The design of a yarn tension control system for the high speed formation of conical packages

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THE DESIGN OF A YARN TENSION CONTROL SYSTEM FOR THE HIGH
SPEED FORMATION OF CONICAL PACKAGES

BY

IAN CLIFFORD WRIGHT

A Doctoral Thesis
Submitted in partial fulfilment of the requirements
for the award of
Doctor of Philosophy of the Loughborough University of Technology

June 1990

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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.

Ian C. Wright.
Abstract

The research described was primarily concerned with working towards the solution of the problem of yarn tension control when building conical packages on an open-end spinning machine at yarn delivery speeds of 500 m/min. Because of the total lack of published work on the mathematical modelling of the parameters affecting tension variation, a considerable amount of effort was spent in the establishment of this basic understanding. This mathematical analysis appears, in the main, in the appendices to this thesis. In addition to the analysis of yarn tension variations depending on machine geometry parameters, mathematical models have also been produced which predict the effect on tension of fitting curved distribution bars of various shapes.

The work includes the development of concept ideas which appeared capable of providing routes for solution of the tension control problem. These ideas were evaluated and compared against a design specification which was also developed as part of this research. Two promising concepts were selected for detailed investigation. These were: a pneumatic passive compensator and, a positive mechanical compensator.

The pneumatic compensator was a novel idea which was conceived by the author, and which eventually provided a solution which met the design specification in every way. The mechanical compensator was shown to be a far less attractive concept with several serious problems which have been explored but not eliminated. A preliminary investigation of a third concept involving microprocessor control was not sufficiently well developed at the time of publication to be included in this thesis.
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Nomenclature

$A_p$ cross sectional area of nozzle orifice without the float

$a_i$ length of yarn laid down in the $i$ th segment

$a_p$ cross sectional area of the nozzle orifice measured at the float

$C_d$ circumference of the rubber drive tyre

$C_m$ circumference of the developed surface of the conical package measured at the mean diameter of the package

$C_p$ nozzle discharge coefficient

$d_p$ diameter of the disc float

$ds$ diameter of the unstretched yarn, i.e. when the axial force is zero.

$D$ diameter of the small end of the package.

$D_p$ diameter of the delivery roller shaft.

$D_r$ diameter of the drive tyre.

$E_p$ height of the nozzle at the throat

$F_p$ force on the float

$h_i$ distance between the two most distant corners from the package focus of a segment on the surface of the package.

$H_p$ height of the nozzle measured at the float

$K_n$ force constant - or stiffness - of a piece of yarn of test length $n$.

$l$ distance of package focus from nearest corner of package surface element

$l_1$ length of the mechanism control link

$l_2$ length of side $ab$ of the mechanism input link

$l_3$ length of side $bc$ of the mechanism input link

$L$ distance between the reciprocating yarn guide and the focus of the conical package.

$M$ width of the package measured on the surface.

$n$ test length for a piece of yarn with stiffness $K_n$.

$n_k$ speed of rotation of the scroll cam (rev/min)

$N$ number of segments across the width of the package.

$P$ distance from the focus of a conical package to a point on the periphery of the
package

$P_m$  peripheral speed of the delivery rollers.

$P_{mi}$  peripheral speed of the delivery rollers in the $i$ th time interval.

$P_q$  peripheral speed of the package at the diameter which contacts the drive tyre.

$P_{qi}$  peripheral speed of the package at the winding on point during the $i$ th time interval.

$Pr$  proportion of a full revolution passed through by the package in a defined time interval.

$P_t$  peripheral speed of the drive tyre.

$Q_p$  volume flow rate of air through the nozzle

$r_1$  radius of mechanism input crank

$r_2$  radius of mechanism control crank

$r_d$  the radius of the drive tyre

$r_l$  radius of small end of package

$r_m$  mean radius of the package

$R$  number of radians through which the package will rotate during a defined time interval.

$R_l$  radius of the large end of the package

$R_p$  ratio of the anti-patterning device.

$s$  circumference of the small end of the package

$s_1$  distance between points $b$ and $d$ in the compensator mechanism

$s_2$  distance between points $3$ and $c$ in the compensator mechanism

$t$  time.

$t_p$  width of the nozzle

$T$  tension (axial force) acting on the yarn.

$T_d$  tension draft

$T_{guide}$  periodic time of the reciprocating yarn guide

$T_{A-K}$  number of teeth on the timing belt pulleys which transmit motion to the delivery roller, drive roller, and scroll cam shafts.

$V$  volume of the free yarn length. i.e between points $m$ and $q$. 
$V_{fm}$ volume flow rate of the yarn immediately prior to passing between the delivery rollers at point m.

$V_{fq}$ volume flow rate of the yarn through the reciprocating guide.

$V_m$ volume of yarn passing point m in unit time.

$V_{mi}$ volume of yarn passing point m in the $i$th time interval.

$V_n$ stretched yarn volume during the $n$th time interval.

$V_P$ peripheral velocity of the delivery roller shaft.

$V_q$ volume of yarn passing point q in unit time.

$V_{qi}$ volume of yarn passing point q in the $i$th time interval.

$V_y$ velocity of the yarn

$w$ distance moved by the guide in unit time.

$x$ length of yarn laid down on the package surface in time $t$

$x_a$ $x$ coordinate of point a on the mechanism control link/input link

$x_b$ $x$ coordinate of point b on the mechanism input crank/input link pivot

$x_c$ $x$ coordinate of point c on the mechanism input link/connecting link pivot

$x_d$ $x$ coordinate of point d on the mechanism control crank/control link pivot

$x_e$ $x$ coordinate of point e on the mechanism connecting link/output link pivot

$x_f$ $x$ coordinate of point f on the mechanism output link

$x_p$ distance of the float centre from the nozzle throat

$X_1$ $x$ coordinate of mechanism input crank pivot

$X_2$ $x$ coordinate of mechanism control crank pivot

$X_3$ $x$ coordinate of mechanism output crank pivot

$U$ velocity of the yarn guide.

$Y_a$ $y$ coordinate of point a on the mechanism control link/input link

$Y_b$ $y$ coordinate of point b on the mechanism input crank/input link pivot

$Y_c$ $y$ coordinate of point c on the mechanism input link/connecting link pivot

$Y_d$ $y$ coordinate of point d on the mechanism control crank/control link pivot

$Y_e$ $y$ coordinate of point e on the mechanism connecting link/output link pivot
\( y_f \) \ y \ \text{coordinate of point } f \ \text{on the mechanism output link}

\( y \) \ the shortest distance between the reciprocating yarn guide and the nearest point on the periphery of the small end of the package.

\( Y_1 \) \ \text{y \ coordinate of mechanism input crank pivot}

\( Y_2 \) \ \text{y \ coordinate of mechanism control crank pivot}

\( Y_3 \) \ \text{y \ coordinate of mechanism output crank pivot}

\( Z \) \ \text{length of yarn path between tension generating nip points}

\( \alpha \) \ \text{an internal angle defining the geometry of a segment of the package surface.}

\( \alpha_p \) \ \text{half angle of the nozzle walls}

\( \beta \) \ \text{the angle spanned by the developed cone.}

\( \Delta P_p \) \ \text{pressure drop across the float}

\( \gamma \) \ \text{angular position of a line through point } b \ \text{and } d \ \text{on the compensator mechanism}

\( \epsilon \) \ \text{included angle between side } ab \ \text{and line } cb \ \text{on the compensator mechanism}

\( \kappa \) \ \text{angular position of side } cb \ \text{on the mechanism input link}

\( \lambda \) \ \text{angular position of the mechanism connecting link}

\( \eta \) \ \text{included angle between the mechanism connecting link and a line from } X_3, Y_3 \ \text{and } x_c, y_c

\( \phi \) \ \text{angular measure around the developed surface of the package}

\( \Phi \) \ \text{included angle of the conical package}

\( \rho \) \ \text{angular position of a line drawn from } X_3, Y_3 \ \text{on the mechanism output crank pivot and } x_c, y_c

\( \rho_p \) \ \text{density of air}

\( \tau \) \ \text{angular position of mechanism control crank}

\( \theta \) \ \text{angular position of mechanism input crank}

\( \psi \) \ \text{angle subtended by a segment of the package surface, ie the angle of the developed cone which is passed through during each time interval.}
\[ \psi \] included angle between sides ab and cb on the mechanism input link

\[ \omega_{A-K} \] rotational speed (rad/sec) of timing belt pulleys which provide geared drives to the delivery roller, drive roller, and scroll cam shafts

\[ \Omega \] rotational speed (rad/sec) of the package

\[ \chi \] angular position of side ab of the input link

\[ \zeta \] distance moved by a point on the drive tyre
Chapter 1

INTRODUCTION
Chapter 1

Introduction

The purpose of this introduction is five-fold. Firstly, it is to describe the structure of the thesis, which differs in some respects from the norm. Secondly, it is to outline the process of spinning in general, and of package winding in particular. Thirdly, it is to describe the specific problems associated with high speed winding, which forms the basis of this work. Fourthly, it is to describe the nature of the work undertaken so that each of the following sections and chapters can be placed in the context of the entire project. Fifthly, it is to relate the work of the author to associated work by other researchers.

Section 1.1 is necessary reading for anyone intending to study the thesis. Sections 1.2 to 1.5 (inclusive) may be omitted by anyone with a good knowledge of current spinning machine technology and the practice of spinning. Section 1.6 describes key elements in the Masterspinner transmission system. Section 1.7 relates the work in this thesis to that of other researchers.

1.1 Structure of the thesis.

This thesis has been written with particular attention paid to the ease with which a reader may study it's contents. As will be expected, the thesis is divided into chapters which are numbered 1, 2, 3 etc., each covering a major area of work. Chapters are divided into sections, with each section covering a particular topic relevant to the chapter. Sections in chapter 1 are numbered 1.1, 1.2, 1.3 etc., and in some cases, when a section is particularly lengthy, a "first level" section may be broken down into second level sections. For example, the third section of chapter 4 (ie section 4.3) is broken down into second level sections 4.3.1, 4.3.2, and 4.3.3. Similarly, appendices, which are numbered A1, A2 etc., are broken down into sections A1.1, A1.2 etc. Each chapter and appendix has its own numbering system. Hence, the pages in chapter 2 are numbered C2-1, C2-2 etc., whilst the pages in appendix 4 are numbered A4-1, A4-2 etc.

In addition to this general organisation, a number of specific techniques have been adopted to make the work easier to follow. In particular, there are three important methods which have been employed by the author. These are:
1. equation numbers are related to the chapter in which they are first presented. For example, the first equation developed in appendix A1 will be numbered 1.A1 and will be referred to as such whenever it appears elsewhere. This makes it easier for the reader to locate the source of an equation.

2. graphics (graphs, diagrams, photographs, and tables) are always placed adjacent to the text which refers to them. This means that some graphics appear more than once, at different parts of the thesis. The advantage of course, is that the reader never needs to search back for a graphic. Each graphic is identified using a code which contains the page number on which it appears, together with a sequence number. For example, the third graphic in chapter 1, which appears on page 7, is numbered C1-7(3).

3. although some minor analytical themes are developed in the central chapters, all extensive analyses are presented in the appendices. The purpose of this is to make it possible for a reader to avoid the mathematical proofs, if an understanding of the aims, methods, and results is all that is required. The analytical work presented in the appendices makes a significant contribution towards the value of this thesis as a PhD submission.

1.2 Commercial spinning.

Spinning is the final process of converting fibres of substances such as cotton, wool, flax, linen, and man-made staple fibres into a continuous yarn. Prior to spinning, the fibres are (if necessary) blended, cleaned, carded, drawn, and combed. The purpose of these pre-spinning processes is to combine the fibres into a continuous, loosely packed structure called a "sliver". The most important characteristic of a sliver is the way in which the fibres all lie roughly parallel to it's longitudinal axis, rather than in the random arrangement observed in surgical "cotton wool". Generally, longitudinal fibre orientation causes the sliver to have little tensile strength. What strength there is, comes from friction between the fibres, rather than from the mechanical locking present in twisted yarns.
The introduction of an acceptable level of tensile strength by the addition of twist, is the purpose of the spinning process. By far the most common spinning processes are ring spinning and rotor spinning. In the case of ring spinning, the sliver is converted into a thinner roving before processing. In rotor spinning however, the sliver is used directly as the raw material.

The output from ring and rotor spinning machines are "packages" of yarn, ready for the next processing stages prior to fabric formation by knitting or weaving. Ring spun packages are cylindrical with conical ends, and have a length:diameter ratio greater than 2:1. Packages from rotor spinning machines are either cylindrical or conical, depending upon whether they are destined for weaving or knitting, or whether the yarn is to be dyed on the package.

