribution bar. Either a spring compensator or a mechatronic compensator has been used in different winding conditions. Fig10.8~10.12 show the experiment records.

(1) With spring tension compensator

A spring tension compensator is currently used in the Masterspinner, it consists mainly of a spring and a bollard disc. The yarn is wired over the pins on the bollard disc during winding. When tension in the yarn builds up, it rotates counterclockwise against the spring. By releasing some yarn, it eases the tension in the yarn. When tension drops, it rotates clockwise to take in some yarn. Fig10.8 has two tension curves, fig10.8b is with spring tension compensator whereas fig10.8a is not. It can be seen that when winding velocity is low, the spring compensator does reduce the amplitude of winding tension variation. It reduces the tension where tension builds up and it increases the tension where tension drops to zero. However, the tension experiment results also indicate that when the winding velocity increases, the remaining tension increases as well. fig10.9a shows that when the winding velocity reaches 250m/min., the remaining tension amplitude reaches the same level as that where no tension compensator is used. Fig10.9b indicates that when winding velocity reaches 400m/min., the tension amplitude is twice as much as that in 150m/min. The tension curve goes up and down very sharply, implies the violent tension variation in the yarn. The tension experiment results indicate that the spring tension compensator can not work properly in the high speed cone winding.

(2) With mechatronic compensator

A mechatronic tension compensator is fitted on the spinning winding tester. The pulses from the cam shaft encoder are used by the mechatronic compensator as the bases of step-rate count to cope with the winding velocity. Several look-up tables are provided to deal with different package size. The tension experiment results are shown in fig10.10~10.12.

Both fig10.10 and 10.11 have two tension curves, this arrangement is for the convenience of comparison. Curve a is obtained with no compensator, whereas curve b is with mechatronic compensator. It is noticed that:

(a) By using a mechatronic tension compensator, there is always some tension remaining in the yarn, no zero tension zone occurs during cone winding.

(b) the amplitude of winding tension variation can be reduced to one-third of its original level.

(c) The remaining tension variation does not increase with the increase of winding velocity.

The tension experiment results indicate that by using a mechatronic tension compensator in the cone winding, yarn slack can be avoided, the amplitude of tension variation can be reduced to an acceptable level and the compensation work can be done pretty well even in the high speed cone winding.

10.4 Remaining Tension Variation Analysis

By using a proper tension compensator, the variation of yarn winding tension can be significantly reduced. However, no matter how well the compensator works, there is still some tension variation left. Analysing this remaining tension variation can help understanding the problem of winding tension variation and find way to further improve the performance of the mechatronic compensator.

(a) remaining yarn path length variation

Figs10.6a and 10.7 are tension experiment records obtained by using a curved yarn distribution bar. It can be seen that at both ends of the package tension rises, especially in fig10.7, where a big size package is being wound. There are two possible reasons for that. One is the error in the manufacturing and fitting the curved bar. This mechanical error still causes some yarn path length variation. This effect in a big package is larger than that in a small package. The other is friction at the yarn-bar contact place. When traverse guide reaches the end of the package and starts to move back, the frictional force prevents the contact point from moving up to the position where it should be, and therefore changes the yarn path length.

(b) effect of winding velocity

Theoretically, the winding velocity should not affect the winding tension variation, but the increase of winding velocity increases the traverse guide inertia and changes the coefficient of friction between yarn and guides. These dynamic factors affect increase the winding tension a little as can be seen by comprising figs10.10a and 10.11a.

(c) higher frequency tension variation

In figs10.10 and 10.11, it is noticed that by using a mechatronic tension compensator, the amplitude of tension variation is reduced, but at the same time, the amplitude of higher frequency oscillation increases as well. The jagged tension curve is probably due to the unevenness of stepping mode and subsequent increase of friction variation between yarn and pins on the bollard disc.

(d) tension peak at the small end

Fig 10.12 is obtained by using a mechatronic compensator. It can be seen that a tension peak occurs when the traverse guide moves to the small end of package. If fig 10.12 is compared with fig 10.7 where no tension compensator is employed, it may be concluded that the tension peak there is the result that the excessive yarn is taken by the the mechatronic compensator.

From the tension experiment results and the remaining tension analysis, it can be seen that the remaining tension variation is relative small compared with the original one and those factors will not crucially affect the compensation work. Since the mechatronic tension compensator is very adaptable, there is a potential of further improving the compensation work by only modifying the look-up tables.