The rate of yarn production on spinning machines depends upon the quality and type of sliver, and the limits set by the customer on yarn quality. In general, quality decreases as production rate increases. Currently, rotor spinning machines produce yarn at up to 150 m/min, and this is commonly considered by the trade to be the maximum achievable speed due to factors inherent in the process.

In response to a desire by yarn producers for lower unit costs, spinning machinery manufacturers have invested heavily in research and development programmes with the purpose of perfecting alternative higher speed open-end spinning processes. None of these high speed (eg production rates above 250 m/min) processes are, as yet, commercially available. One of the most promising high speed processes is friction spinning. Several forms of this process are under development, with perhaps the most highly developed system being that pursued by Platt Saco Lowell in the form of the Masterspinner. Speeds of between 200 m/min and 300 m/min are possible with this machine.

Generically, rotor and friction spinning processes are called open-end spinning systems. The origin of this generic term is the method used by both processes of completely separating each fibre in the sliver, and introducing them into an air stream. This separation or "opening" enables fibres to be introduced to the rotor or friction rollers at the velocity, orientation, and spacing, necessary for successful spinning. In contrast, the roving used in ring spinning is drawn (pulled out) and twisted without total separation of the fibres. Ring spinning is not therefore classified as open-end spinning.
1.3 Package building

The packages produced on friction spinning machines are, like those produced on rotor spinning machines, either cylindrical or conical. In general, the source of a package (i.e., either a rotor or friction machine) cannot be determined by examination of its size or angle of taper, as the method of building the package may be similar regardless of which open end process is adopted.

Although details are different on machines produced by different manufacturers, the principles of package building remain largely the same, with the essentials for a cylindrical package being as shown in the C1-5(1).
Reciprocating guide
(drive cam omitted for clarity)
Drive roller shaft
Distribution bar
Drive roller
Rubber drive tyre
Delivery roller
Delivery shaft

C1-5(1) Essential package building elements
In all current commercially available open-end spinning machines, the yarn is produced at a constant rate. The process requires the yarn to be "drawn out" of the production zone (rotor or friction rollers), and this is done by passing the yarn between a contacting roller and shaft which have constant rotational speeds. After passing between these "delivery rollers", the yarn is taken over a distribution bar, and through a reciprocating guide, before being taken up by the rotating package.

1.4 The winding problem

One of the objectives whilst building the package is to introduce tension into the yarn. This tension is necessary to give the package structural stability. Because the cardboard tube onto which the package is wound does not have "sides", the package must be capable of maintaining it's own shape, even when subjected to knocks and other "reasonable" mishandling. The tension introduced during package building provides the friction forces and mechanical interlocking necessary for this stability.

Analysis of the generation of yarn tension forms a major part of this thesis. However, for now, it is sufficient to say that average tension is primarily related to the relative peripheral speeds of the package and the delivery roller shaft.

Maintaining yarn tension at a level which is sufficiently high to achieve package stability is only one of the criteria for the specification of a good package building system. Equally important, is that fluctuations in tension are kept to a minimum. This is important; firstly because uneven tension causes uneven package density, which in turn results in uneven dye take up in yarn which is dyed on the package; and secondly, reduction in tension at the ends of the package can result in "stitching", where a short length of yarn escapes from the body of the package and bridges a segment of the end. Stitching is a particularly troublesome fault in conical packages where it causes yarn breakage whilst unwinding, and will be mentioned in that context later on. It was mentioned earlier in this section that yarn tension is influenced by the ratio of package peripheral speed to delivery roller peripheral speed. In fact adopting the convention used in the industry for the purpose of building cylindrical packages:

\[
\frac{\text{Package peripheral speed}}{\text{Delivery roller peripheral speed}} = \text{Tension draft (T_d)} \ldots..(1.\ C1)
\]
However, $T_d$ is not the only parameter which influences tension.

It will be shown in sections 2.1, 2.2, and 5.1 that when building cylindrical packages, tension also depends upon:

1. the length ($Z$) of the yarn path between the delivery rollers and the point where the yarn first contacts the package surface. This length is known as the "free yarn length".

2. the velocity of the reciprocating guide which distributes the yarn across the package surface.

Because variations of yarn path length influence tension, several manufacturers of open-end spinning machines fit curved distribution bars, to keep the path length constant regardless of the position of the reciprocating guide. Platt Saco Lowell fit such bars to their rotor and friction spinning machines when cylindrical packages are being built.

When conical packages are being built, the yarn tension depends upon tension draft, yarn path length, and guide velocity, just as it does in the case of cylindrical packages. However, because the peripheral speed of the rotating cone decreases, the nearer it is measured to the apex, the effective tension draft is also decreased as the guide moves from the large to the small end of the package. Manufacturers of open-end spinning machines reduce variations in yarn tension during cone winding by fitting "compensators", which are either "passive" or "positive" in action.

A passive compensator is a spring mass device (see the diagram (C1-8(2) over the page) which is used to "smooth out" tension variations. In effect, they act as accumulators, which respond to tension variations by "taking-up" or "letting-out" yarn as the demand changes. The Masterspinner is fitted with a passive compensator which is based upon a spiral spring. This provides torsional force to a circular disc, on which are mounted two pins. The yarn passes around and between the pins, and imparts an oscillatory motion to the disc against the spring force as yarn tension changes.

Passive compensators work well whilst the yarn delivery rate is low. The Masterspinner...
passive compensator functions satisfactorily up to a yarn delivery rate of around 250 m/min. However, all spring mass systems have limitations on their speed of response. The ever changing difference between yarn supply and demand rates, means that the compensator spring must be able to accelerate the mass at a rate sufficient for it to maintain an "adequate" force on the yarn.

In this position, the compensator is holding a large amount of yarn, and is responding to a drop in tension.

In this position, the compensator is holding a small amount of yarn, and is responding to a rise in tension.

C1-8(2) Diagrammatic representation of passive compensation.

C1-8(3) The bollard compensator fitted by Platt Saco Lowell.
At a delivery rate above 300 m/min, the compensator on the Masterspinner cannot follow tension changes. High speed video prepared using SERC equipment clearly shows how loops of slack yarn develop before the compensator can move to take them up. The still photographs below are from one of these sequences, and show the effect clearly.

C1-9(4) Sequence showing effect of compensator inertia
Compensator response can be made faster by increasing spring force, or by reducing mass. However, the former is not an option for the passive compensator because the force must match the yarn tension or it will not work at all. The extent by which mass can be reduced obviously has its limits set by mechanical strength and rigidity. The current Masterspinner compensator is already a low mass system. Spring mass accounts for a considerable proportion of the total mass, and any possible reductions in the mass of the pins, disc, or the disc shaft would have little effect.

As will be shown in appendix 8, a specially curved distribution bar may be fitted to keep the yarn path length between the reciprocating yarn guide and the delivery roller approximately constant, regardless of the position of the guide. Because of the effectiveness of the Masterspinner compensator at speeds up to 250 m/min, machines set for conical packages are fitted with straight distribution bars. These cost slightly less than the curved bars, but their use means that the compensator must cope with tension variations resulting from path length changes, as well as those caused by tension draft and guide velocity changes.

As will be shown later, the combination of these variations induce a complex motion in the compensator, with ensuing high levels of compensator acceleration. These demands could be reduced by the fitting of a curved distribution bar.

The major commercial rivals of Platt Saco Lowell in the marketing of open end spinning machines are Schlafhorst (West Germany), Rieter (Switzerland), Savio (Italy), and Murata (Japan). Only the first three currently market machines to produce packages at similar acute angle as Platt Saco Lowell, and all of these now fit positive compensators to their machines. Murata produce a fasciated yarn by a quasi open end process. The resulting yarn is compact and relatively hairless, thereby giving it a low friction coefficient, and ensuring that it can be pulled off shallow angle packages which do not need compensation during the building process.

All of the positive compensators fitted to machines currently on the market run the free yarn around a guide fixed to the end of a swinging arm (see the diagram at the top of the next page). The arm is driven backwards and forwards about its pivot by a mechanical system based upon cams and levers.

One of the major problems associated with positive compensation, stems from the
need to vary the amplitude of movement as the package size increases during winding. This effect is analysed in chapter 5 and its associated appendices, but the result is that current positive compensators are fitted with feedback links, which are connected to the package support assembly. As the package grows in size, the feedback links move, and adjust the compensator mechanism so that the amplitude is reduced.

The basic means by which this is done relies upon a sliding pivot.

Positive compensators such as these work well at speeds of up to 200 m/min, but have
inherent problems for operation at higher speeds. Sliding contact, especially under conditions where adequate lubrication is problematic, is an unlikely high speed solution, but is present in these devices at the pivot and the cam. Backlash due to wear can also cause difficulties.

1.5 Specification for a solution

Having outlined, in general, the need for package building and the methods currently available, it is necessary to establish the objectives of the programme of work which is the basis of this thesis. Certain general objectives have developed from the considerations already made. These can be identified as follows:

1. a compensator system is required which will operate at speeds above those attainable by current systems.

2. tension variations during winding must be controlled within "acceptable" limits to maintain package quality.

The first of these two objectives can be quantified. Rotor spinning and friction spinning currently produce yarn at around 150 m/min and 250 m/min respectively. Although rotor spinning is generally considered to have reached the maximum production rate possible, there are a number of indicators which suggest that the friction spinning process might be developed to produce at a much higher rate than at present. In addition, several new processes are known to be under development which have potential for speeds in excess of the presently available friction process. Extrapolating the increase in production rates which have occurred over the past ten years, it is possible to forecast yarn production at 500 m/min by the year 1992. It is reasonable therefore, that this figure should be taken as an objective for the design of a new compensator system.

The second of the two objectives (ie control of tension variation) is much more difficult to quantify. The need for tension control stems from the requirement for producing packages of acceptable quality, but quantifiable measures of quality do not exist. However, defining the specification for a solution to the winding problem demands that an attempt is made to quantify important aspects of quality.
Since packages currently produced at 300 m/min are acceptable in terms of package build quality, it is tempting to define "adequate quality" in terms of a comparison with existing packages. This however, does not remove the necessity to specify quantitatively the features which are to form the basis of the comparison.

In broad terms, quality might be assessed on the basis of the following features:

<table>
<thead>
<tr>
<th>Quality feature</th>
<th>Effect of feature</th>
<th>Relevant winding parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stability</td>
<td>Package is not easily damaged by knocks or other types of misuse.</td>
<td>Adequate tension to ensure mechanical interlocking of threads</td>
</tr>
<tr>
<td>Constant density</td>
<td>When dyed on the package, yarn takes up equal amounts of dye regardless of the position in the package.</td>
<td>Tension control, to ensure that variations are within acceptable limits</td>
</tr>
<tr>
<td>Absence of stitching</td>
<td>Yarn can be unwound from the package at any speed without stitching snags causing breakage</td>
<td>Precise control of yarn guide movement, and tension control at the ends of the yarn guide stroke</td>
</tr>
<tr>
<td>Well formed corners</td>
<td>Package appearance is improved and contributes to an impression of stability</td>
<td>Precise control of yarn guide movement, and tension control at the ends of the yarn guide stroke</td>
</tr>
<tr>
<td>An even feel, in terms of package stiffness</td>
<td>Provides an easily carried out test of density distribution</td>
<td>Control of tension across the package surface</td>
</tr>
<tr>
<td>Firmness</td>
<td>A stiff package indicates a high winding tension, and hence a stable package.</td>
<td>Tension maintained at a high level throughout the package build</td>
</tr>
</tbody>
</table>

C1-13(7) Table showing the effect of winding parameters on package quality.

As can be seen from the right hand column, the parameter which influences each quality feature is tension. To accomplish the manufacture of a package of acceptable quality, tension must therefore be controlled in two respects:

1. the mean level of yarn tension as it is wound onto the package must be maintained at a level which will result in a stable package, which will unwind at high speed.
2. The deviation of tension from the mean must be constrained to lie within a range which will result in a package which will dye evenly, have well formed corners, and a consistent feel in regard to stiffness.

If these two requirements are accepted as indicators for quality, then the most convenient specification for a high speed tension control system is that it should produce packages where the mean level of tension does not drop below that attained with currently used systems, and tension deviation from mean is never greater than that experienced at 250 m/min.

It will be shown later, that the mean level of winding tension is controlled by the tension draft \( T_d \), the velocity of the reciprocating yarn guide, the stiffness of the yarn, and the length of the yarn path between delivery rollers and package. These factors are not directly influenced by either a passive or positive compensator, and therefore, the principal functional specification for a compensator designed to operate in excess of 500 m/min will be:

\[
\text{The deviation of tension from the mean during any stage of the winding of a conical package, must never be greater than that which occurs during the winding of a package of acceptable quality on the Platt Saco Lowell Masterspinner at a yarn production rate of 250 m/min.}
\]

The data which has been accepted as the standard for this comparison was supplied by Platt Saco Lowell in the form of a tension vs time plot recorded on the Masterspinner during the winding of a 28s cotton count 50/50 cotton/acrylic package. The data was recorded using a Rothchild tensometer and associated bridge, and shows the variations in tension which took place during the winding of a package from the outset to completion when the large diameter of the cone measured 300mm. The graph shows, on the horizontal time axis the diameter of the large end of the package which had been attained at certain times. The graph is over a metre in length, and is not reproduced in this thesis. The information relevant to this work is however encapsulated in the graphs shown on the next page.
Long term tension variation during package build for 50/50 cotton/acrylic 28 cotton count.

Short term tension variation during package build for 50/50 cotton/acrylic 28 cotton count.

The work described in this thesis is mainly concerned with the short term tension variations caused by yarn guide movement.

1.6 The Platt Saco Lowell Masterspinner.

The Masterspinner has a number of rotating shaft elements, all interconnected by toothed timing belts to ensure synchronisation. Only three of these main shafts are of
interest when considering yarn tension, all of these will be introduced in the following brief description.

Power is passed to the delivery roller shaft from the main motor drive. Within the limits of measuring accuracy the peripheral speed of the delivery roller shaft is the same as the yarn delivery speed, and this parameter is set to the required value by means of a variable speed gearbox on the motor.

One of the most important factors influencing yarn tension is the tension draft which, as defined in section 1.4, is the ratio of package peripheral speed to yarn delivery speed. Therefore, for a particular tension draft, the package peripheral speed will be "constant". This fixed ratio is achieved by driving the drive roller shaft from the delivery roller shaft by means of timing belts. In turn, rubber tyres mounted on drive rollers on the drive roller shaft, rotate the package which is in frictional contact with them. On a full size Materspinner, a variable speed gearbox is placed in the drive train between the delivery and drive roller shafts. This allows tension draft (and hence yarn tension) to be set to any required value. In the 4-position spin tester used at Loughborough University of Technology (LUT), tension draft was adjusted by means of change gears on a quadrant mounted lay shaft.

The final drive train of interest is from the delivery roller shaft to the scroll cam shaft. The ratio of delivery roller shaft speed to scroll cam shaft speed is also important. The scroll cam carries a ceramic guide which reciprocates across the width of the package. The velocity of the guide, relative to the surface velocity of the package influences yarn tension, as well as the structure of the package. Although a timing belt train maintains a nominally constant ratio between delivery roller and scroll cam shaft speeds, a variable speed unit causes variations in the scroll cam shaft speed.

The introduction of variations to the scroll cam shaft speed, is to prevent "patterning" of the yarn on the package surface. At certain values of the ratio "package rotational speed : scroll cam rotational speed", zones of closely spaced yarn can appear on the package surface. These "ribbons" or "patterns" cause a localised increase in package density which affects dyeing properties, and can result in yarn breakages during unwinding. The introduction of variations in scroll cam shaft speed is to reduce the tendency for patterns to form. For this reason, the variable speed unit between the delivery roller and scroll cam shafts is called the "anti-patterning device".

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Over a period corresponding to 260 revolutions of the scroll cam shaft, the anti-patterning device increases, and then decreases the shaft speed by plus or minus 2% of the mean value. The form of the variation is approximated by the following graph.

![Graph showing scroll cam shaft speed against time resulting from anti-patterning device.](image)

**C1-17(10)**  
**Scroll cam shaft speed against time resulting from anti-patterning device.**

The changes in scroll cam speed resulting from the anti-patterning device cause the low frequency cyclical variations visible in graph C1-15(9).

1.6.1 Machine element speeds

The purpose of this section is to establish the speed relationships between the shafts. In particular, the way in which the tension draft is determined will be described. Throughout this section, reference will be made to the following diagram (C1-18(11)).
The major parts of the scroll cam shaft power transmission can be identified on the photograph of the Masterspinner shown in C1-19(12).
1.6.2 Calculation of tension draft.

Tension draft may be defined as the ratio of: "drive tyre peripheral velocity / delivery roller shaft peripheral velocity".

Tension draft should not be confused with the yarn draft, which is dealt with fully in appendix 2. In cone winding, yarn draft, which is the ratio of "yarn velocity leaving the free length : yarn velocity entering the free length", changes constantly because of the change in package peripheral speed from small to large end. Yarn draft is also influenced by the winding angle, the cone angle, and the package diameter. Because of the effect of the winding angle, yarn draft will, in general be greater than tension draft.

The delivery roller shaft takes its power from the main drive via a pair of timing pulleys (only the one on the end of the shaft being shown in C1-18(11)). The rotational speed of the delivery roller shaft is set so that its peripheral velocity is the same as the
required production rate of the yarn. This shaft is 38mm diameter, so for a production rate of 150 m/min, the rotational speed is given by:

\[
\text{Speed of delivery roller shaft} = \frac{150 \times 1000}{38 \pi} = 1256.5 \text{ rev/min}
\]

The power train to the drive roller shaft is taken via timing belts and pulleys from the delivery roller shaft as shown above. Pulleys 'A' and 'B' are normally kept at a fixed ratio of teeth, and the tension draft is varied by changing gears 'C' and/or 'D'. To allow for the change in centre distance which occurs when 'C' or 'D' are changed, gears 'B' and 'C' are mounted on the same layshaft which is mounted on a quadrant swinging about the delivery roller shaft. As the quadrant arm is rotated, the centre distance between gears 'A' and 'B' remains the same, but the distance between 'C' and 'D' alters to take account of the requirements of the change gears. If the number of teeth on gears 'A', 'B', 'C', and 'D' are \( T_A \), \( T_B \), \( T_C \), and \( T_D \) respectively, and these correspond to rotational speeds of \( \omega_A \), \( \omega_B \), \( \omega_C \), and \( \omega_D \) radians /minute, then the relationship between the speeds of the delivery and drive roller shafts are as follows:

\[
\frac{\omega_A}{\omega_B} = \frac{T_B}{T_A} \quad \text{(2.C1)}
\]

and

\[
\frac{\omega_C}{\omega_D} = \frac{T_D}{T_C} \quad \text{(3.C1)}
\]
but because gears 'B' and 'C' are on the same lay shaft:

\[ \omega_B = \omega_C \]

so,

\[ \frac{T_A}{T_B} \frac{\omega_A}{\omega_D} = \frac{T_D}{T_C} \] \hspace{1cm} (4.C1)

The peripheral velocity \((V_p)\) of the delivery roller shaft is given by:

\[ V_p = \omega_A \frac{D_p}{2} \] \hspace{1cm} (5.C1)

where \(D_p\) = the diameter of the delivery roller shaft.

Similarly, the peripheral velocity \((V_q)\) of the tyre on the drive roller is given by:

\[ V_q = \omega_D \frac{D_r}{2} \] \hspace{1cm} (6.C1)

where \(D_r\) = the drive tyre diameter.

Therefore,

\[ \frac{T_A}{T_B} \frac{V_p}{V_q} \frac{D_r}{D_p} = \frac{T_D}{T_C} \] \hspace{1cm} (7.C1)

and,

\[ \text{Tension draft} = \frac{V_q}{V_p} = \frac{T_A}{T_B} \frac{T_C}{T_D} \frac{D_r}{D_p} \] \hspace{1cm} (8.C1)

On the 4-position spin tester used at the University, \(T_A = 34\), and \(T_B = 51\). A number of change wheels were available to change tension draft via \(T_C\) and \(T_D\). For example, using \(T_C = 54\) and \(T_D = 74\), the tension draft is given by:

\[ \frac{34}{51} \frac{54}{74} \frac{76.6}{38} = 0.981 \]

Thus if the yarn delivery rate (ie the peripheral velocity of the delivery roller) is 150 m/min., the peripheral velocity of the drive tyre with \(T_C = 54\), and \(T_D = 74\) is:

\[ V_q = V_p \times \text{tension draft} = 150 \times 0.981 = 147.15 \text{ m/min} \]
1.6.3 Calculation of scroll cam speed.

The scroll cam, identified in the following photograph (C1-22(14)), provides motion to the reciprocating yarn guide.

C1-22(14) The scroll cam

The relationship between the delivery roller shaft and the scroll cam shaft is important because of its influence on winding tension. This relationship is established with reference to the diagram shown on the next page (C1-23(15)).

\[
\frac{\omega_E}{\omega_F} = \frac{T_F}{T_E} \quad \ldots \quad (9.1)
\]

and

\[
\frac{\omega_F}{\omega_H} = \frac{T_H}{T_G} \quad \ldots \quad (10.1)
\]
whilst
\[ \frac{\omega_j}{\omega_H} = R = \text{ratio of the anti-patterning device} \quad \ldots \quad (11.\text{C}1) \]

and
\[ \frac{\omega_j}{\omega_K} = \frac{T_K}{T_J} \quad \ldots \quad (12.\text{C}1) \]

hence
\[ \omega_F = \omega_H \frac{T_H}{T_G} \quad \ldots \quad (13.\text{C}1) \]

and
\[ \frac{\omega_E}{\omega_H} \frac{T_G}{T_H} = \frac{T_F}{T_E} \quad \ldots \quad (14.\text{C}1) \]

but
\[ \omega_H = \frac{\omega_j}{R} \quad \ldots \quad (15.\text{C}1) \]
substituting (15.C1) into (14.C1):

\[ R \frac{\omega_E}{\omega_j} \frac{T_G}{T_H} = \frac{T_F}{T_E} \]  \hspace{1cm} (16.C1)

but

\[ \omega_j = \omega_K \frac{T_K}{T_j} \]  \hspace{1cm} (17.C1)

therefore

\[ R \frac{\omega_E}{\omega_K} \frac{T_J}{T_K} \frac{T_G}{T_H} = \frac{T_F}{T_E} \]  \hspace{1cm} (18.C1)

but if

\[ V_p = \omega_A \frac{D_p}{2} \]  \hspace{1cm} (19.C1)

where \( V_p \) = the peripheral velocity of the delivery roller (mm/min)

\( D_p \) = the diameter of the delivery shaft

and \( \omega_A = \omega_E \)

then:

\[ R \frac{2 V_p}{D_p} \frac{T_J}{T_K} \frac{T_G}{T_H} = \frac{T_F}{T_E} \]  \hspace{1cm} (20.C1)

and finally, the angular velocity of the scroll cam shaft is given by:

\[ \omega_K = R \frac{2 V_p}{D_p} \frac{T_J}{T_K} \frac{T_G}{T_H} \frac{T_E}{T_F} \]  \hspace{1cm} (21.C1)

1.6.4 Period of motion of the reciprocating guide

From a knowledge of \( \omega_K \) and the profile of the scroll cam, it is straightforward to calculate the periodic time of the reciprocating yarn guide.
If:

\[ n_K = \text{the speed of rotation of the scroll cam shaft in rev/sec} \]

then:

\[ n_K = \frac{\omega_K}{2 \pi 60} \] \hspace{1cm} (22.C1)

With the present scroll cam, the reciprocating yarn guide makes one complete cycle during 6.5 revolutions of the scroll cam shaft.

Therefore:

\[ \text{number of cycles per second} = \frac{n_K}{6.5} = \frac{\omega_K}{2 \pi 60 (6.5)} \] \hspace{1cm} (23.C1)

and the periodic time \( T_{\text{guide}} \) of the reciprocating yarn guide is given by:

\[ T_{\text{guide}} = \frac{2 \pi 60 (6.5)}{\omega_K} \] \hspace{1cm} (24.C1)

Hence:

\[ T_{\text{guide}} = \frac{2 \pi 60 (6.5) D_P T_K T_H T_F}{2 R V_P T_J T_G T_E} \] \hspace{1cm} (25.C1)

and:

\[ T_{\text{guide}} = \frac{46558.36}{R V_P} T_K T_H T_F T_J T_G T_E \] \hspace{1cm} (26.C1)

The periodic time for the guide at various yarn delivery speeds can be calculated from the above equation. Setting \( T_E=34 \), \( T_F=51 \), \( T_G=44 \), \( T_H=51 \), \( T_J=51 \), \( T_K=34 \), and \( R=1 \), table shows C1-26(16) the periodic time for the guide at different yarn delivery speeds.
<table>
<thead>
<tr>
<th>Yarn delivery speed (m/min)</th>
<th>100</th>
<th>200</th>
<th>300</th>
<th>400</th>
<th>500</th>
</tr>
</thead>
<tbody>
<tr>
<td>Periodic time of yarn guide (seconds)</td>
<td>0.54</td>
<td>0.27</td>
<td>0.18</td>
<td>0.13</td>
<td>0.11</td>
</tr>
</tbody>
</table>

C1-26(16) Yarn guide periodic times

1.6.5 Machine modifications to achieve higher yarn delivery speeds

The machine supplied by Platt Saco Lowell to Loughborough University was limited to a maximum yarn delivery speed of 298 m/min. In order to complete an experimental investigation of compensation at speeds near to 500 m/min, modifications to the machine had to be carried out. It was considered that the motor driving the delivery roller shaft was insufficiently powered to achieve the desired speed merely through a change to the gearing ratios. A larger motor was therefore supplied by Platt Saco Lowell, and fitted by university technicians. Diagram C1-27(17) indicates the changes which were then made to the gearing.
On the machine supplied by Platt Saco Lowell, timing belt pulleys 1 and 2 carried 34 and 56 teeth respectively. This gave a ratio of motor shaft speed to delivery roller shaft speed of 34:56 (ie 1.65). After the new motor was fitted, pulley 1 was replaced by one with 56 teeth, and pulley 2 by one with 34 teeth. This gave a ratio of 56:34 (ie 0.61).