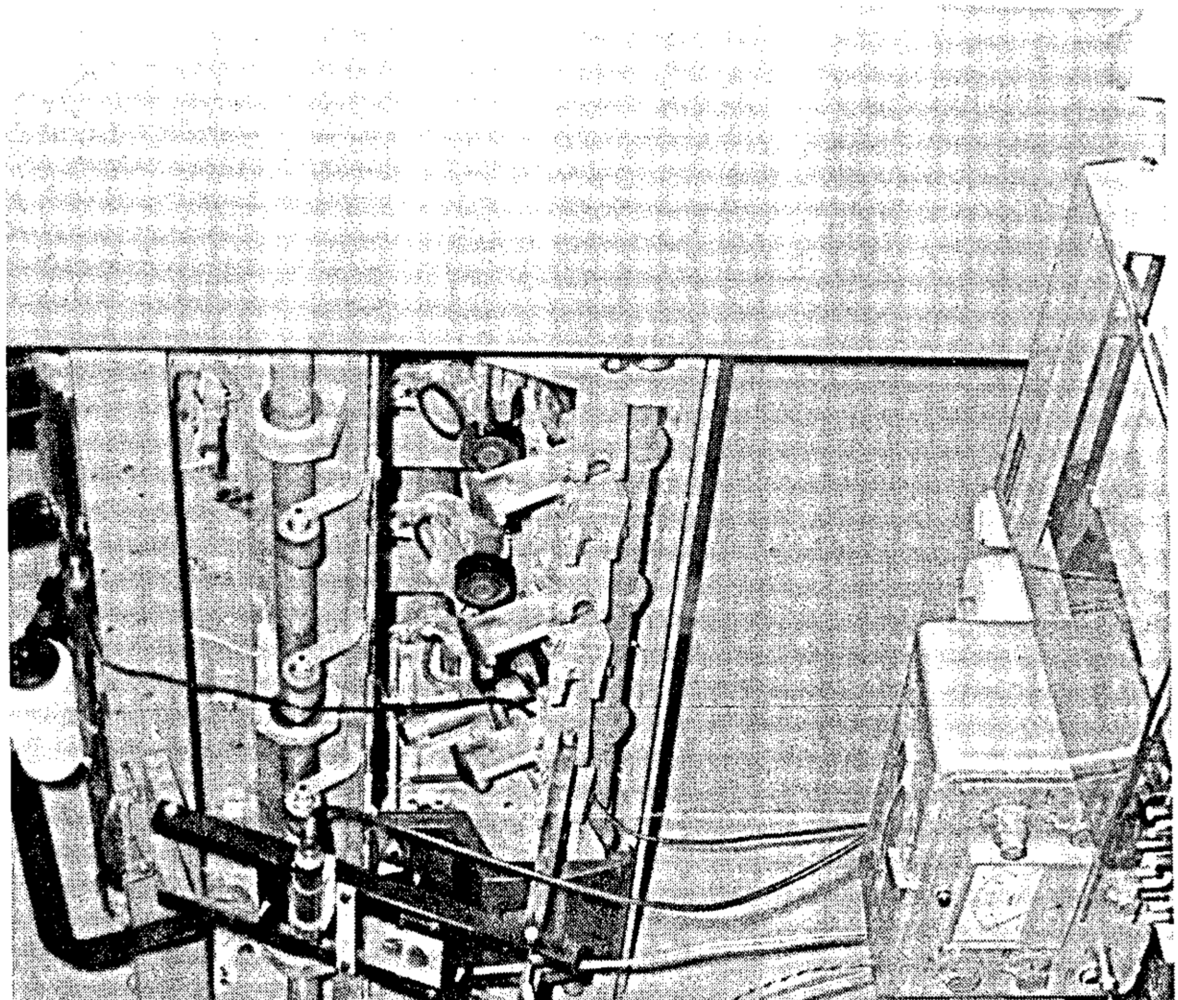


Fig. 10.1 Tension Test Rig

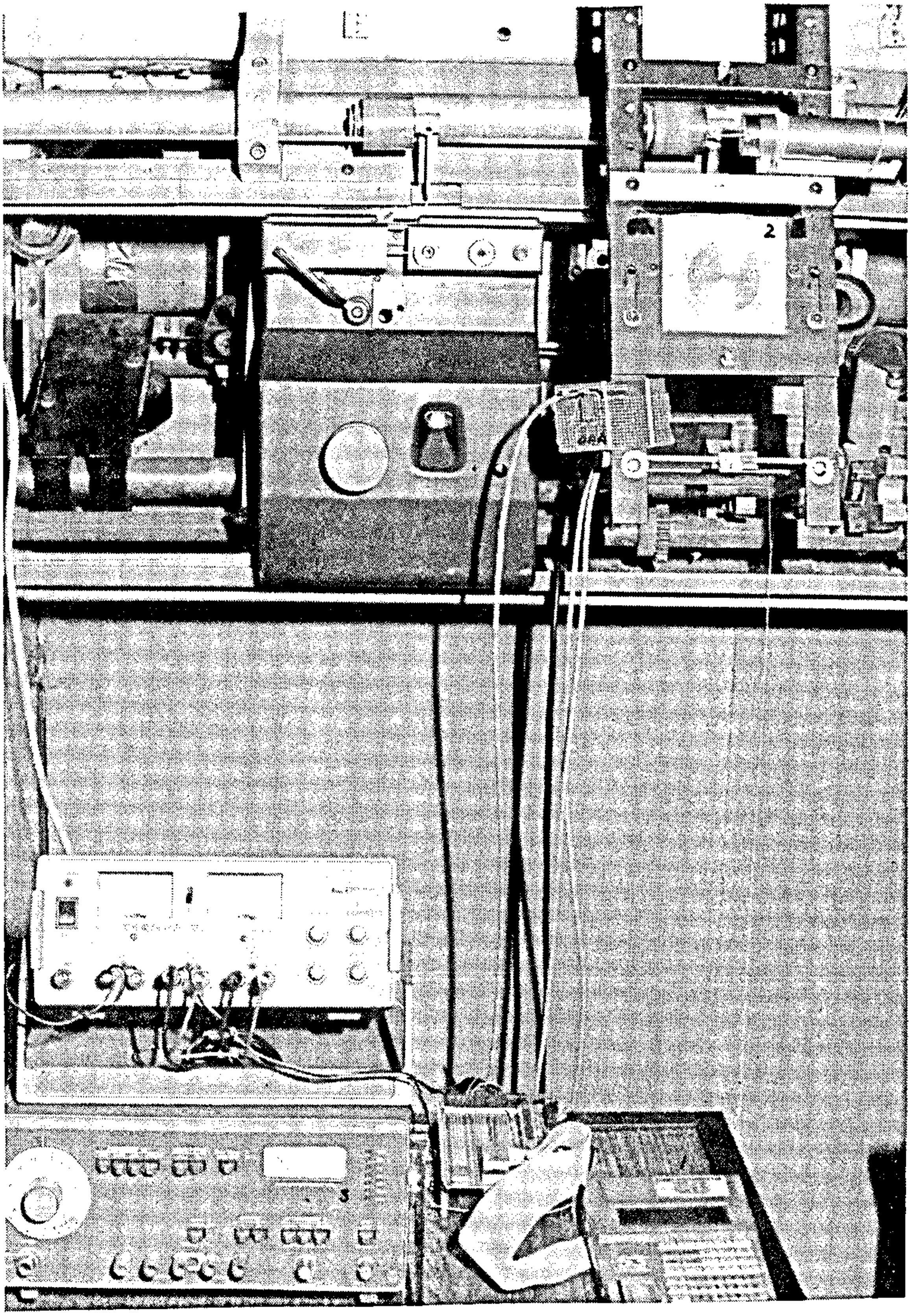


Fig. 10.2 Mechatronic Tension Compensator

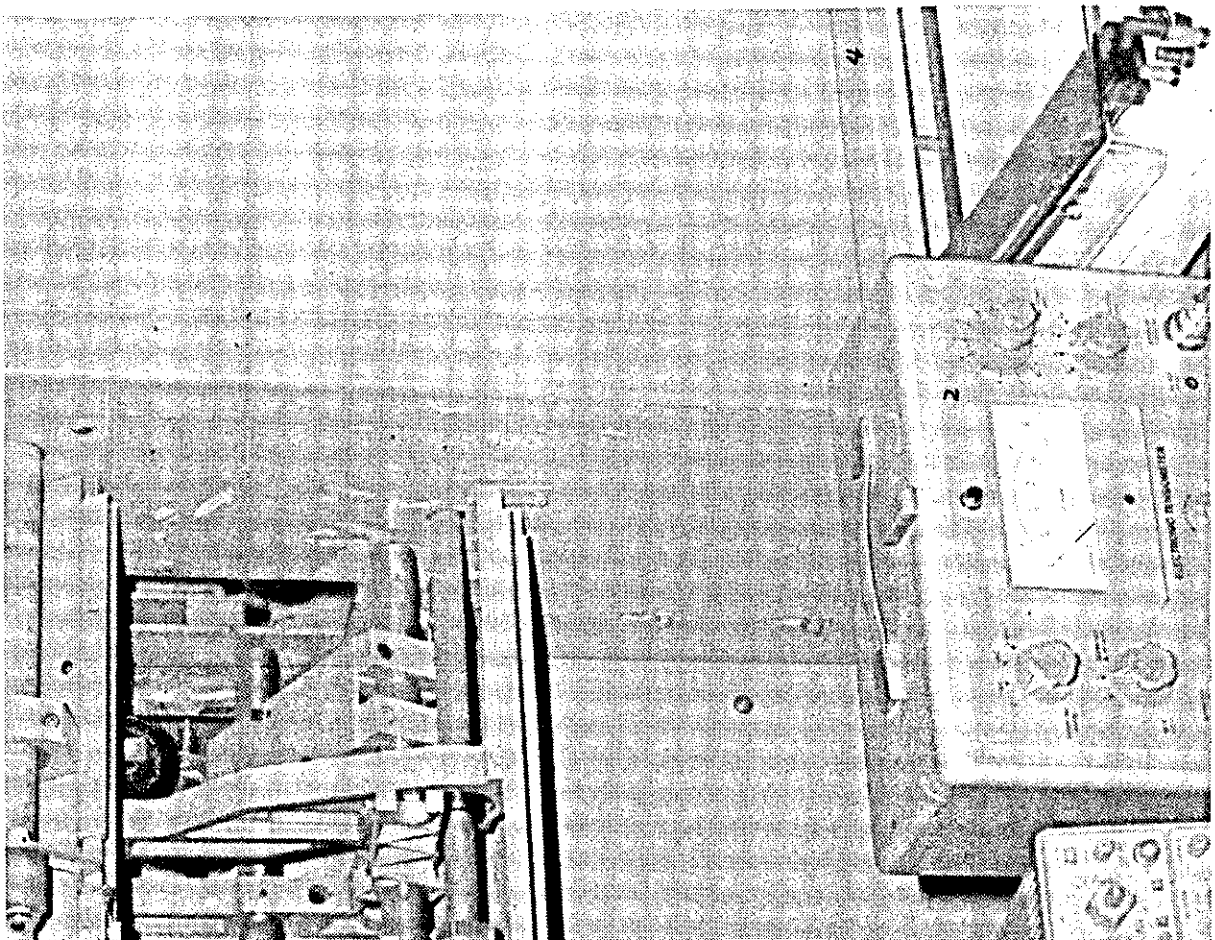


Fig. 10.3 Tension Measuring and Recording Devices

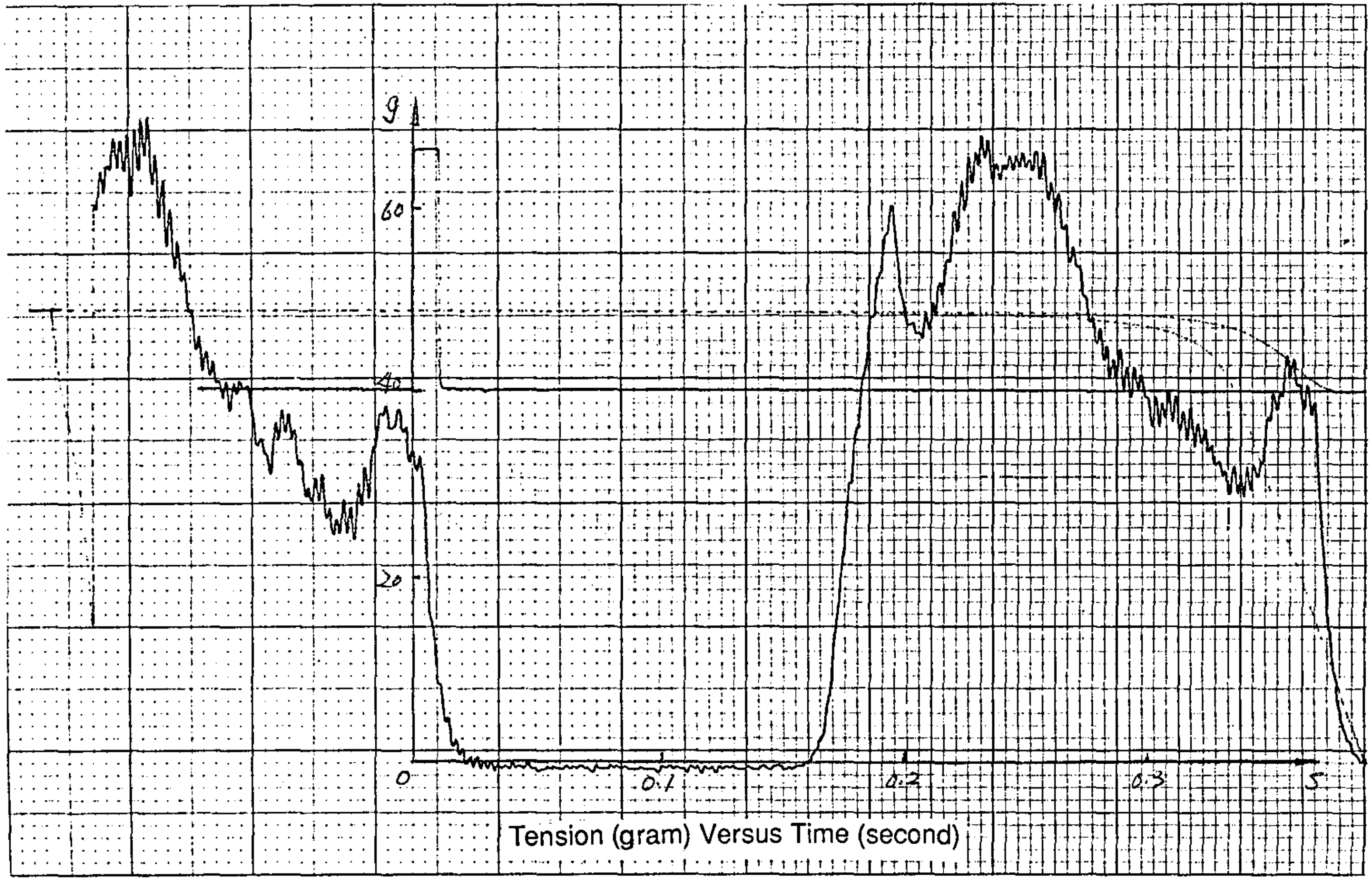


Fig 10.4 Tension Record 1

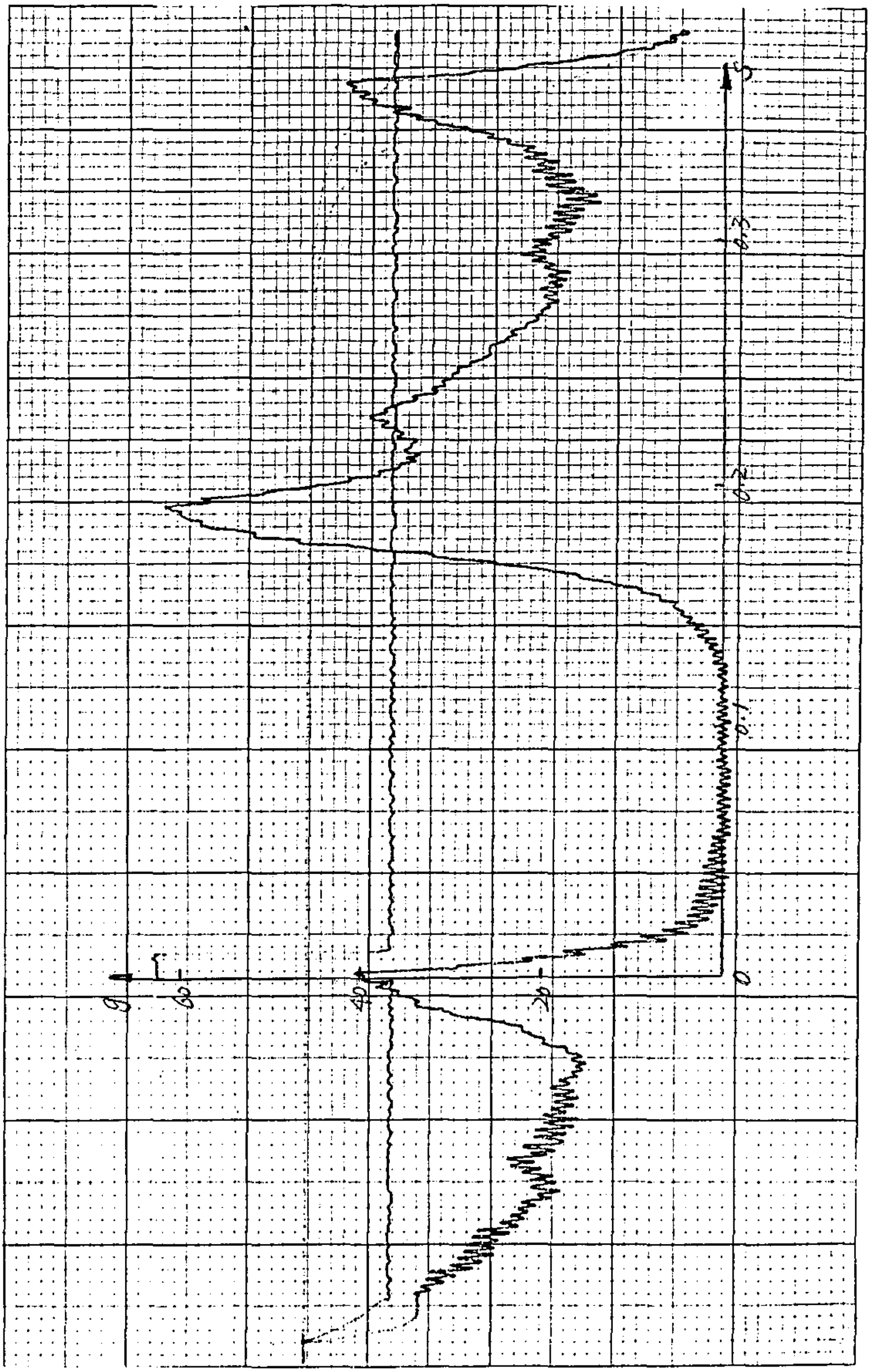


Fig 10.5 Tension Record 2

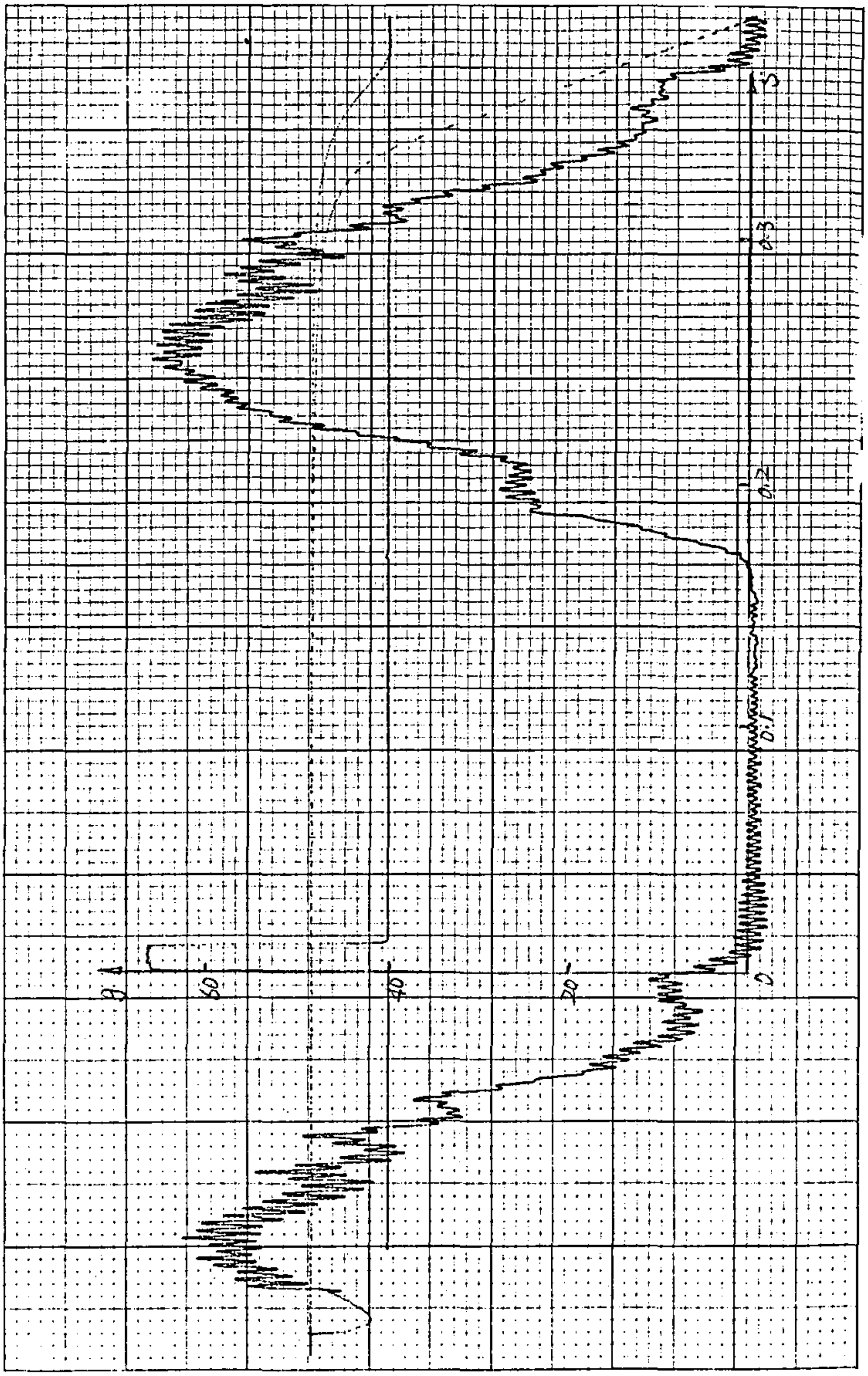


Fig 10.6a Tension Record 3

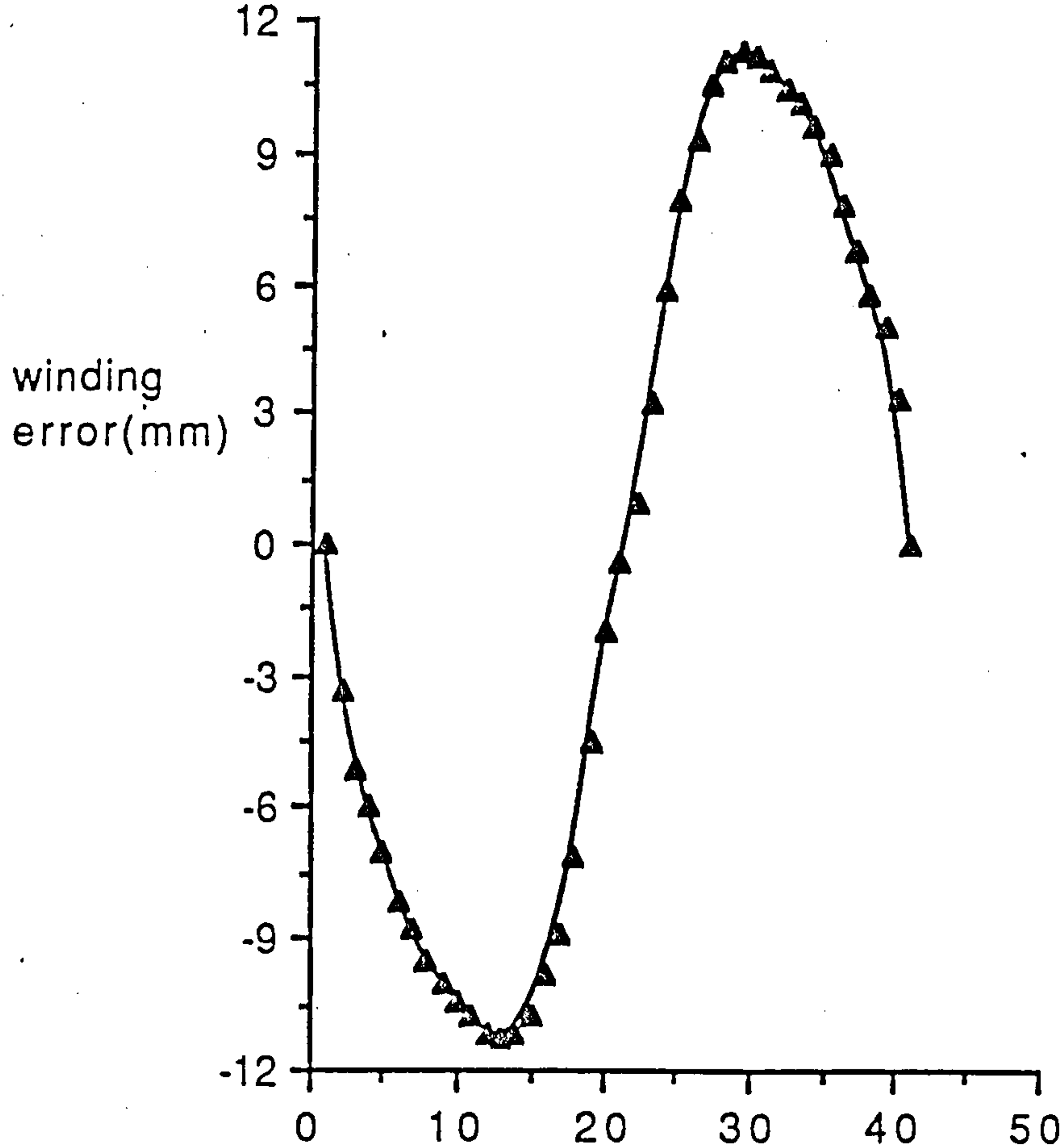


Fig 10.6b Winding Error Simulation

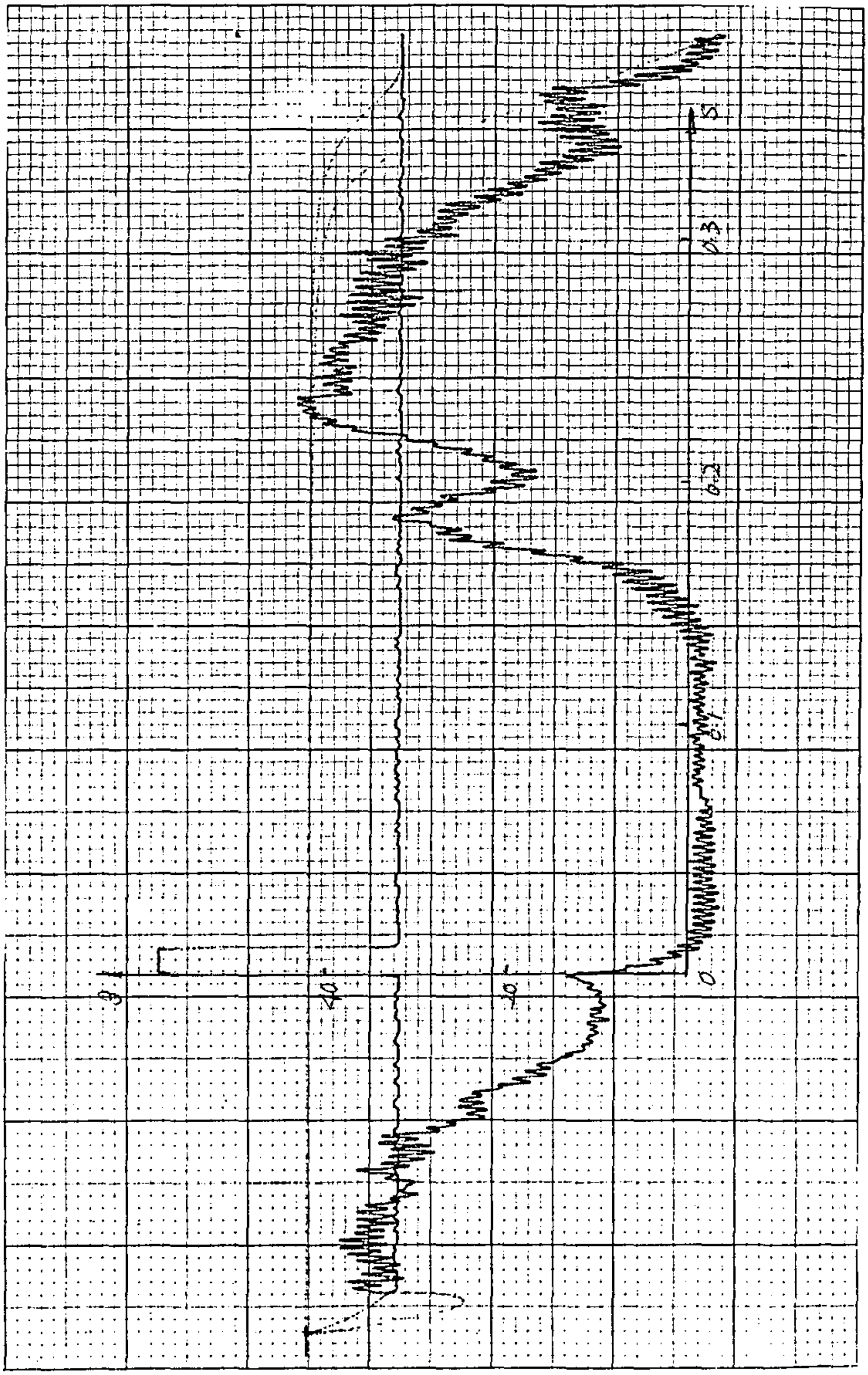


Fig 10.7 Tension Record 4

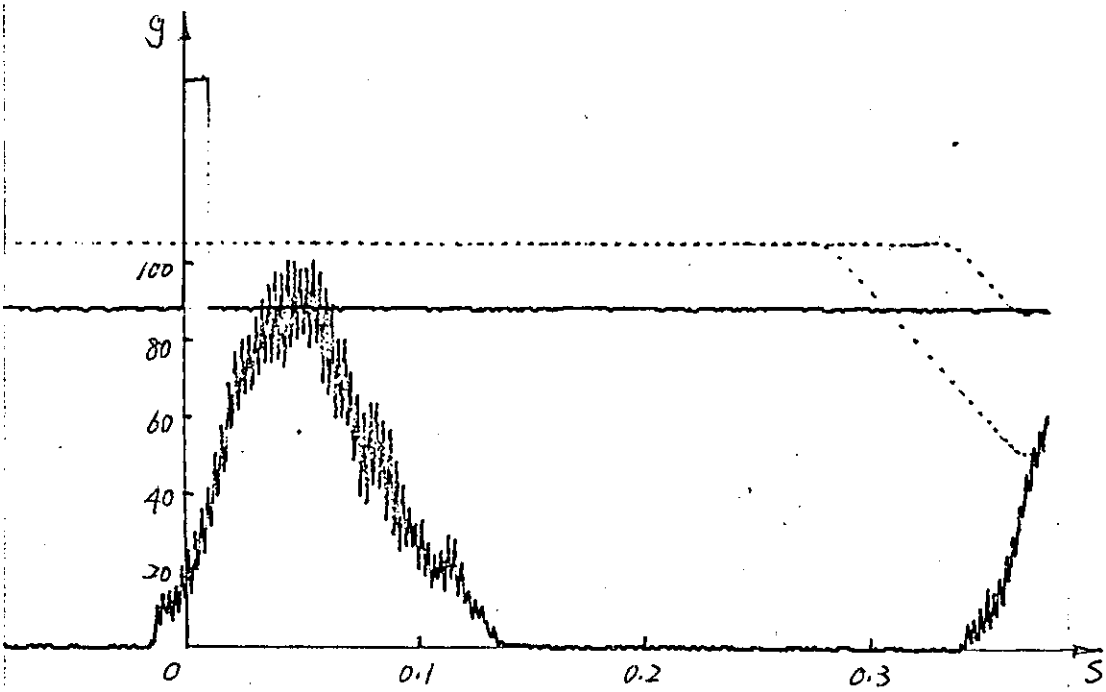


Fig 10.8a Tension Record 5a

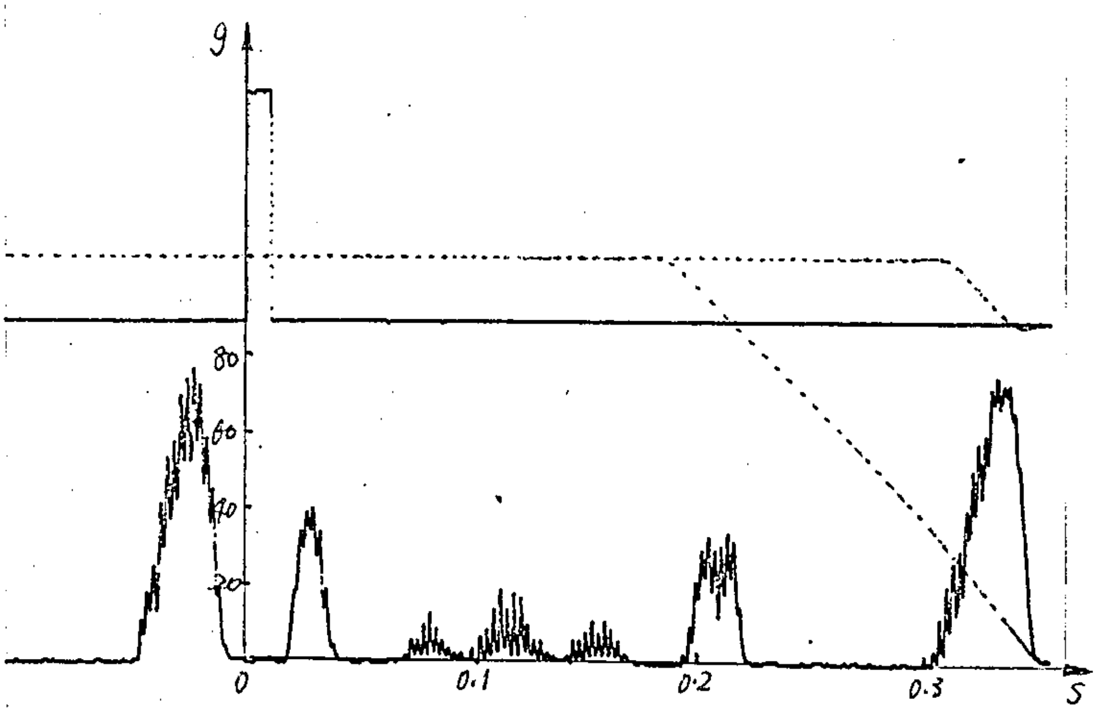


Fig 10.8b Tension Record 5b

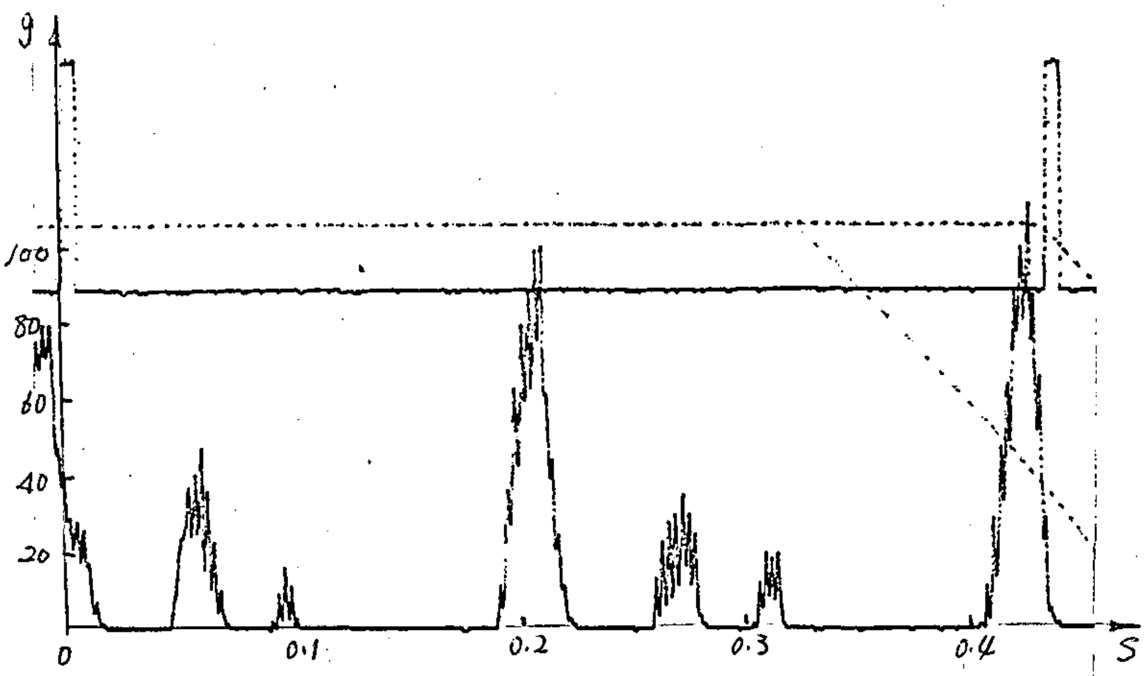


Fig 10.9a Tension Record 6a

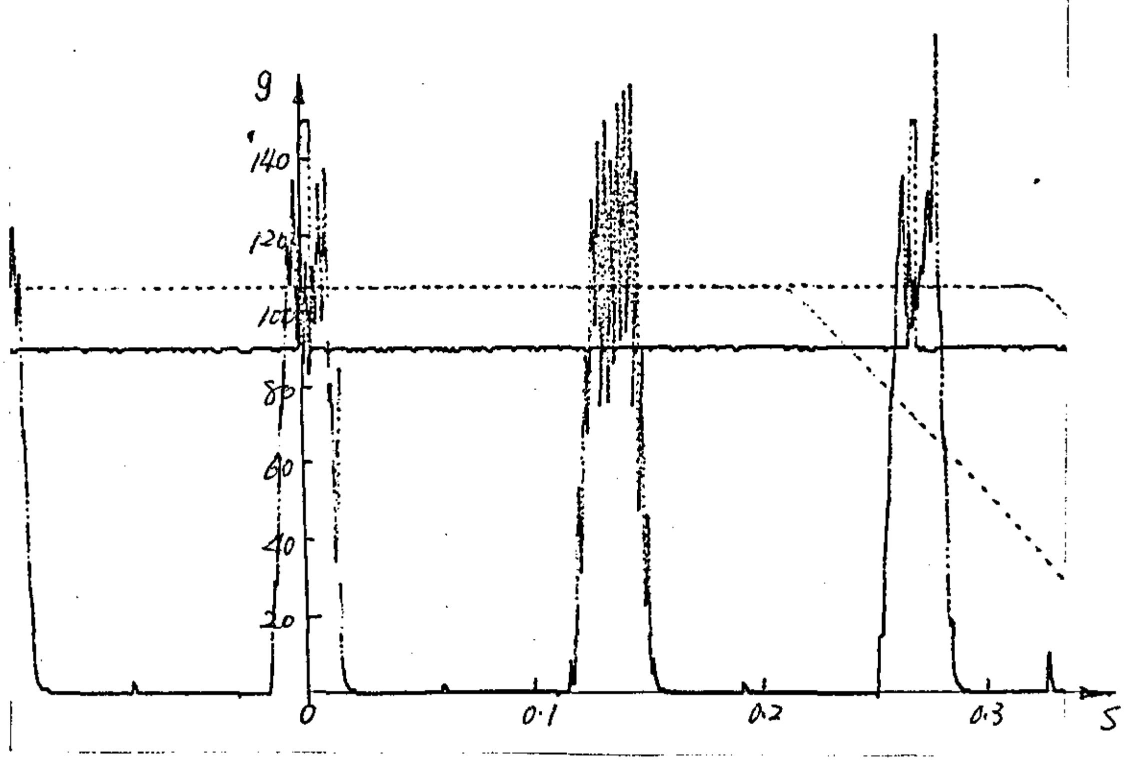


Fig 10.9b Tension Record 6b

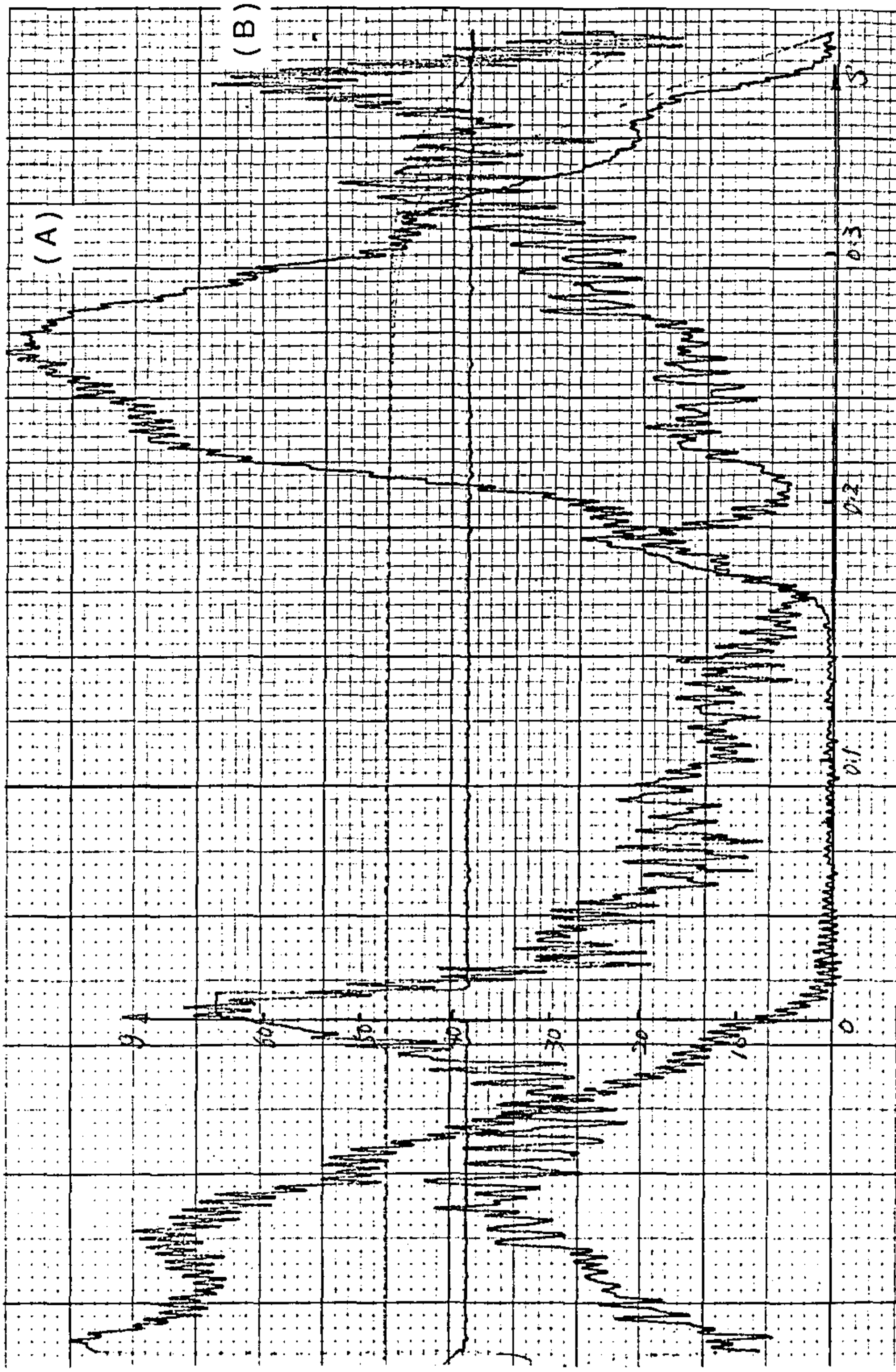


Fig 10.10a Tension Record 7a

Fig 10.10b Tension Record 7b

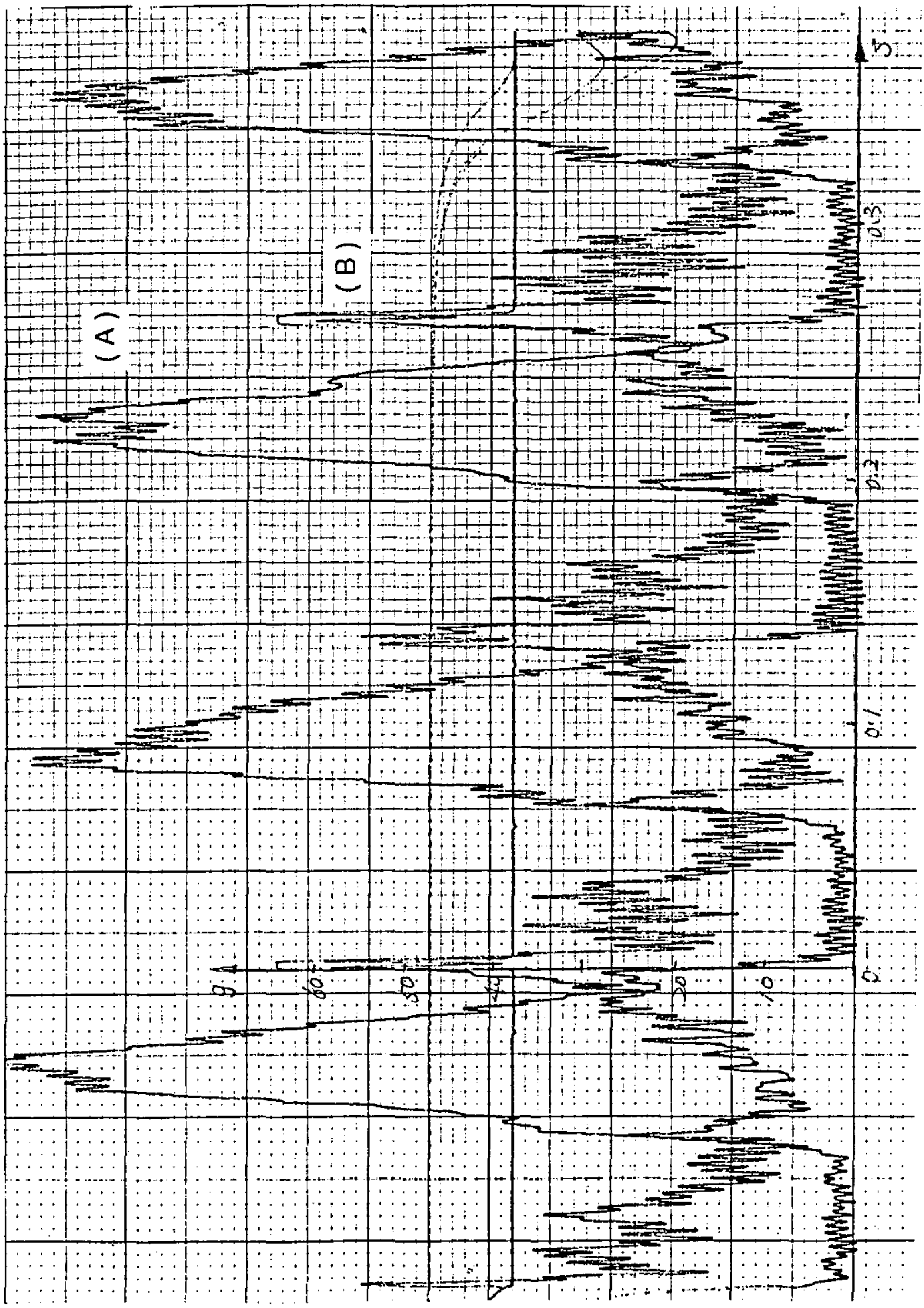


Fig 10.11a Tension Record 8a

Fig 10.11b Tension Record 8b

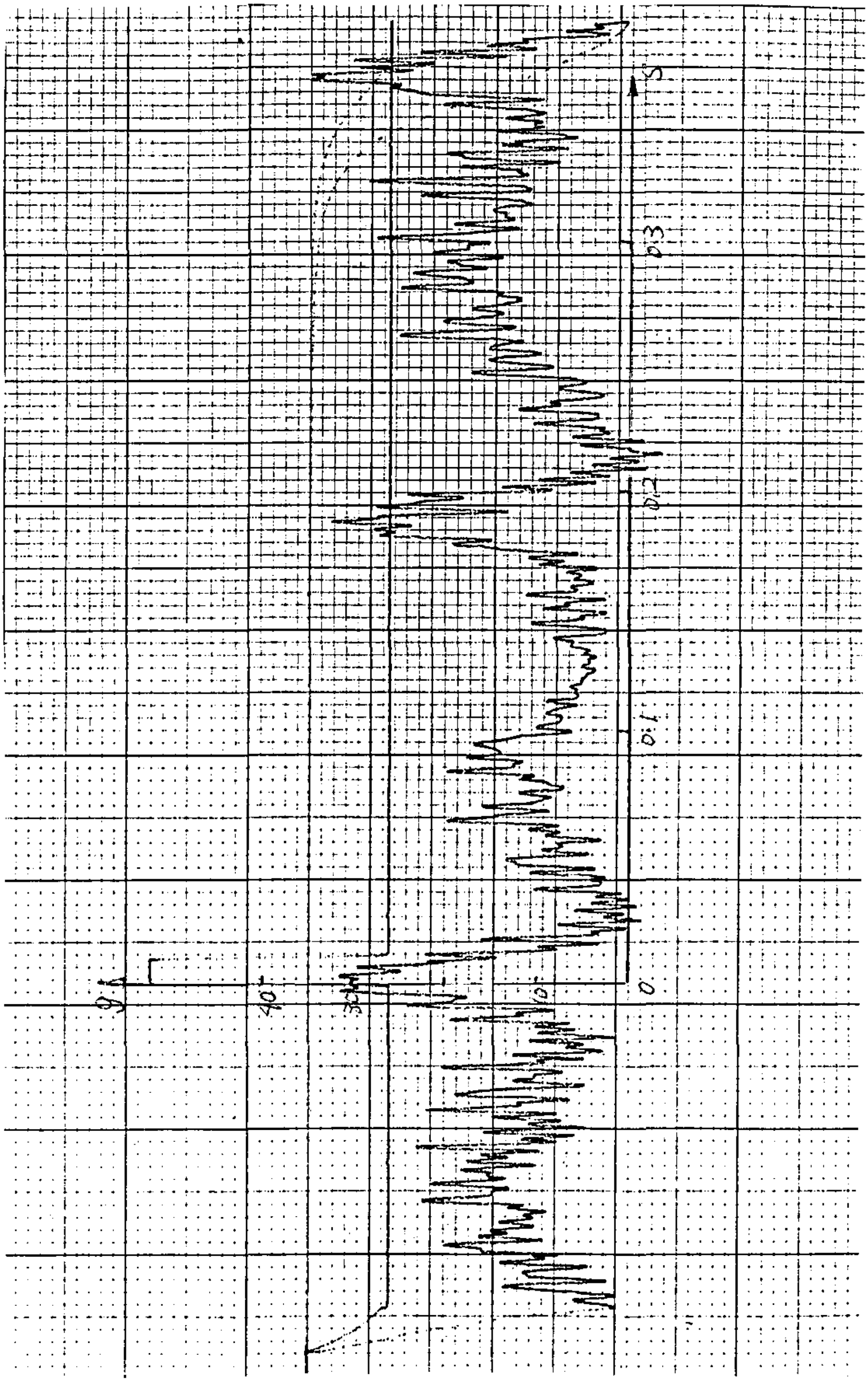


Fig 10.12 Tension Record 9