The potential increase in speed resulting from this modification was therefore 1.65/0.61 giving a maximum speed of 806 m/min. However, after these modifications had been completed, it was found that electronic speed limitation circuits fitted to the machine would not allow production rates in excess of 480 m/min.

1.6.6 The scroll cam.

As described in section 1.6, the scroll cam is used to reciprocate the yarn guide across the surface of the package. The cam rotates at a speed which varies within plus or minus 2% of mean.
As will be shown later, the velocity of the reciprocating yarn guide influences yarn tension, and for this reason the path followed by the cam track is important. As can be seen from the following photograph (C1-28(18)), the yarn guide is secured to a mounting plate which, in turn, is fastened to a nylon block which slides upon a pair of constraining rods.

![Scroll cam assembly](image)

**C1-28(18) Scroll cam assembly**

Protruding from the base of the block (not visible on the photograph) is a rotatable shaft carrying a boat shaped follower which fits inside the slot which forms the cam track. As the cam rotates, the follower and hence the block and yarn guide, are made to reciprocate across the width of the package.

The displacement-time graph of the follower, together with higher derivative graphs are particularly interesting. Examination of the displacement-time graph C1-29(19) shows that the guide makes four complete passes (from left to right or from right to left) of the package in completing a full cycle of the scroll cam.
One complete cycle of the scroll cam

Guide displacement against time

The motion has been given this form so that guide reversal points do not coincide on alternate arrivals at the ends of the package. This effect reduces the tendency to produce hard ends on the package because of the extra yarn laid down during the deceleration and acceleration of the follower near the ends of the traverse. A full cycle of the scroll cam corresponds to 13 revolutions of the scroll cam shaft.

An ideal scroll cam would, in one respect, have a cam track which would impart a constant velocity motion to the yarn guide. This would ensure that the angle at which the yarn was wound onto the package surface would always be equal to a pre-determined optimum value. However, such a motion is not achievable because of the theoretically infinite accelerations which would occur at the reversal points. The Masterspinner has a track profile which imparts nominally constant velocity over the middle part of the cam travel. However, near the reversal points, inertia forces are reduced by means of deceleration and acceleration zones.

It is important for the winding of good quality packages, that the tyre mounted on the drive roller does not slip against the package surface while driving it around. To ensure that the package presents a hard surface to the tyre, the yarn guide is caused to slow down over an appropriate section of its travel. This reduction in the guide speed results in a relatively hard "band" running around the package under the tyre contact point, thereby providing the conditions for effective traction.

The cam barrel is manufactured from a filled thermoplastic into which the track is milled...
on a nc machine tool. The following graphs of velocity and acceleration are plotted from some of the 769 coordinate points used to control the machine.

![Guide velocity against time](image1)

**C1-30(20) Guide velocity against time.**

![Guide acceleration against time](image2)

**C1-30(21) Guide acceleration against time**

These graphs have axis values appropriate to a yarn delivery rate of 500 m/min.

It is interesting to note that the acceleration-time graph shows discontinuities in the region where the acceleration and deceleration zones meet the constant velocity profile. These could contribute to machine vibration and cam wear.

**1.6.7 The package support mechanism.**

The package support mechanism is shown in the following photograph and diagram.
The package is held between a pair of connected support arms. Package location is by means of two bearing mounted bosses, one on each arm, which locate inside the ends of the cardboard or plastic tube upon which the package is wound. The package is forced downwards, into contact with the drive tyre, by means of a spring. At one end, the spring is attached to the machine frame, and at the other, to an anchorage point on the support arm assembly. Damping is also provided between the support arm and
machine frame to reduce the amplitude of package vibration.

1.7 Relationship of this thesis to other work.

It is obvious to anyone surveying the history and present state of spinning and winding, that a great deal of research and development has been carried out in the perfecting of associated machines and processes. However, much of this work has been carried out by, or on behalf of, companies whose business it is to market the end result. Hence, the most frequent aspects of this research which are published, are those which assist the companies to market their products. It is relatively easy therefore, to find quasi-technical reports which praise particular machines or process, but much more difficult to obtain in-depth analytical and experimental studies of the subject.

Fortunately, some good research papers have been published, although some of these are rather dated due to moratoriums being placed upon their publication by sponsoring companies. Section 1.7.1 surveys some key literature of this type.

By far the most important source of information, and one which reflects the unpublished research work undertaken by companies, is patents. Section 1.7.2 surveys this information source.

1.7.1 Survey of technical papers.

Although little has been published about the generation and control of tension between the delivery roller and the package, there are a number of papers relating to tension between the rotor and delivery rollers on rotor spinning machines. Perhaps the most complete experimentally based publication on this subject is a PhD thesis by Afshari (21).

Of interest, but not of direct relevance, are a series of papers which undertake theoretical and/or experimental studies of tension control in warp threads during weaving. High speed weft insertion encountered on modern machines demands that a small shed height is used. This means that variations in warp tension could cause threads with low tension to droop into the shed and foul the travelling weft. Typical of such papers are Rouse (22), Lunenschloss et al (23), and (24).
Paper (22), rather than addressing the control of tension in each thread, describes a method of controlling the mean tension of all the warp threads by varying the speed of the roller onto which they are wound. Control of the roller speed is achieved by a magnetic clutch/brake fitted to the drive motor. A follower arm resting on the warp threads, and an attached transducer, send a signal proportional to thread tension to a control unit. In turn, the control unit adjusts the clutch/brake to the appropriate setting.

The experimental evaluation of yarn tension has received more attention in weaving than spinning. An example of the type of study undertaken in the determination of weaving tension is that by Lunenschloss and Schlichter (23), and a particular solution to the problem has been recorded in the trade press (24).

The use of tension gates (or "yarn brakes") is widespread in winding, where yarn can be drawn from a supply package on demand. In these cases, there is no need to accumulate yarn as in spinning, where the delivery rate varies. Investigations relating to tension gates are numerous, with papers by Bona et al (25), Simon (26), Bhargava (27), and Vlasor et al (28) being typical. Papers (25), (26), and (27) undertake theoretical and/or experimental studies of tension gates including open and closed loop systems. In general, the cyclical variations in tension with which these devices are designed to operate are of a much lower frequency than those which occur during spinning onto conical packages. Consequently, frequency response is not such an important aspect of the work covered by these papers. This point is also true for (28), which undertakes a theoretical study of a tension gate with an electronic control system. This system varies the clamping force between the two friction plates by making changes to an electro-magnetic field used to draw the plates together.

1.7.2 Patent survey.

This section outlines the major findings of a patent survey carried out in June 1987. The survey utilised the on-line database service provided by the Loughborough University of Technology Library. The outcome of the survey was compiled into a report for Platt Saco Lowell on 24 November 1987, and this section is, in the main, composed of information from the report.

In order to consider the various patents which are relevant to this thesis, it has been convenient to group them according to their principle of operation. Six groups have
been defined, although some individual patents sit uncomfortably across two groups, and others seem to warrant a group of their own.

**Group 1. Passive compensators (mechanical).**

Patents in this group are typified by that filed by Barber-Coleman (10), which relates to the compensator presently fitted to the Platt Saco Lowell Masterspinner. Interestingly, one of the major claims for this patent is that it is "self threading" due to ramps which guide the yarn over the spring loaded pins. However, this particular feature is not fitted to the PSL compensator, and threading is carried out manually by parking the pins in the threading position.

The sprung arm described in patent (6) filed by Maschinenfabrik Rieter is a further example of a passive compensator. Although the patent describes the device as being suitable for winding, there would seem to be no reason why it should not be used to aid package building on a spinning machine.

Yarn tension control is required when delivering warp threads from a beam. Patent (13) filed by Jidoshokki discloses a method of using weights and a spring to load an arm against the yarn.

All of patents (10), (6), and (13) use spring mass systems to tension the yarn. They all suffer from the limitations on frequency response, which is imposed by the inertia of the system.

**Group 2. Passive compensators (pneumatic).**

Two patents, filed by Taylor (1), and Phillips Electronic and Associated Industries Ltd (4), address the problem of system inertia by using a flow of air to tension a moving "strand". In both cases, the invention is concerned with maintaining tension in wire, whilst it is being wound onto a bobbin. These patents are outlined in this section, but are described in detail in chapter 4.

Patent (1) has the wire passing down a small diameter tube, along which a strong airflow passes in the opposite direction. The wire therefore receives "back tension" due to the influence of the air on it, and friction between the wire and the wall of the tube. In effect, this device will work like some of the yarn brakes described later. Even if the device could be made to apply controlled tension to the strand, it would not operate effectively unless the slack yarn could be collected and released as necessary.
Patent (4) is a true tensioner and compensator. During the winding of wire onto a bobbin, tension variations occur. The device provides a chamber into which a loop of yarn is sucked at low tension. The force is provided by the pressure drop across the loop, which is closely fitted to the walls of the chamber. This device, although it has some similarity with the pneumatic compensator designed by the author and described in detail in this thesis, would not work for yarn compensation. The reasons for this are described in detail in chapter 4, but are basically due to the problem of maintaining an adequate pressure drop across the loop of yarn because of its small cross sectional area.

Group 3. Friction devices (mechanical).
These devices are usually fitted to winding machines, where yarn is pulled from one package to another. If the yarn is being wound onto a conical package, the yarn velocity will vary depending upon whether the winding point is towards the large or small end. Since, however, the yarn will be drawn off the supply bobbin on demand, there will be no need to store surplus as is the case when the yarn is being "spun" at a constant rate. During winding therefore, there is no need for compensation, and tension can be introduced simply by passing the yarn through tension gates of the type described in patents filed by Otto Zollinger Inc (14), Palitex Project Co GmbH (18), Kaisha (19), and McBride Jnr. (20).

Patent (19) describes an interesting, but somewhat involved device which tensions the yarn by causing it to take a sinuous path over, and under, pins and rings. Friction between the yarn and pins produces the required drag. It is claimed that as tension increases, the relative position of pins and rings changes, thereby reducing the angle of lap of yarn around the pins, and causing the friction force to decrease.

Patent (20) refers to another, simpler, device which acts as a tension gate. In this case, gravity causes a cylindrical roller to press down upon the yarn. The moving yarn is pressed between the roller and a lower surface, and thereby has a tension induced in it due to friction. The force with which the roller presses on the yarn can be increased by placing steel balls above it, thereby adding to the mass of the downwards acting elements. Patent (14) is similar to (20) except that the yarn is pressed between an annular seating and a metal sphere. Again, additional friction force can be applied by adding more weights above the sphere.
The patent described in (18) is typical of a number of tension gates based upon similar principles. This simple device consists essentially of two plates which are spring loaded together on their flat surfaces. The yarn passes between the plates and is tensioned by the friction forces. The device is designed for use on winding machines.

**Group 4. Friction devices (electro-mechanical with feedback).**

The tension gates typified by group 3 devices are largely unresponsive to tension changes resulting from demand variations. They are, in effect, open loop devices.

Several tension gates have been devised which permit tension to be controlled by means of feedback, and examples of these are to be found in patents filed by Memminger (2), Singer (5), and Appalachian Electronic Industries Inc. (8), and (11). Patent (2) is particularly interesting. This invention consists, in the main, of a small electric motor fitted with a pulley about which the yarn is passed during its passage from the delivery rollers to the package. A tension measuring device passes a control signal to the motor causing it to rotate in the opposite direction to that in which the yarn passes. As the tension drops, the motor is caused to rotate more quickly, thereby increasing the force on the wrapped yarn, and raising the tension to the required value.

Patents (5), (8), and (11) describe inventions based upon a similar principle of operation. In all three cases, the yarn is caused to pass between two elements (either two plates or a ball and plate). Electromagnetic means are employed to vary the force with which the elements are attracted together, thereby changing the friction force on the yarn. Variation of the electromagnetic field is achieved by a control system activated by a yarn tension sensor.

**Group 5. Variable speed devices.**

These devices are typified by the inventions described in patents filed by Toray Industries Inc. (15), Industrie-Werke Karlsruhe-Augsburg Aktiengesellschaft (16), and E.I. Du Pont de Nemours and Company (17). All of them are concerned with maintaining yarn tension when winding onto cylindrical packages.

Tension is controlled by varying the peripheral speed of the package onto which the yarn is being wound. In the case of (15) and (17), each package has its own electric motor, whilst (16) uses a single motor to drive three packages. In all three cases, a transducer "senses" yarn tension, and this is converted into a signal which is used to control motor speed. Hence, if tension drops, the package speed is caused to increase,
thereby increasing tension to the required (preset) value. Conversely, a rise in tension above the preset value would result in a decrease in package peripheral speed, thereby reducing yarn tension.

The main purpose of this group of inventions is to control the package peripheral speed, and hence maintain tension draft and yarn tension at a nominally constant value. The rate of change of peripheral speed due to a cylindrical package build is very small in comparison with the short term variations in cone winding. None of the systems described are designed to make cyclic changes to the package speed at frequencies approaching the 10 Hz as would be required at a yarn production rate of 500 m/sec.

**Group 6. Other devices**

The inventions in this group appear to be individualistic in that no other patent based upon a similar principle was found.

The invention filed by Seisakusho (3) is simply a curved plate over which the yarn passes prior to it being wound onto a cylindrical package. Most producers of rotor spinning machines fit a similar device to ensure that the yarn path length from delivery rollers to winding point is constant, regardless of the position of the yarn guide. However, the plate described in (3) is shaped differently from those currently fitted, and is constructed so that the yarn path length is longest when the guide is at the small end of the package, and the shortest when the guide is at the largest end. On casual inspection, this arrangement seems logical, since there will be an increase in yarn path length at a time when package peripheral speed at the winding point is decreasing, and visa versa. An inspection of a graph of surplus yarn and path length against time (C1-38(24)) will however, show that the two variations are out of phase, thereby preventing the device from smoothing tension variations.
Phase difference between yarn demand and path length variation with the device described in (3) fitted.

The author concludes that the device described in (3) would not work.

A Patent filed by Lucas Industries Ltd. (7) describes a device which, because it relies on friction, might have been included in group 3. However, the novelty by which the friction force is applied has caused it to be placed here. The device, which is for tensioning wire, consists of two discs, each with a hole in their centre. The discs are situated some small distance apart, with the wire running through the central holes. The discs are connected together by elements (possibly wires) which are secured to the outer peripheries of the discs. Whilst one disc is fixed, the other is caused to rotate about the axis of the travelling wire, thereby causing the centre of each element to move inward. Eventually, the elements contact the moving wire and apply a frictional braking force to it.

A patent filed by J. and J.A. Porter (9) describes a special barrel cam to impart motion to the yarn guide. By causing the yarn guide to move with a higher velocity near the small end of the package, the yarn demand in that region is increased, and tension draft is maintained at a more constant level. The problem with such a device is that it will result in a package of uneven density, and an uneven winding angle. This will
affect the dye take up characteristics, and may influence the ability to unwind at high speed.

The invention described in a patent filed by PPG Industries Inc. (12) purports to control tension in mono-filaments when they are being wound onto a layered package. This control is achieved by passing the filament along a groove which is cut in the periphery of a disc. The disc is caused to rotate either with or against the motion of the filament, thereby imparting a frictional force to it. As with other friction devices, the invention is not suitable for spinning, where yarn accumulation is required.
Chapter 2

THE GENERATION OF YARN TENSION
Chapter 2
The generation of yarn tension

Chapter 1 has outlined the requirement for maintaining yarn tension within acceptable limits to ensure that a package of suitable quality is produced. The generation and control of yarn tension is therefore, a key factor in machine design, and must be thoroughly understood before compensator design can commence. Sections 2.1 and 2.2 summarise the detailed analytical developments of supporting appendices 1, 2, and 3, and provide the basis for the conceptual and detail design work in subsequent chapters. Sections 2.1 and 2.2 describe the parameters which affect tension based upon the assumption that no compensator (either passive or positive) is fitted. Section 2.3 of this chapter then examines the effect of including a passive spring/mass compensating device of the type currently fitted.

2.1 Steady state and step response.

Appendix 1 develops an analytical model which explains how tension may be generated in a strand which is passing between the nips of two pairs of rotating rollers, as shown in the following diagram (C2-1(1)).

Tension is introduced if the peripheral speed of the second pair of roller is greater than that of the first pair. If \( P_q > P_m \) then as the strand passes through the nip of the second pair of rollers it will need to have a smaller diameter than it had when passing through
the first pair if the mass flow rate through the two pairs is to be equal. Under steady state conditions (i.e., the tension is constant), the mass flow rate through the first and second pair of rollers must be equal. The necessary reduction in the yarn diameter at the second pair of rollers is achieved by an increase in strand tension.

The second graph, which is extracted from appendix 1, shows how a steady state tension is achieved which depends upon the ratio of the speeds of the two pairs of rollers. In the case of both curves, the peripheral speed of the first pair of rollers is 8333 mm/sec which corresponds to a strand velocity of 500 m/min. The initial tension for both curves is 10 grams.

![Graph showing the effect of roller peripheral speed on tension](image)

**C2-2(2) Effect of roller peripheral speed on tension**

An important outcome of the analysis of appendix 1 was the establishment of the relationship between tension and the free strand length. In fact, the steady state tension resulting from particular peripheral velocity settings of the two pairs of rollers is independent of the length of the strand path (the 'free yarn length') between them. However, the time taken for the steady state to be achieved after a disturbance to the peripheral velocity of one or both pairs of rollers is affected by the free yarn length. The following graph shows this effect.
C2-3(3) Effect of free strand length on transient response

The major conclusions of appendix 1 with respect to the generation of tension in the strand are:

- the steady state tension induced in a strand gripped by and fed between two pairs of driven rollers depends upon the relative peripheral velocity of the rollers

- the same steady state will be achieved, regardless of the initial value of tension in the strand

- the rate at which the initial tension approaches the steady state is proportional to:
  - the difference between the initial tension and the steady state tension
  - the peripheral velocity of the rollers

- the rate at which the initial tension approaches the steady state is inversely proportional to:
2.2 Response to continually changing output conditions.

Section 2.1 and appendix 1 are concerned with developing an understanding of the use of differential speed to generate tension in a strand. The model is based upon the analysis of the changes which occur when the strand is passed between two pairs of rollers which are either rotating with constant (but not equal) peripheral speed, or are subjected to step changes in peripheral speed. This analysis is essential in gaining an understanding of important parameters, but does not fully represent the conditions when winding onto a conical package.

In the construction of conical packages, the yarn velocity through the first pair of rollers (the delivery rollers) is constant. However, the rate at which the yarn is demanded by the package (this being equivalent to the velocity through the second pair of rollers) is constantly changing due to the way that the peripheral velocity of the package is different at all points across its width.

By dividing the surface of the cylindrical package into discrete patches, and examining the manner in which the yarn demand velocity changes as the yarn guide traverses across its width, appendix 2 develops a series of equations which allow tension to be calculated. Equation 15.A2, which is reproduced below, shows how yarn demand velocity \( V_y \) depends upon other system parameters.

\[
V_y = P_{qn} = \frac{1}{\partial t} \left( h_n^2 + w^2 - 2h_n w \cos \alpha \right)^{0.5}
\]

where:

- \( V_y \) = yarn demand velocity
- \( P_{qn} \) = the peripheral speed of the package during the \( n \)th time interval
- \( w \) = the length of each segment on the package surface

• the free strand length
• $h_n =$ the width of the $n$th segment
• $\alpha =$ an internal angle in a segment of the package surface
• $\delta t =$ the time interval during which the yarn is laid down in segment $n$

In equation 15.A2, $w$, and $\alpha$ are constant, but $h_n$ increases towards the large end of the package. Hence yarn demand velocity also increases with increasing distance of the yarn guide from the small diameter end.

Appendix 2 develops a computer program called "TENSION" based upon the mathematical models, and this allows predictions to be made of tension variation.

Graph C2-5(4) which is a reproduction of A2-24(13) shows the theoretical variation in yarn tension predicted by "TENSION", for the case of a straight yarn guide bar. The key in C2-5(4) gives the small diameter of the package and the direction of the guide travel.

![Graph C2-5(4) Theoretical tension variation with straight guide bar](image)

The model predicts that tension will be at a peak when the guide is approaching the mid-position of the package from the large end, and will be at a minimum moving from the small end to the mid-position. This effect is to be expected because of the difference in peripheral speed of the package from small to large ends. The model also predicts that the straight distribution bar will cause further tension increases towards the ends. This effect is particularly obvious in C2-5(4) for the largest diameter package.
The model was checked by undertaking experimental tension measuring on the Masterspinner. Graph C2-6(5) which is a reproduction of A2-28(17) shows the result of the experimental check.

C2-6(5) Experimental yarn tension with straight guide bar

The experimental curves of graph C2-6(5), which shows the tension variation when the yarn guide is moving from the small end to the large end, reveals the same general trends as those produced by the theoretical model in C2-5(4), with tension increases at both ends and low or zero tension near the centre. An increase in package size results in a general reduction of tension, and this is again predicted in C2-5(4). Theoretical and experimental tension curves for a profiled curved bar (as analysed fully in appendix 8) show the same similarities. Graph C2-7(6) shows the output of "TENSION" in the prediction of tension with a curved bar, and is a reproduction of A2-29(20).
C2-7(6) Theoretical tension variation with a curved guide bar

As would be expected, now that the stretching effect of the straight guide bar has been removed, the prediction is that there is zero tension at the small end of the package where the yarn demand velocity is less than the delivery velocity. The model also predicts that tension will reduce as the package increases in size, and that the variation of tension in respect to guide position varies depending on whether the guide is moving from left to right (small end to large end) or from right to left (large end to small end). The key identifies the curves by indicating the package diameter at the small end and the direction of guide movement. Hence, "100 l to r" identifies the curve for a package of 100 mm diameter at the smaller end with the guide moving from left to right, whilst "200 r to l" identifies the curve for a package with a smaller end diameter of 200 mm with the guide moving from right to left.

The next graph (C2-8(7)) shows a graph of the experimental variation of tension against guide position with a curved guide of the profile determined in appendix 8 fitted to the Masterspinner.
2.3 The effect of passive spring/mass compensation.

Most manufacturers of high speed open-end spinning machines fit a compliant element into the yarn path between the delivery rollers and the point of take up on the package. This is true even when the machine is fitted with a positive compensator, and at least partly reflects the inadequacy of their positive compensator in moving with the precise motion required to avoid tension variations during the cycle of the yarn guide.

In the case of the PSL machine, the passive compensator is the only compensator fitted, and its motion provides all of the variation in yarn path length which occurs during the cycle. For system equilibrium, the force applied to the yarn by the spring in the compensator is always directly proportional to the tension in the yarn, and because the spring force changes throughout the extension/retraction cycle, it follows that yarn
tension can never be kept constant with this type of device. The purpose of the passive compensator is therefore to act as an accumulator of yarn which is paid out on demand against variations in tension. This reduces (or smooths) the violent fluctuation which would otherwise occur.

2.3.1 A simple mathematical model of spring mass compensation.

The motion of a passive compensator is forced upon it by the variations in supply and demand of the yarn which passes through it. The compensator may, therefore, be considered as a spring mass device subjected to forced oscillations. The forcing frequency in the case of a yarn compensator is a function of the variations in yarn demand caused by variations in package peripheral speed and yarn path length, plus the perturbations due to anti-patterning activity. However, because these variations are basically of a sinusoidal nature, this analysis is based upon such a motion. There is no doubt that the assumption of a sinusoidal forcing function under-estimates some of the high accelerations which are present in the real function. The analysis does however help in providing a better understanding of the factors involved in passive compensator design, and the closeness of the results to the observed behaviour of the compensator suggests that the approximation is near to reality.

The passive compensator is represented by the spring mass system shown in the following diagram. The compensator has mass $M_C$ and stiffness $K_C$. In the diagram, the compensator is shown disconnected from point 1, which will provide the compensator motion (ie it provides the forcing function). The compensator is shown with it's spring in the relaxed position (ie zero extension), and the distance $x_0$ is therefore the minimum spring extension which can occur when forcing is present. Therefore, $x_0 K_C$ is the spring pre-load.
The equations of motion of point 1 are:

\[ x_c = \frac{A_c}{2} - \frac{A_c}{2} \cos(\omega_c t) \]

where \( A_c \) is the length of stroke of the motion.

\[ \frac{dx}{dt} = \frac{A \omega_c}{2} \sin(\omega_c t) \]

\[ \frac{d^2x}{dt^2} = \frac{A_c (\omega_c)^2}{2} \cos(\omega_c t) \]

With the compensator driven by the yarn, the situation is as shown in C2-11(9).
The equation of motion for the compensator is now given by:

\[ M_c \frac{d^2 x}{dt^2} + K_c (x_c + x_0) = F_c \]

Where \( F_c \) is the force applied to the yarn by the compensator.