CHAPTER 11 CONCLUSIONS AND THE FUTURE WORK

11.1 Conclusions

In the research of reduction of tension variation in high speed cone winding, a mathematical model of cone winding process has been formulated. The variation of yarn winding error and the yarn path length have been analysed. The winding tension experiments have proved that the mathematical model for the yarn winding error is correct.

Two possible solutions have been investigated. One is using a servo motor to keep constant winding speed. The other is using a curved yarn distribution bar to keep constant yarn path length. The first solution has been proved not feasible at the present stage mainly because of lack of suitable servo motors. The second solution does reduce the tension variation in the cone winding process.

Since the variation of winding velocity is inevitable in cone winding, two tension compensators have been designed on the basis of winding error analysis. One is an optimised five bar linkage mechanism and the other is a microprocessor controlled stepper motor, both are used to control the yarn supply rate to the winding unit.

Designing a tension compensator, which will operate at yarn delivery speed of 500 m/min, is the demand of the ever increasing industrial productivity. Keeping the quality of yarn package, which should maintain at least its current level, is a requirement of the commercial success.

The computer simulation of the compensation work which is supposed to be done by the mechanical compensator has indicated that the optimised compensator is able to reduce the yarn winding error to a level which corresponds to the acceptable level of winding tension variation.

The tension experiments with mechatronic tension compensator have shown that the amplitude of the remaining tension variation can be reduced to one third of its original level and can maintain this same level at high speed winding, which is obviously better than that with the current spring compensator especially at high speed cone winding. A high speed video film has shown that by using a mechatronic tension compensator, no yarn slack occurs during cone winding process. This certainly improves the quality of the yarn package.