It will be obvious that the force applied by the compensator to the yarn should always be positive, i.e., to the right in C2-11(9). Since the yarn cannot take a compressive force a negative value of \( F_c \) means that the compensator and yarn have lost contact, and that a loose loop of yarn with zero tension is forming.

The following graphs plot the compensator's equation of motion.
C2-12(10) Variation in theoretical acceleration force throughout the compensator cycle for various values of compensator mass (1 cycle in 0.11 sec at 500 m/min delivery speed)

C2-12(11) Variation in theoretical acceleration force during the compensator cycle for various values of angular frequency

The two graphs clearly show the effect upon the force required to accelerate the
compensator by increasing its mass and rotational frequency. In cases where the negative acceleration force is greater in absolute magnitude than the returning force available in the compensator spring, the compensator will fail to remain in contact with the yarn because it will not be able to accelerate at a sufficiently high rate. Reducing the compensator mass will also reduce the magnitude of the largest negative acceleration force, thereby allowing the device to run at a higher speed before the problem described is encountered. However, the currently used bollard compensator has a mass of less than 10 gram, and prospects for reduction are small.

The next section illustrates the problems of using the passive compensator at high speed by an experimental investigation on the Masterspinner.

2.3.2 An examination of compensator motion on the Masterspinner.

An experimental investigation was carried out to determine the motion of the passive compensator fitted to the Platt Saco Lowell Masterspinner. A graduated bezel was fitted to the fixed external housing of the compensator, and a small pointer secured to the moveable pin carrying inner disk. To enable the angular motion of the inner disk to be observed, a high speed video sequence was recorded at 200 frames per second. The following diagram shows the layout, and the datum from which angular displacements of one of the compensator pins was taken.
C2-14(12) **Bollard compensator and datum for measurement of rotation**

An examination of the high speed video was undertaken using a frame-by-frame analysis of the angular displacement of the pins and the corresponding position of the reciprocating yarn guide over a complete cycle of the guide cam. The following table presents the data collected by this investigation.

<table>
<thead>
<tr>
<th>Guide position</th>
<th>Angular rot'n of guide (deg')</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>69</td>
</tr>
<tr>
<td>1</td>
<td>30</td>
</tr>
<tr>
<td>2</td>
<td>20</td>
</tr>
<tr>
<td>3</td>
<td>25</td>
</tr>
<tr>
<td>4</td>
<td>55</td>
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<td>5</td>
<td>65</td>
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<td>6</td>
<td>69</td>
</tr>
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<td>7</td>
<td>71</td>
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<td>8</td>
<td>80</td>
</tr>
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<td>40</td>
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<td>10</td>
<td>26</td>
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<tr>
<td>15</td>
<td>66</td>
</tr>
<tr>
<td>16</td>
<td>69</td>
</tr>
</tbody>
</table>

C2-14(13) **Table of guide position and bollard rotation with a straight distribution bar**

Guide positions during the full cam cycle are defined on the following diagram,
C2-15(14) **Identification of yarn guide positions during 2-cycles of the guide (1-cycle of the scroll cam)**

and the data is plotted on the following graph.

C2-15(15) **Compensator rotation at different positions of the yarn guide**
Graph C2-15(15) clearly shows the "irregular" and rather complex motion of the compensator which results from the combination of yarn demand variation and the stretching effect on the yarn by the straight distribution bar. As the guide moves away from the small end of the package (from position 0 to 2, and from 8 to 10), yarn demand is less than supply because the guide is at the small end of the package. In addition, the yarn path length is decreasing over this stage of the motion because the point where the yarn passes over the distribution bar is moving towards the centre. Both of these factors contribute to a reduction in yarn tension, and when they act in conjunction the compensator needs to respond rapidly to remain in contact with the yarn (this effect being shown clearly in C2-15(15)). The graph indicates that this stage requires the highest acceleration from the compensator, and the high speed video confirms that it fails to keep contact with the yarn at delivery speeds above 350 m/min. The solution to this problem is either a compensator of ultra low mass, or a positive compensator capable of operating at the required speeds. This thesis examines aspects of both of these options.
Chapter 3

DESIGN SPECIFICATION AND CONCEPTS
Chapter 3
Design specification and concepts

In chapter 1, the subjective specification of an "acceptable" package was discussed. This, together with the understanding of the variation in yarn loop size required to maintain constant tension which was developed in chapter 2 and its associated appendices, provides the basis of a specification for the design of a compensator. There are however, many more factors which must be addressed in specifying requirements, and these are the subject of section 1 of this chapter.

After the establishment of an initial specification in section 3.1, section 3.2 contains descriptions of the major concept categories developed with the objective of finding an engineering solution to the problem. Finally, section 3 compares the concepts against the specification, and describes the reasons why some were selected for detailed investigation in the subsequent chapters and appendices of this thesis.

3.1 The initial specification
3.1.1 The specification of power requirement
With 144 spinning stations on a full size Masterspinner, the power consumption of the compensator is of obvious importance. Ideally, the new device would operate within the spare capacity of the machine power units currently fitted. The implications of designing a device with "large" power requirements may be seen as a need for larger or additional electric motors or pneumatic systems. It is practically impossible to place a realistic value on the maximum acceptable power consumption of each compensator when a step increase in machine speed is under consideration. The important measure therefore, must be the total cost of the new compensator in terms of its value to a higher speed machine.

3.1.2 The specification of cost
The basic objective of the new compensator design is to facilitate the building of packages at twice the current rate of 250 m/min. At 1988/89 prices, each station on a Platt Saco Lowell Masterspinner costs in the region of £1600 to a customer (calculated by dividing the machine selling price by the number of stations). Assuming that all other factors (eg yarn quality) remain unchanged at the higher production rate, it would be arguable that a customer might be prepared to pay £3200 per station for a 100% increase in speed. However, a major influence on the decision to design a faster
machine must be the prospect of increasing market share against international competition. In the absence of market intelligence on the ability of other companies to enter the market with competitive high speed machines, it is proposed that a 50% increase in cost will give an acceptable market advantage if associated with a 100% increase in productivity.

Of the resulting £800 per station charged for speed increase, there will be fixed and variable constituents. The fixed constituent will be required to provide improved main drive units, pneumatic compressors and suction units, and other machine elements the cost of which is largely independent of the number of stations. The division of the available £800 cost increase between these major engineering systems is difficult at the early design phase, and only rough estimates can be made. The only objective apportionment is based upon the division of total costs as they stand at present. The following pie chart indicates this.

![Pie chart](image)

**C3-2(1) Proportion of engineering costs allocated to existing elements**

If the added cost for the new compensator is 10.26% of £800, then this gives a figure of £83 which, at a mark up of 2:1 represents a prime manufacturing cost of approximately £28. Since the prime cost of the existing compensator is £40, the total allowable prime cost for the new high speed compensator is £40 + £28 = £68.

**3.1.3 Specification of the effect on yarn quality**

A primary requirement for the new compensator is that it must not cause any measurable deterioration in yarn quality. This is to be true for all measures of quality,
but especially cleanliness, strength, and appearance. In practice, maintaining yarn cleanliness demands that the compensator design avoids the possibility of yarn contamination with oil, grease, or other substances which are a part of the compensator system. The minimisation of the effect of the compensator on yarn strength and appearance implies that the structure of the yarn is unaffected by its passage through the device. This itself implies that the yarn is not to be subjected to forces, or friction, or bending through small radii of curvature, which will cause structural damage.

3.1.4 Specification of design integrity

The integrity (definition: soundness) of a design covers those aspects which relate to its ability to operate within defined specified functional limits over a specified period of time. The problem is to define generally acceptable definitions of a "specified period of time".

In general, the life of a major sub-system of a machine is expected to have little variation to the expected life of the machine as a whole. There are of course exceptions to this. For example, motor vehicle exhaust systems are not expected to last for the life of the vehicle. This 'acceptable' reduced expectation is due to both their perceived operational function (ie a harsh environment), and the low cost of replacement. An added complication is the necessity to define 'life'. Taking another example from the motor car, the clutch sub-system is not considered to have reached the end of its life when the plates need renewing. The requirements for the replacement of components operating in harsh environments is however usually seen as part of a routine maintenance programme. Therefore, whilst life should be specified to reflect the overall machine life, attention will need to be given to specifying acceptable replacement periods and maintenance schedules for components operating under harsh conditions. The following specification of these factors is offered as an acceptable basis on which to proceed during the conceptual and creative design phases.

1. compensator sub-system life - 10 years
2. minimum maintenance interval for any component or part of the sub-system - 6 months
3. maximum cost of each maintenance operation per station (materials and semi-skilled labour) - £3.40 (based upon 1.25% of...
Within these defined life, maintenance, and cost figures, the compensator must operate within the limits of functionality defined throughout this chapter.

3.1.5 Specification of operational safety

The Health and Safety at Work etc Act of 1974 lays down clear duties for the designers, manufacturers, and suppliers of machinery of all types. Any person, in any of these categories, who places other persons in risk as a result of his or her work is liable to be prosecuted. Duty falls under three main headings. They must:

- ensure in so far as is reasonably possible that the article is so designed and constructed as to be safe and without risks to health when properly used

- carry out or arrange for the carrying out of such testing and examination as may be necessary for the performance of duty

- take steps as are necessary to secure that there will be available in connection with the use of the article at work adequate information about the use for which it was designed and has been tested, and about any conditions necessary to ensure that, when put to that use, it will be safe and without risk to health.

The first of these three clauses obviously places prime responsibility onto the designer of the "article". Furthermore, it is not reasonable to claim that a person injured in using the machinery was not using it properly when he or she "fell into it". The machinery must therefore not only be without risk to health when it and the operator are both behaving in the way foreseen by the designer, but also when the operator behaves in an unforseen way.

In practical terms, the compliance with the Act will mean that the design of the compensator must be such that health and safety is not put at risk. It should not be possible for a worker to become trapped by, or drawn into the device, and thereby injured, when carrying out operations necessary to the compensators function, or when in its proximity for any other reason. Additionally, the compensator must not produce noise emission at a level which is detrimental to health.
The first general requirement is for employers to ensure that employees are not subjected to noise levels above $90\text{dB}(A)_{L_{eq}(8\text{hr})}$ (ie an average of 90 decibels over an 8 hour day), or to instantaneous pressures greater than 600 Pa. Where this is not possible by means of machine design, other methods must be employed (eg ear protectors). If the exposure is likely to be above $105\text{dB}(A)_{L_{eq}(8\text{hr})}$ the employer must provide for audiometric testing to be carried out. It is more than likely that in the not too distant future, EEC regulations will require machine noise to be reduced to less than $78\text{dB}(A)_{eq(8hr)}$ if ear protectors are not to be mandatory. It is therefore taken as a requirement for the compensator design that combined noise emission does not exceed $78\text{dB}(A)_{eq(8hr)}$.

3.2 Concept solutions to the compensation problem
This section describes the design concepts developed and assessed by the author prior to undertaking detailed analysis and/or synthesis. As should be the case in the propagation of ideas, the "brainstorming" techniques used to develop the ideas allowed for no criticism until the concept had been recorded and given positive consideration. Those ideas which were dropped from the list of potential solutions at an early stage because of fundamental design problems, are not included in the following exposition.

As with all tasks, time was an object, and the choice of ideas for detailed development was influenced not only by the potential of the idea to yield an acceptable solution, but also by the perceived timescale for the research, design, and development, and whether this timescale fitted within the framework of the SERC supported programme. In practice, this constraint exerted less influence on which ideas were chosen than on the number of ideas which were selected for detailed evaluation.

The sketches of the concepts referred to in the following sections are presented on pages 6, 7, 8, and 9. These four pages are, in the main, a collage of the original concept drawings produced at an early phase of the work, and as such do not pretend to indicate more than the basic principles under consideration.
CONCEPTS 2
MECHANICAL COMPENSATION & AMPLITUDE CONTROL.
CONCEPTS 3
PNEUMATIC COMPENSATION

CHAPTER 3 PAGE C3-8
MECHANICAL AMPLITUDE CONTROL

PIVOTING ARM

OPTICAL ENCODER OR POTENTIOMETER

TDCO-GENERATOR.

CONCEPTS 4
AMPLITUDE CONTROL & ELECTRONIC SYSTEMS.
3.2.1 Mechanisms and cams
Strictly speaking, a cam is a mechanism, but because of its particular properties and applications it is convenient to consider it separately. Concept sketches v, vi, vii, viii, ix, and x show ideas for a number of cam driven compensators. The advantage of cams, whether radial, face, or barrel, is that they may be machined to produce a perfect (ideal) output motion. There would therefore, be no reason why a cam should not be produced to give the exact motion predicted by the analyses presented in appendix 4. The critical problem in using a cam drive for the proposed compensator is its lack of ability to withstand the high acceleration forces which will be present. The extent of this difficulty is illustrated by the existing yarn guide cam at high speeds. This guide currently operates at around 5 Hz with a yarn production speed of 250 m/min. Increasing the speed of the machine to speeds in excess of 400 m/min is known to result in a marked and unacceptable decrease in cam life. At the desired production rate of 500 m/min the guide operates at 10 Hz, and company engineers have no doubt that a new design concept is required for providing yarn guide motion at this speed. This observation becomes relevant when it is noted that the mass of the yarn guide and its associated mounting plate and follower have a combined mass which is probably a factor of ten less than anything which can be envisaged for the moving parts of the compensator.

There can be no doubt that a spring loaded cam of the type shown in concept sketches vi, and viii would be unfeasible due to the high spring forces necessary to ensure that the follower maintained contact with the cam surface. Equally, a barrel cam (sketches ix and x) would be likely to suffer from the breakdown of the track surface at points close to where maximum acceleration occurs. In addition to the forces due to high acceleration and component mass, there is the added loading which takes place as the follower(s) move over from one track surface to the other.

A number of initial concepts for linkage drives are indicated in sketches i, ii, iii, and iv. With the potential for partial balancing, the use of polymers to reduce mass, and the use of high grade rolling bearings to eliminate the sliding, skidding, and bounce which may be a problem with high speed cams, linkages offer a more realistic means of providing high speed compensator motion at the desired speeds. The main problem with linkages is that, unlike cams, they generally cannot exactly provide the desired displacement-time characteristics. A linkage solution to the compensator problem can only be approximate, and it becomes important to know how close a particular solution
is to the ideal, and what the result of the error is on yarn tension.

Sketch i illustrates a 4-bar mechanism which could be used to drive the compensator. Regardless of the accuracy with which such a mechanism could match the ideal requirements of motion, there remains a problem of providing a means of decreasing the amplitude of motion as the package grows in size. A number of means of achieving this variation have been identified, and these are discussed in section 3.2.4. Whilst any configuration of the 4-bar mechanism only has the scope to produce a motion with a fixed amplitude, 5 and 7-bar mechanisms are able to provide variable output amplitude. Such mechanisms are indicated in sketches ii and iii. In general, increasing the number of links in a mechanism provides the designer with increased opportunities to synthesise an output motion which is closer to the ideal than would be the case for a smaller number of links. This advantage however, is balanced by a resulting increase in mass (and hence acceleration force), a greater number of bearings (with an associated increase in cost), and increased backlash due to the increased total of bearing clearances. The design task therefore is to select a mechanism with the minimum number of links which will provide a motion sufficiently close to the ideal to result in acceptable tension variations.

Sketch v shows a scotch yoke, which may be used to produce simple harmonic motion. This device however is not suited to high speed operation because of the large amount of sliding which takes place between the pin and the yoke.

Sketch xx shows a rather different device where the yarn runs directly in the slot of a female helical barrel cam, thereby altering the length of the yarn between runners 'b'.

3.2.2 Pneumatic and hydraulic devices.

Providing compensator motion by the action of pressure on a piston moving in a cylinder is attractive for a number reasons. Sketch xi shows a single acting cylinder where reciprocating motion is obtained by alternating positive pressure and suction on one side of the piston, with the option of using a spring to aid piston return. Sketch xii shows a double acting cylinder where positive pressure can be applied to either side of the piston.

Both of these devices can be driven by either air or oil. The required operating frequency of 10 Hz is however beyond the capability of pneumatic devices bearing in
mind the required accuracy of motion and the compressibility of air. Cylinder manufacturers and suppliers were asked to evaluate the design requirements against the known characteristics of their products, and all stated their belief that a solution based upon this type of device was not possible with either pneumatic or hydraulic pressure. In addition, the author has reservations about the use of hydraulics in close proximity to the yarn path from the point of view of cleanliness.

3.2.3 Motor driven devices.
Sketch xvi shows a motor carrying an arm at the end of its shaft. It would be possible to pass a loop of yarn around the pin at the end of the arm and affect compensation by rotating the motor. Continuous motor rotation in one direction would not provide the desired change of yarn length in the loop. However, a solution could be envisaged where the motor shaft oscillated, by rotating first in one direction and then in the other. The design may be one where there is only a single pin at the end of an arm which extends in one direction as in sketch xvi, or the arm may extend in two opposite directions, with a pin at each end as shown in sketch xvii.

It is conceivable that the motor on such drives might be dc-servo, stepper, pneumatic, or hydraulic. However, the relatively high inertia, and the problems identified in the previous section makes it unlikely that pneumatic or hydraulic drives would provide a satisfactory solution.

Sketch xiix shows a linear motor, which is an obvious extension of the idea of a motor driven compensator.

3.2.4 Devices to control motion amplitude.
The need to reduce the amplitude of variation in the size of the yarn loop in the compensator is reflected by the devices in this section. The devices can be broken down into two main categories. These are: 1. actuating devices which bring about a change in compensating amplitude when provided with the necessary information about (say) package size, and 2. devices which provide the feedback of necessary information to the actuating device.

3.2.4.1 Actuating devices.
The double crank mechanism shown in sketch iv provides an inbuilt amplitude varying function. This type of mechanism is analysed in detail in a paper by the author (29).
The control however tends to be rather crude when only two cranks are present, and it is thought likely that three input crank stages might be necessary to provide the necessary motion and control of the output member.

Sketch xviii outlines a rotating mechanism where the radius of action of the point about which the yarn passes may be decreased by moving the ends of the mechanism apart. The drawback with this device is that the action is basically SHM and cannot easily be modified. In the form shown, there is the additional problem that the yarn could not be threaded behind the rotating guide point once the end had been made. It might however, be possible to conceive a cantilevered form of the device where access by the running yarn might be possible.

Sketch xix shows an interesting development of the previous concept. In this case the compensator is also the yarn distribution bar over which the yarn runs immediately prior to being wound onto the package. The bar would be manufactured from a flexible material, and would rotate with a frequency equal to the guide traverse rate. By maintaining the appropriate phase angle between the bar and the guide, it might be possible to accomplish the changes in yarn path length necessary to maintain constant yarn tension. The decreasing compensative amplitude associated with increasing package size would be achieved by moving the ends of the rod apart, thereby reducing the effective radius of operation.

Sketches xx, xxii, and xxiii illustrate the principle of moving the yarn runners to achieve changes in the amplitude of compensation. Sketch xxiii shows the principle clearly. A loop of yarn is formed by passing the yarn through point 'a' which is attached to the output member of a compensator. Point 'a' reciprocates, thereby changing the size of the loop (which is the yarn length b to a to b). For a particular amplitude of motion of point 'a', the resulting amplitude of length of the yarn loop can be reduced by moving the runners 'b' further apart, or by maintaining the distance between them constant but bringing them closer to 'a'.

Sketch xxi shows a slotted link, whereby a fixed amplitude of motion on the input side can be modified on the output side by changing the position of the adjustable pivot point. As discussed elsewhere, this device is not suitable for high speed operation due to the high relative sliding velocities.
3.2.4.2 Feedback devices.

The amplitude of compensation must reduce as package size increases. There is a need therefore to measure a parameter which is directly associated with package size and to feed back this information to the actuating device. The parameter to be measured may be the rotational speed of the package, the size of the package, or the variation in yarn tension. If the purpose of the measurement is to provide control to the type of device already discussed, the first two options seem to offer the most practical and cost-effective solutions.

Sketch xxx shows a tacho-generator fitted to the package arm and taking input from the rotation of the package. Decreasing package rotational speed indicates a growth in package size (or possibly slip between the package and the drive tyre) and this information is available as feedback to the actuator.

There seems to be little advantage in considering the adoption of the idea of sketch xxx in preference over a device which measures package size directly. Sketch xxix shows a device where either an optical encoder or potentiometer is fitted to the package support arm in a position where it can take input from the relative rotation of the arm support shaft.

Alternative mechanical devices to the concept shown in xxix are shown in sketches xxviii and xxix. The first of these shows how feedback might be accomplished by an arm pivotally connected to the package support arm, while the second shows how a pin at the end of the arm might run in a slot of the appropriate shape on the arm.

3.2.5 Electronic systems.

Sketches xxxi and xxxii illustrate the type of solutions which might be envisaged in a complete electronic system. Sketch xxxii shows an open loop system where the stepper motor 'D' is driven by a controller 'B' which takes the necessary information from a "look-up" table. The data from the look-up table may be accessed in a sequential manner, the accessing of information being kept in phase with the guide movement by means of proximity switch 'A'. Information about package size is gathered by device 'C' (sketches xxix or xxx) and is used to tell the controller which data set to access.

Sketch xxxi shows a system where a tension measuring transducer 'T' provides
continuous information to controller 'A'. The information from 'T' to 'A' will be in the form of tension data which will carry both the mean and cyclical tension values. It is conceivable that the mean value may be separated and used to control the speed of individual motors driving each package instead of the friction tyre system used currently. This method would enable the mean tension value to be maintained at a constant level. By subtracting the mean from the cyclical component, controller 'A' could provide the stepper motor driver 'B' with information necessary to control the motion of motor 'C'. In addition to the concepts of xxxi and xxxii, it is possible to envisage hybrid solutions where a yarn tension transducer is used to update the information in a look-up table rather than for the direct control already described.

3.2.6 Passive compensation

Sketches xiiv, xiv, and xv illustrate variations on the existing theme of passive compensation. Sketch xv illustrates a bollard compensator of the type currently fitted to the Masterspinner. The use of a spiral spring can however be extended to alternative forms of the device. The sketch shows that the yarn may be passed around two pins as is currently the case or merely around a single pin secured to an arm. The main advantage of a spiral spring over other types of spring arrangements is that the opportunity to package a long length of spring in a small space provides a "flat" spring force (quasi constant rate), thereby matching the requirements more closely than would otherwise be the case. Sketches xiiv and xiv however, show that other options may be considered including cantilever and helical compression and tension springs. All of these devices however are likely to suffer from the mass / inertia problems associated with the currently used device at high speeds.

The sketches on page C3-8 all relate to the use of pneumatics as a means of providing passive compensation devices with a sufficiently low mass to operate successfully at high speeds. The straight sided suction tube shown in xxiv used a flow of air to draw a loop of yarn into the tube. The principle is that the size of the loop will grow and reduce as tension changes. The principle has however been shown not to work. The reason for this failure is discussed fully in the chapter on pneumatic compensation. The device shown in sketch xxiv is a novel variation on the suction tube principle in which the tube is turned into a tapered nozzle, and a disc is introduced to "capture" the yarn and provide a large projected area across which to develop an increased pressure drop.

One of the attractions of considering a pneumatic compensator is the possibility of
producing a device which, as well as providing compensation, will permit the storage of large lengths of yarn during the splicing process. Sketch xxvi illustrates a concept for the design of such a device where a yarn loop is sucked down a coiled tube which provides a compact storage reservoir which may be attached to the side of the compensator. To prevent the yarn loop twisting around itself, it is proposed that the bore of the tube is fluted so that each "leg" of the loop rests in a separate "chamber" on the tube wall. The remaining sketches on page C3-8 are discussed at appropriate stages of the thesis.

3.3 Combination and evaluation of ideas.

The chart shown on the next page illustrates the possible combinations of compensator and amplitude control concepts.

<table>
<thead>
<tr>
<th>Prime motion</th>
<th>xx</th>
<th>xxii</th>
<th>xxiii</th>
<th>xxiv</th>
<th>xxv</th>
<th>xxvi</th>
<th>xxvii</th>
<th>xxviii</th>
<th>xix</th>
<th>xxi</th>
<th>xxi</th>
<th>xxi</th>
<th>xxvi</th>
<th>xxi</th>
<th>xxvii</th>
<th>xxi</th>
<th>xxviii</th>
<th>xxi</th>
<th>xxviii</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplitude control</td>
<td>y</td>
<td>p</td>
<td>d</td>
<td>y</td>
<td>y</td>
<td>y</td>
<td>y</td>
<td>y</td>
<td>y</td>
<td>n</td>
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<td>n</td>
<td>n</td>
<td>n</td>
<td>n</td>
<td>n</td>
<td></td>
</tr>
</tbody>
</table>

**C3-16(2) Combination of concepts.**

The key to C3-16(2) is as follows:
- "y" - yes, combination of these ideas is possible
- "n" - no, these ideas cannot be combined
- "p" - perhaps, these ideas could be combined if the amplitude control offered by the prime mover is insufficient for requirements.

Diagram C3-17(3) shows the same chart with certain combinations excluded on the basis of design considerations discussed on the previous pages.
C3-17(3) Concepts selected for further development

Because of the previous work of the author on double crank mechanisms, it was considered that this type of mechanism was less likely to be able to provide the necessary inbuilt amplitude control than iii, and it was decided that work on this device could be "held back" until later if necessary. Diagram C3-17(4) shows how the major concepts still being considered compare with the specifications already discussed in section 1.