In the mechanical compensator design, for the simplification of design, the difference of yarn winding error between the first half cycle and the second half cycle has been neglected, whereas in the calculation of look-up tables for the mechatronic tension compensator, this difference has been taken into account, so, the look-up table for the first half cycle is slightly different from that of the second half cycle. Therefore, the mechatronic tension compensator works more delicately than the mechanical one can.

Considering the specification which was defined at the commencement of this research work, both the optimised mechanical compensator and the mechatronic compensator can satisfy this specification and provide a adequate control of the yarn tension variation. Both have potential for industrial application.

11.2 The Future Work

The research work on the design of a mechanical compensator has largely concentrated on the kinematic synthesis of mechanism. However, the dynamic effects on the compensation mechanism itself has not been fully considered. At high speed winding, the inertia force exerted on the compensation mechanism, the corresponding deformation of links and the machine vibration should be considered, since these factors certainly will affect the delicate compensation work. It is suggested that dynamic analysis of the mechanical compensator and its dynamic balancing be done to further improve the design of the mechanical compensator.

In the formulation of the mathematical model of yarn winding process, the pattern breaking effect has been neglected. The input/output ratio of variable belt drive is assumed to be 1. Actually, the pattern breaking is a required process to obtain a quality yarn package. For the further improvement of look-up tables, the effect of pattern breaking can also be considered.

The tension experiment with a mechatronic tension compensator has indicated that when the traverse guide is at the small end of the package, the compensator takes more yarn than it should do, and so causes a tension peak there. Modification of the look-up tables to improve the performance of the mechatronic compensator is therefore recommended for further work.

The 8031 microcontroller is a powerful vehicle. In this mechatronic tension compensator, only part of its functionability has been used. So, the microcontroller can play a important part in the automation of the high speed spinning-winding machinery.

APPENDIX

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COMPUTER PROGRAMME

```
      PROGRAM YARN
      REAL X(8),XL(8),XU(8),G(6),WK(1000),RPRM(20),OBJ
      INTEGER IWK(800), IPRM(20),NRWK,NRIWK,NDV,NCON,
     *        IPRINT,MINMAX, INFO
      NRWK=1000
      NRIWK=800
      DO 10 I=1,20
      RPRM(I)=0.
10     IPRM(I)=0
c      IPRM(2)=-1
c      RPRM(1)=-0.1
c      RPRM(2)=0.001
c      RPRM(3)=0.01
c      RPRM(4)=0.003
c      IPRM(13)=15
      METHOD=1
      NDV=8
      NCON=6
      IPRINT=4
      MINMAX=0
c     OPEN(15,FILE='RECORDN',STATUS='NEW')
      OPEN(14,FILE='ydatan1',STATUS='old')
      DO 15 I=1,NDV
15    READ(14,*) XL(I),X(I),XU(I)
      CLOSE(14)
      INFO=0
20    CALL DOT(INFO,METHOD,IPRINT,NDV,NCON,X,XL,XU,OBJ,
     * MINMAX,G,RPRM,IPRM,WK,NRWK,IWK,NRIWK)
      IF (INFO.EQ.0) STOP
      CALL EVALN(OBJ,X,G)
      GO TO 20
c      CLOSE(15)
      END
```

```fortran
      subroutine EVALN(obj, x,g)
c        this is the main subroutine to calculate objective function
      real x(*),g(*),ea(560),eb(560),obj
      open(13,file='eow',status='old')
      call fibar(x)
      read(9,30) (ea(i),i=1,560)
      read(13,30) (eb(i),i=1,560)
30     format(e13.6)
      obj=0.
      do 50 i=1,560
      obj=obj+(ea(i)+eb(i))*(ea(i)+eb(i))
50     continue
      call constr(x,g)
      close(9,status='delete')
      rewind(13)
      return
      end

      subroutine constr(x,g)
c        to calculate constrain function
      real x(*),g(*),ce1,ce2,ce3,ce4,re1,se1,re2,se2
      con=atan(1.)/45.
      ce1=x(8)*con
c        the range of x(8) is 56 degrees, the up position is used for
c     loading
      ce2=(x(8)-56.)*con
      ce3=sqrt((266+x(4)*cos(ce1))**2+(212+x(4)*sin(ce1))**2)
      ce4=sqrt((266+x(4)*cos(ce2))**2+(212+x(4)*sin(ce2))**2)
      if (ce3.gt.x(3)) then
        re1=ce3
        se1=x(3)
      else
        re1=x(3)
        se1=ce3
      endif
      g(1)=(re1+x(1)-x(2)-se1)/100.
      g(2)=(x(3)-x(2)-ce3+x(1))/100.
      g(3)=(ce3+x(1)-x(2)-x(3))/100.
      if (ce4.gt.x(3)) then
        re2=ce4
        se2=x(3)
```

```
      else
        re2=x(3)
        se2=ce4
      endif
      g(4)=(re2+x(1)-x(2)-se2)/100.
      g(5)=(x(3)-x(2)-ce4+x(1))/100.
      g(6)=(ce4+x(1)-x(2)-x(3))/100.
      return
      end

      subroutine fibar(x)
      real x(*)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      common /rd/nth,npts,th20,th50,delth2,delth5
c      read link geometry,fixed pivot coordinates and input crank
c      motion parameters
          open(10,file='ldatan1',status='unknown')
      call rdpos(x)
c      set up do loop for position analysis
      do 20 j=1,nth
           do 5  n=1,2
        do 10 k=1,npts
        th2=th20+(k-1)*delth2
        th5=th50+(j-1)*delth5
        call fibarp(x)
        call length
10        continue
5        continue
20      continue
          rewind(10)
      call erom
      close(10,status='delete')
      return
      end

      subroutine rdpos(x)
c      read position
      real x(*)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
```

```fortran
      common /rd/nth,npts,th20,th50,delth2,delth5
100   format(18x,'KINEMATIC MECHANISM ANALYSIS FIVE-BAR
  1    MECHANISMS')
      open(7,file='fdatan1',status='old')
      call rdgeom(x)
      call rdfixd(x)
      call rdinpt(x)
      return
      end

      subroutine rdgeom(x)
c        read geometry parameters for linkage
      real x(*)
      integer m
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      con=atan(1.)/45.
      read(7,*) s2,p2,s4,p4,s5,p5
      ph2=p2*con
      ph3=x(6)*con
      ph4=p4*con
      ph5=p5*con
      read(7,*) m
      return
      end

      subroutine rdfixd(x)
c        read fixed pivot coordinates
      real x(*)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      read(7,*)p(1,1),p(1,2),p(8,1),p(8,2)
      return
      end

      subroutine rdinpt(x)
c        read input parameters for the crank and rocker
      real x(*)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      common /rd/nth,npts,th20,th50,delth2,delth5
```

```
      read(7,*)nth,npts,delte2,delte5
      con=atan(1.)/45.
      th20=x(7)*con
      delth2=delte2*con
      th50=x(8)*con
      delth5=delte5*con
      rewind(7)
      return
      end

      subroutine fibarp(x)
c        five bar linkage position analysis
      real x(*)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      call crank2(1,2,5,x(1),s2,ph2,th2,p)
      call crank2(8,4,9,x(4),s5,ph5,th5,p)
      call pdyad(m,2,4,3,x(2),x(3),th3,th4,p)
      call pos(2,3,6,x(2),x(5),ph3,th3,p)
      call pos(4,3,7,x(3),s4,ph4,th4,p)
      return
      end

      subroutine crank2(n1,n2,n3,r,s,phi,theta,pj)
c        crank position analysis
      dimension pj(30,2)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      rx=r*cos(theta)
      ry=r*sin(theta)
      sx=s*cos(theta+phi)
      sy=s*sin(theta+phi)
      pj(n2,1)=pj(n1,1)+rx
      pj(n2,2)=pj(n1,2)+ry
      pj(n3,1)=pj(n1,1)+sx
      pj(n3,2)=pj(n1,2)+sy
      return
      end

      subroutine pdyad(m1,n1,n2,n3,r1,r7,th1,th7,pj)
c        dyad position analysis
```

```
      dimension pj(30,2)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      logical prnt
      integer m1,n1,n2,n3
      prnt=.true.
      if(n1.lt.0) prnt=.false.
c       n1=iabs(n1)
      dlx=pj(n2,1)-pj(n1,1)
      if (abs(dlx).le.1.e-10) dlx=1.e-10
      dly=pj(n2,2)-pj(n1,2)
      phi=atan2(dly,dlx)
      ssq=(pj(n2,1)-pj(n1,1))**2+(pj(n2,2)-pj(n1,2))**2
      s=sqrt(ssq)
      test1=s-(r1+r7)
      if (test1) 40,40,500
40     test2=abs(r1-r7)-s
      if (test2) 50,50,500
50     continue
      cosin=(r1**2+ssq-r7**2)/(2.*r1*s)
      alpha=atan2(sqrt(1.-cosin**2),cosin)
      if (m1) 200,100,100
100    theta=phi+alpha
      goto 300
200    theta=phi-alpha
300    continue
      pj(n3,1)=pj(n1,1)+r1*cos(theta)
      pj(n3,2)=pj(n1,2)+r1*sin(theta)
      th1=atan2((pj(n3,2)-pj(n1,2)),(pj(n3,1)-pj(n1,1)))
      th7=atan2((pj(n3,2)-pj(n2,2)),(pj(n3,1)-pj(n2,1)))
      return
500    if (prnt) write(0,600)
600    format(5x,'DYAD CANNOT BE ASSEMBLED')
      return
      end

      subroutine pos(n1,n2,n3,r,s,phi,theta,pj)
c         position analysis for the links in the dyad
      dimension pj(30,2)
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
```

```
      ct=cos(theta)
      st=sin(theta)
      cp=cos(phi)
      sp=sin(phi)
      pj(n2,1)=pj(n1,1)+r*ct
      pj(n2,2)=pj(n1,2)+r*st
      pj(n3,1)=pj(n1,1)+s*cp*ct-s*sp*st
      pj(n3,2)=pj(n1,2)+s*cp*st+s*sp*ct
      return
      end

c      subroutine wrpos
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      common /rd/nth,npts,th20,th50,delth2,delth5
      con=atan(1.)/45.
      thet2=th2/con
      thet3=th3/con
      thet4=th4/con
      thet5=th5/con
      write(8,100)thet2,thet5,thet2,thet3,thet4,thet5
      write(8,200)
      do 400 j=1,npts
      write(8,300)p(6,1),p(6,2)
100   format(//10x,'POSITION RESULTS FOR
      THETA2=',f6.2,5x,'THETA5=', f6.2,'DEG'
     1 //5x,'th2=',e11.4,3x,'th3=',e11.4,3x,'th4=',e11.4,3x,'th5=',
     1 e11.4,'deg')
200   format(//2x,'POINT',7x,'POSITION',14x/'NUMBER',5x,
     1 'X',10x,'Y',5x)
300   format(2e13.6)
      return
      end

      subroutine length
      common /fibr/m,s2,ph2,ph3,s4,ph4,s5,ph5,p(30,2),
     1 pi(30,2),th2,th3,th4,th5
      data x1,y1,x2,y2,r10,r20,r/-52.,44.,-45.,0.,2.,2.,2./
      con=atan(1.)/45.
         x6=p(6,1)
         y6=p(6,2)
```

```
      b1=sqrt((x6-x1)*(x6-x1)+(y6-y1)*(y6-y1)-(r+r10)*(r+r10))
      b2=sqrt((x6-x2)*(x6-x2)+(y6-y2)*(y6-y2)-(r+r20)*(r+r20))
      z1=atan((r+r10)/b1)
      z2=atan((r+r20)/b2)
      f1=atan((y1-y6)/(x6-x1))
      f2=atan((y6-y2)/(x6-x2))
      p1=con*90.-f1+z1
      p2=con*90.-f2+z2
      c1=p1*r10
      c10=p1*r
      c2=p2*r20
      c20=p2*r
      e=b1+b2+c1+c10+c2+c20
      write(10,15) e
15     format(e13.6)
         return
         end

      subroutine erom
         dimension d(41),em(40)
      open(9,file='eom',status='unknown')
      do 40 k=1,14
      read(10,18) (d(i),i=1,40)
18     format(e13.6)
      d(41)=d(1)
      do 30 i=1,40
      em(i)=d(i+1)-d(i)
30     continue
      write(9,25) (em(i),i=1,40)
25     format(e13.6)
40     continue
      rewind(9)
      return
      end
```

```
      program looktab
c     this is to find number of encoder counts for each step
c     start searching for encoder count at which
c     slack = amount of yarn taken up by 1 step away from start
c     start position = 0 for now
      integer m(3250),n(100),lookup(300),iturn(10)
      integer k,k2,k3,istep,istart,look
      real yarn(100),takeup(100),angle(3250),slack(3250)
      real n_slack,o_slack,take
      logical up,o_up
      open(6,file='looktab2.dat',status='unknown')
      open(7,file='ang-errf-re.dat',status='old')
      open(8,file='yarn-in-ec.dat',status='old')
      read(8,5)(n(j),yarn(j),takeup(j),j=-36,36)
5     format(i4,2f10.3)
      close(8)

c     write headings for print output
c     write(6,10)
10    format('step  encoder  slack  takeup  E.C.'/
     1  'No.  counts  (mm)  (mm)  position'//)
      do 140 l=0,6
      istart=0
      istep=istart+1
      o_slack=0.
      o_up=.true.
      k=1
c     k2=1

c     find number of encoder count for each step
c     i is ebcoder count, 250 count for each camshaft revolution
      do 20 i=1,3250
      read(7,30) m(i),angle(i),slack(i)
      n_slack=slack(i)
30    format(i4,2f10.4)

c     decide going direction
      if (n_slack.lt.o_slack) then
       up=.true.
      else
       up=.false.
```

```
      endif
c     decide if it is a turning point
      if(up.neqv.o_up) then
c       this is a turning point, so reverse search direction
        if (up) then
          istep=istep+1
        else
          istep=istep-1
        endif
c       iturn(k2)=i
c       k2=k2+1
c       write(6,32)
32      format('   turn direction')
      else
c       this is not a turning point
        if(up) then
c          slack is increasing so look for next step
           take=takeup(istep)
           if(-n_slack.gt.take)then
c             there is enough slack to take next step
c             write(6,40)k,i,n_slack,take,istep
40            format(i4,i7,2f10.3,i7)
              lookup(k)=i
              k=k+1
              istep=istep+1
           endif
        else
c          slack is decreasing
           take=takeup(istep)
           if(-n_slack.lt.take) then
c             a step must be paied back
c             write(6,40)k,i,n_slack,take,istep
              lookup(k)=i
              k=k+1
              istep=istep-1
            endif
     endif
      endif
      o_slack=n_slack
      o_up=up
```

```
20      continue

c       print out the results
        lookup(k)=3250
c       print the size of yarn package
        write(6,45)l*10
45      format(//'      Lookup Table for Electronic Compensator'/
     1 '     ****************************************'/
     1 '     yarn package size = ',i4,' mm'/)
c       print the number of lookup table entries found
        write(6,50)k
50      format(//'number of lookup table entries=',i7//)
c       convert the lookup table data to incremental form
        do 60 i=k,2,-1
         lookup(i)=lookup(i)-lookup(i-1)
60      continue

c       print out the lookup table
        write(6,65)
65      format(//'lookup table'/)
        do 70 i=1,k,10
         write(6,80)(lookup(j),j=i,i+9)
80       format(10i7)
70      continue
c       print out the lookup table in 256-n form
        write(6,90)
90      format(//'lookup table in 256-n form'/)
        do 100 i=1,k,10
         write(6,80)((256-lookup(j)),j=i,i+9)
100     continue
c       calculate total lookup table entries
        itot=0
        do 110 i=1,k
         itot=itot+lookup(i)
110     continue
        write(6,120)itot
120     format(1h, //'lookup table total count =',i7)
        do 130 i=1,300
        lookup(i)=0
130     continue
140     continue
```

```
close(7)
close(6)
stop
end
```

ASSEMBLY PROGRAMME

```
@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@
;           THE YARN TENSION COMPENSATION PROGRAM
@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@

RST:      SJMP    MAIN            ;power-on reset

          ORG     03H             ;cone-size interrupt rout.
IEO_SR:   CLR     TCON.4
          MOV     30H,TL0
          MOV     31H,TH0
          MOV     TL0,#00H
          MOV     TH0,#00H
          SETB    TCON.4
          RETI

          ORG     1BH             ;step-rate interrupt rout.
TF1_SR:   MOV     TH1,#0FFH
          MOV     TL1,R3
          SETB    TCON.6
          PUSH    ACC
          PUSH    DPL
          PUSH    DPH
          ACALL   OUTPUT
          ACALL   INP1
          POP     DPH
          POP     DPL
          POP     ACC
          RETI

@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@
;           executable code starts here
@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@@

MAIN:     MOV     DPTR,#DTAB0     ;initialisation
          MOV     34H,DPH
          MOV     35H,DPL
MAIN1:    MOV     IP,#00H
```

```
          MOV     SP,#07H
          MOV     TL1,#00H
          MOV     TH1,#00H
          MOV     TL0,#00H
          MOV     TH0,#00H
          MOV     IE,#89H
          SETB    IP.3
          MOV     TMOD,#59H
          MOV     TCON,#11H
          MOV     R0,#00H
          MOV     R2,#00H
          ACALL   INPUT
          ACALL   PARK
          MOV     TH1,#0FFH
          MOV     TL1,R3
          SETB    TCON.6
          ACALL   INP1
CNTTST:   JNB     TCON.1,CNTTST   ;wait for valid count......


COMP:     CLR     IE.0            ;update routine
          MOV     A,30H
          JNZ     CP
          INC     30H
CP:       MOV     R6,30H
          MOV     R7,31H
FTCH:     MOV     R1,#00H
FTCH1:    MOV     DPTR,#CNTCMP    ;load CNTCMP: high addr.byte
          MOV     A,R1
          MOVC    A,@A+DPTR
          MOV     33H,A
          INC     R1
          MOV     A,R1
          MOVC    A,@A+DPTR
          MOV     32H,A
CMP:      MOV     A,R6
          ADD     A,32H
          MOV     A,R7
          ADDC    A,33H
          JC      UPDAT
```

```
        INC     R1
        CJNE    R1,#CCEND-CNTCMP,FTCH1
        SJMP    FTCH
UPDAT:  MOV     DPTR,#DTAPTR    ;load DTAPTR: high addr. byte
        MOV     A,R1
        MOVC    A,@A+DPTR
        CLR     IE.0
        MOV     35H,A
        DEC     R1
        MOV     A,R1
        MOVC    A,@A+DPTR
        MOV     34H,A
        MOV     IE,#89H
        JNB     P3.3,Y_BRK      ;check yarn-break
IDLE:   JNB     TCON.1,IDLE     ;idle in cone-size counts
        SJMP    COMP

Y_BRK:  MOV     IE,#00H         ;yarn break routine
        MOV     TCON,#00H
        MOV     DPH,34H
        MOV     DPL,35H
        JMP     MAIN1


INP1:   MOV     DPH,R4          ;step-pattern routines
        MOV     DPL,R5
INPUT:  CLR     A
INPUT1: MOVC    A,@A+DPTR
        JZ      CMD
        MOV     R3,A
        INC     DPTR
        MOV     R4,DPH
        MOV     R5,DPL
        RET
CMD:    INC     DPTR
        MOVC    A,@A+DPTR
        JZ      DIR
        MOV     DPH,34H
        MOV     DPL,35H
        MOV     R2,#00H
```

```
        SJMP    INPUT
DIR:    INC     R2
        INC     DPTR
        SJMP    INPUT1


OUTPUT: MOV     A,R2            ;step output
        JNB     ACC.0,RVSOUT
        SJMP    FWDOUT
FSTART: MOV     R0,#00H

FWDOUT: MOV     DPTR,#MSTEP     ;load MSTEP: high addr. byte
        INC     R0

        MOV     A,R0
        MOVC    A,@A+DPTR
        JZ      FSTART
        MOV     P1,A
        RET
RSTART: MOV     R0,#RSTEP-MSTEP

RVSOUT: MOV     DPTR,#MSTEP     ;load MSTEP: high addr. byte
        DEC     R0

        MOV     A,R0
        MOVC    A,@A+DPTR
        JZ      RSTART
        MOV     P1,A
        RET


PARK:   JB      P3.0,FSTART1    ;find start position
        MOV     R0,#RSTEP-MSTEP
R_NXT:  ACALL   RVSOUT
        ACALL   DELAY
        JB      P3.0,YB
        JMP     R_NXT
FSTART1:MOV     R0,#00H
F_NXT:  ACALL   FWDOUT
        ACALL   DELAY
```

```
        JB      P3.0,F_NXT
YB:     JNB     P3.3,YB
        MOV     R0,#RSTEP-MSTEP
YB1:    ACALL   RVSOUT
        ACALL   DELAY
        JNB     P3.0,YB1
YB2:    ACALL   RVSOUT
        ACALL   DELAY
        JB      P3.0,YB2
SYNC:   JB      P3.1,SYNC       ;wait for sync......
        RET

DELAY:  MOV     R7,#04H
DELAY2: MOV     R6,#0FFH
DELAY1: DJNZ    R6,DELAY1
        DJNZ    R7,DELAY2
        RET


;DATA TABLES START HERE
*************************************************


MSTEP:  DB      00H
        DB      1EH,1BH,3BH,13H,17H,12H,32H,1AH
RSTEP:  DB      00H

DTAPTR: DW DTAB0,DTAB1,DTAB2,DTAB3,DTAB4,DTAB5,DTAB6

CNTCMP: DW
        0E796H,0EB30H,0EF00H,0F2FFH,0F5FFH,0FA00H,0FFFFH
CCEND:  DB      00H

TURN    EQU     0000H
REPEAT  EQU     0001H
```

```
;       Lookup Table for Mechatronic Compensator
;       ****************************************
;       yarn package size =    0 mm

DTAB0:  DB      251,251,252,251,252,250,252,250,251,250
        DB      249,249,248,248,247,244,245,245,246,245
        DB      245,246,246,246,245,246,243,242,241,240
        DB      240,237,234,227,222
        DW      TURN
        DB                                  120,229,240,241,243
        DB      243,239,247,246,247,247,247,249,248,250
        DB      249,250,249,250,251,250,250,250,251,250
        DB      250,250,250,250,249,250,249,248,248,249
        DB      247,248,248,248,248,249,249,249,250,248
        DB      250,250,250,249,250,249,249,249,248,248
        DB      248,246,246,246,244,244,242,242,242,237
        DB      231,210
        DW      TURN
        DB              175,192,220,226,236,237,239,245
        DB      243,243,242,243,243,242,244,242,243,245
        DB      246,248,248,250,249,250,251,251,251,251
        DB      251,252,251,252,251,251,251,252,250,251
        DB      250,250,249,249,247,247,245,245,244,244
        DB      245,244,245,245,246,245,243,241,240,239
        DB      239,237,234,223,208
        DW      TURN
        DB                                  150,225,238,240,242
        DB      239,244,245,247,246,248,247,249,249,248
        DB      250,250,249,250,250,250,251,249,251,249
        DB      250,250,249,249,249,247,249,247,247,248
        DB      247,248,248,249,248,249,250,249,250,249
        DB      250,248,250,249,249,248,248,247,246,246
        DB      245,244,243,240,241,241,232,221
        DW      TURN
        DB
                                                       152,205
        DB      217,230,233,236,240,241,246,243,242,244
        DB      243,243,243,243,243,244,245,247,247,249
        DB      249,251,250,250,251,251,251,252,251,252
        DW      REPEAT
```

```
;       Lookup Table for Mechatronic Compensator
;       ****************************************
;
;       yarn package size =   10 mm

DTAB1:  DB      250,250,250,250,249,249,248,248,245,244
        DB      238,237,237,240,239,241,240,239,230,231
        DB      228,221,209,190
        DW      TURN
        DB                          181,214,232,237,229,243
        DB      242,244,246,245,248,247,248,248,248,249
        DB      248,248,248,248,247,247,245,243,241,243
        DB      243,245,246,247,247,248,248,248,247,248
        DB      246,246,247,244,243,241,238,239,238,234
        DB      216
        DW      TURN
        DB          135,185,209,224,230,237,242,236,236
        DB      237,232,235,240,242,246,247,248,249,250
        DB      249,251,249,251,249,250,250,249,248,248
        DB      246,244,241,237,234,237,238,240,240,235
        DB      228,229,227,210,193
        DW      TURN
        DB                              140,226,235,228,241
        DB      241,246,244,245,247,247,247,249,248,249
        DB      247,248,248,247,246,245,243,242,242,241
        DB      244,246,246,248,247,247,248,247,248,246
        DB      246,245,244,242,240,239,234,237,224
        DW      TURN
        DB                                              118
        DB      196,210,224,229,235,243,236,238,237,236
        DB      234,236,241,244,245,249,248,249,250,249
        DB      251,250
        DW      REPEAT

;       Lookup Table for Mechatronic Compensator
;       ****************************************
;
;       yarn package size =   20 mm

DTAB2:  DB      248,249,249,248,247,247,243,240,226,226
        DB      233,233,234,224,219,217,195,187
        DW      TURN
```

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        DB                                      152,222
        DB      229,248,254,234,239,242,243,245,245,246
        DB      247,247,247,246,247,244,244,239,234,236
        DB      239,243,245,246,246,246,247,245,245,244
        DB      242,239,237,232,235,225
        DW      TURN
        DB                              101,180,200,221
        DB      232,236,232,227,220,230,239,244,245,249
        DB      248,248,249,249,249,249,247,248,246,242
        DB      238,220,226,233,233,227,216,218,194,190
        DW      TURN
        DB      147,221,220,237,239,242,244,244,246,246
        DB      247,246,247,245,245,243,238,233,236,239
        DB      243,244,245,246,246,246,245,244,243,242
        DB      238,233,230,231
        DW      TURN
        DB                      91,189,205,217,229,238
        DB      232,232,225,222,236,241,246,246,248,249
        DB      249,249
        DW      REPEAT

;       Lookup Table for Mechatronic Compensator
;       ****************************************
;       yarn package size =  30 mm

DTAB3:  DB      247,249,247,246,245,241,229,207,225,227
        DB      215,207,191,196
        DW      TURN
        DB                      91,221,253,244,235,236
        DB      241,243,244,245,245,245,245,246,242,239
        DB      223,229,236,243,244,245,244,246,244,241
        DB      242,237,229,230,228
        DW      TURN
        DB                          101,172,190,214,238
        DB      223,222,200,233,242,245,246,248,248,248
        DB      248,247,246,245,241,229,203,225,226,203
        DB      208,178,189
        DW      TURN
        DB                  160,216,244,254,228,237,239
        DB      243,243,245,245,246,245,243,241,233,224
```

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        DB      228,241,242,244,245,244,245,241,242,238
        DB      233,229,226
        DW      TURN
        DB                  194,184,194,212,233,225,227
        DB      206,221,238,244,247,247,248,248
        DW      REPEAT

;       Lookup Table for Mechatronic Compensator
;       ****************************************
;       yarn package size =   40 mm

DTAB4:  DB      246,248,245,244,242,223,251,254,195,222
        DB      204,196,177,184
        DW      TURN
        DB
                            139,197,228,237,241,241
        DB      244,244,244,244,243,238,215,219,238,243
        DB      243,243,245,241,241,238,228,226,229
        DW      TURN
        DB                                          112
        DB      164,178,205,234,221,178,231,242,245,245
        DB      248,247,248,245,245,243,231,179,219,196
        DB      199,166,188
        DW      TURN
        DB                  173,195,227,237,239,243,243
        DB      244,244,244,240,235,203,228,241,243,243
        DB      243,243,242,238,233,226,222
        DW      TURN
        DB                                  106,183,179
        DB      206,233,218,187,254,252,213,239,244,246
        DB      247,247
        DW      REPEAT

;       Lookup Table for Mechatronic Compensator
;       ****************************************
;       yarn package size =   50 mm
DTAB5:  DB      245,247,245,242,229,168,212,179,171,185
        DW      TURN
        DB       98, 235,254,225,230,240,240,243,244,243
        DB      241,238,207,253,254,215,241,242,242,243
```

```
        DB      242,239,236,227,220,212
        DW      TURN
        DB                                      189,170,160,184
        DB      229,211,160,236,243,245,247,245,247,246
        DB      243,238,165,210,175,170,187
        DW      TURN
        DB                                          104,244,248
        DB      254,254,226,235,238,242,242,243,243,240
        DB      232,188,232,239,243,243,241,241,235,234
        DB      215,216
        DW      TURN
        DB              178,174,166,189,226,212,165,226
        DB      242,245,246,246
        DW      REPEAT

;       Lookup Table for Mechatronic Compensator
;       ****************************************
;       yarn package size =  60 mm
DTAB6:  DB      245,245,244,238,152,207,163,164,185
        DW      TURN
        DB                                                  128
        DB      229,254,221,229,241,239,243,241,242,239
        DB      222,254,248,253,253,254,252,253,253,252
        DB      254,253,254,223,239,240,243,240,241,233
        DB      226,219,205
        DW      TURN
        DB                  244,140,161,167,226,200,253
        DB      252,210,254,211,241,244,245,246,246,245
        DB      239,232,135,163,158,193
        DW      TURN
        DB                                  115,246,243,224
        DB      232,238,240,243,243,239,236,171,254,254
        DB      237,239,241,243,238,237,230,213,207
        DW      TURN
        DB                                              203
        DB      175,173,169,222,208,148,237,243,245,246
        DW      REPEAT

        END     ;THE END
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