C3-17(4) Final concept evaluation

At this stage of the design process, the author decided to concentrate research effort
on concepts iii, xxiv, and xxiv. Concepts ii, xxvi, and xxix were subsequently made the
subject of investigation by Dr. T.G. King (a colleague of the author in the Mechanical
Engineering Department at Loughborough University of Technology) and Mr. Sen Yang (a research student from the Peoples Republic of China who was supervised
jointly by Dr. King and the author).

The comparison made between concepts in diagram C3-17(4) is bound to be crude
because of the lack of detailed information at the time that the work was carried out.
This is reflected in a compacted "rating scale" of from 1 (fits the specification very well)
to 3 (fits the specification less well). Concepts with the lowest values of summed ratings
are therefore considered to be the most promising.

Concept iii was rated more highly than i because it combined a means of providing
prime motion and a means of amplitude control. In fact, i could be combined with xxii
for a possible solution to the amplitude problem, and although this was initially rejected
as probably inferior, it was found later that the combination gave interesting results.

3.4 Conclusions on this chapter

The various sections in this chapter define a specification for the compensator,
introduce concept solutions, and select some of these concepts for subsequent
consideration.

The first idea is for a positive compensator based upon a 7-bar mechanism comprising
the basic form of concept iii, with the amplitude control mechanism of concept xxix.
The analysis and design of this idea is the subject of a major part of this thesis.

The second idea is for a passive compensator utilising a low mass disc moving in an
airflow. Analysis, design, and experimentation based upon this idea also forms a major
part of this thesis.
Chapter 4

COMPENSATION USING A NOVEL PNEUMATIC DEVICE
Chapter 4
Compensation using a novel pneumatic device.

The idea of using a stream of air to act as a compensator by collecting and releasing a loop of yarn, has several attractive advantages. For example; the inertia, in comparison to that of the spring/mass device currently used is much reduced, and secondly, there are likely to be few moving mechanical parts. However, all is not quite as easy as it might seem. Platt Saco Lowell had previously undertaken some research into a pneumatic device which could not be made to operate in a satisfactory manner. This work has never been published, but was based upon a simple suction tube which pulled in a loop of yarn as a result of a strong air flow.

![Diagram of a pneumatic compensator](image.png)

**C4-1(1) Compensator based upon a parallel sided suction tube**

This "compensator" was placed in roughly the same position as the existing spring/mass device.

There were two major problems with this device which discouraged further development.

1. If the force applied to the yarn was less than the yarn tension induced by the yarn draft, the yarn did not form a loop in the suction tube. In other words the device had no effect on the yarn whatsoever. On the other hand, if the applied tension was greater than that induced by
yarn draft, an unstable loop was formed in the suction tube. ie the loop grew continuously.

In theory, because the yarn draft varies throughout a cycle of the yarn guide, with a larger tension at the large diameter end of the package than at the small end, it might be argued that the air flow should be set to match the mean of these two extremes. This might be expected to produce a loop which would grow and shrink between two limits. However, it was found to be impossible to devise a system of the type described with a stable yarn loop.

2. Because of the small projected area of the yarn loop, an unacceptably high air flow rate was required to apply sufficient force to the yarn.

When work commenced at Loughborough University of Technology on the compensation problem, the author was unaware of the previous work carried out at Platt Saco Lowell on pneumatic compensation. Initially, some experiments were carried out with parallel sided suction tubes of the type investigated by Platt Saco Lowell. Two of the tubes used in the LUT experiments are shown in the next photograph. The tubes were of circular cross section, one with a bore diameter of 7 mm, the other with a bore diameter of 2 mm. Using the suction services available on the 4-position spin tester, the maximum achievable flow rate through the tube was 25 litres/min, and this failed to provide a sufficiently large force on the yarn to create a loop with the nominal yarn tension set at 30 grams. With the smaller diameter tube fitted to the machine, the maximum flow rate achieved was 20 litres / min, and this arrangement was successful in drawing yarn into the tube when the yarn tension was set as before. However, a stable loop could not be achieved.
C4-3(2) Parallel sided pneumatic compensation tubes.

Discussions with PSL after the initial pneumatic compensator work, revealed the previous work undertaken by the Company, and supported the findings of this earlier research.

4.1 Principles and patents

Having become interested in the possibilities of pneumatic compensation, the author undertook literature and patent surveys. In effect, very little was revealed by these surveys. The literature survey, which covered publications in journals and at Conferences, provided no relevant information at all. The patent survey revealed only two inventions with direct relevance. The first was a patent (4) by Phillips Electronic and Associated Industries entered in May 1976. The following diagram outlines the principle.
C4-4(3) **Pneumatic wire tensioning device patented by** Phillips Electrical Industries.

The device is designed to induce tension in a continuously moving strand of wire as it is wound onto an asymetric core. The wire passes around a roller before entering between two plates (in the plane of the paper in the above diagram). The gap between the plates is only slightly larger than the diameter of the wire, and therefore provide a "seal" between themselves and the wire. Air (or "gas" as the patent specifies) is continuously drawn from the cavity between the plates. This produces a pressure drop from one side of the wire to the other and, pulling the wire into a loop, forces it against the side of the cavity. This contact force between the wire and the cavity wall causes frictional drag on the wire, and provides the tension required.

It was felt that this design was not suitable for adaptation to yarn tensioning. Firstly, the yarn has a much smaller diameter than the wire, and much closer fits would have to be maintained to produce the forces required without excessive air flow. Secondly, friction of the type described could adversely affect the quality of the yarn surface.

The second patent (1) related to an invention by Messrs Taylor and Young which was registered in August 1985, and can again be used for the tensioning of wire. Outline details are given in the following diagram.
Wire, which is being wound upon a bobbin, is made to pass along a tube which is only slightly larger in internal diameter than the wire. Air is sucked through the tube in the opposite direction to the direction of travel of the wire. This causes a drag on the wire because of both air friction, and the friction developed by the wire being pressed against the side of the tube.

Again, this device was not considered suitable for developing into a yarn compensator. The high velocity of air passing down the tube would be likely to "texture" the surface of the yarn. In addition, there would be a large difference in tension between the yarn on the upstream side of the device, and that on the downstream side. The upstream yarn, under certain conditions, would be likely to lose all tension, in which case it would sag or "flap". This would almost certainly require the design and fitting of an upstream accumulator to control this slack yarn.

To overcome the problems described so far, an entirely novel pneumatic device was devised by the author of this thesis. The device (diagrams C4-6(5) and C4-8(6)) consists, essentially, of an air suction tube of rectangular cross section, where the height "$H_p$" is several times larger than the width "$l_p$". The inside of the tube is shaped so that $H_p$ increases with distance from the throat, where the yarn enters. A further improvement on previous designs is achieved by the insertion of a disc into the tube. The yarn runs around a groove in the disc, and is tensioned by the force imparted to the disc by the pressure drop in the tube.
The yarn enters the tube, and passes around the disc, which is "floating" in the air flow caused by the suction of air through the port at the back of the tube. The disc which is marginally smaller in width than the sides of the tube, is free to rotate as the yarn passes around it.

A major development found in this device over the method examined previously by Platt Saco Lowell is the effect upon performance of the tapered inner walls of the tube. As the yarn tension varies as a result of cone angle, the disc (and hence the yarn loop) moves backwards and forwards along the tube. When the tension is high, the disc is pulled towards the throat of the tube. When the tension reduces, the disc moves down the tube away from the throat. There is however, much less tendency for an unstable loop to form than in the straight tube design. This is because, as the disc moves down the tube under decreasing tension conditions, the gap between itself and the tube wall increases, thereby reducing pressure drop and decreasing the force which is applied to it by the air flow. Conversely, as tension increases, and the disc is pulled towards the throat, the gap decreases, thereby increasing both the pressure drop and the force on the disc. The pneumatic compensator is, in effect, a pneumatic spring with a rate which can be defined by the profile of the tube walls. In addition to this, the mass of the device is little more than that of the disc.
4.2 Characteristics of pneumatic compensation.

This section describes the analysis which was undertaken, to define the characteristics of the novel pneumatic compensation method just described. The air flow down the tube, and around the rotating and translating disc is extremely complex. A visual inspection of the flow using smoke traces and high speed video revealed an unsteady turbulent regime on the downstream side of the disc. In addition to the motion imparted by the yarn, there was a tendency for the disc to move up and down in the tube. It was recognised that some of this effect was certainly due to out of balance of the disc. However, it was considered likely that some perturbations were caused by vortex shedding off the edge of the disc.

During the early phase of concept development, the need was for a set of design equations which could be used to investigate the basic relationship between the design parameters. It was considered that a rigorous, lengthy, and painstaking analysis of the complex airflow was likely to be unproductive for this purpose. For this reason, the analysis was limited to a development of the equations describing the relationships between air flow rate, tension, disc diameter, and tube dimensions, within a steady flow regime. These equations are developed below.

The orifice equation (30) applied to a rotometer was considered to be a reasonable starting position for the analysis of the device. This equation, which relates air flow to orifice size, pressure drop and discharge coefficient is:

\[
Q_p = C_p A_p \left( \frac{2 \Delta P_p}{\rho_p \left( \frac{A_p^2}{a_p^2} - 1 \right)} \right)^{0.5}
\]

where, for a nozzle with a movable float:

- \(Q_p\) = volume flow rate of air through the nozzle
- \(C_p\) = nozzle discharge coefficient
- \(A_p\) = cross sectional area of the nozzle
- \(\Delta P_p\) = pressure drop across the obstruction (floating disc)
\(\rho_p\) = density of air

\(a_p\) = area of the orifice (ie \(A_p\) - cross sectional area of the obstruction)

The following diagram shows two orthographic views of the nozzle, and defines the geometric parameters involved. The nomenclature for the nozzle geometry is as follows:

\(\alpha_p\) = the half angle of the nozzle walls

\(d_p\) = the diameter of the float

\(T\) = yarn tension

\(x_p\) = the distance of the float centre from the nozzle throat

\(E_p\) = the height of the nozzle at the throat

\(H_p\) = the height of the nozzle at the position of the float

\(t_p\) = the width of the nozzle

From the nozzle and float geometry shown above:

\[ A_p = H_p \ t_p \]  \hspace{1cm} (2.C4)
but,

\[ H_p = E_p + 2 x_p \tan(\alpha_p) \] ................................. (3.C4)

therefore, combining 2.C4 and 3.C4 gives:

\[ A_p = \left[ E_p + 2 x_p \tan(\alpha_p) \right] t_p \] ................................. (4.C4)

The force \( (F_p) \) on the float due to the pressure drop \( (\Delta P_p) \) across it is:

\[ F_p = \Delta P_p d_p t_p \] ................................. (5.C4)

Equation 5.C4 assumes that the thickness of the float = \( t_p \). ie. there is negligible gap between the float and the walls of the nozzle.

But yarn tension \( (T) \) is given by:

\[ T = \frac{F_p}{2} \] ................................. (6.C4)

Therefore, from 5.C4 and 6.C4:

\[ T = \frac{\Delta P_p d_p t_p}{2} \] ................................. (7.C4)

Substituting 4.C4 and 7.C4 into 1.C4 yields:

\[ Q_p = C_p \left( E_p + 2 x_p \tan(\alpha_p) \right) \left\{ \frac{4 T}{d_p t_p^0} \left[ \frac{(E_p + 2 x_p \tan(\alpha_p))^2 t_p^2}{(E_p + 2 x_p \tan(\alpha_p) - d_p)^2 t_p^2 - 1} \right] \right\}^{0.5} \] ................................. (8.C4)

Equation 8.C4 is a useful design equation for obtaining a better understanding of the influence of system parameters on performance. Care however must be taken, since the equation contains the related parameters \( C_p \) and \( x_p \), and \( C_p \) will certainly vary with the distance of the float from the nozzle throat. In fact, no experimental work was
available at the time that this equation was developed to indicate a suitable value for $C_p$ at any disc position. However, as a design equation, the purpose of 8.C4 was to assist the understanding of the sensitivity of the design to parameter value changes.

As a first attempt at designing a nozzle, it was decided to plot sensitivity graphs for a nozzle with the following arbitrary parameter values:

$$
\begin{align*}
C_p &= 1 \\
E_p &= 25 \text{ mm} \\
T_p &= 2 \text{ mm} \\
T &= 500 \text{ N} \\
\rho &= 1 \\
x_p &= 0
\end{align*}
$$

Inserting these values into 8.C4, and calculating $Q_p$ for various values of $d_p$ provides data from which the following graph is plotted.

![Graph](image)

**C4-10(7) Flow rate v disc diameter from equation 8.C4**

Obviously, as the float diameter increases to fill the nozzle throat, air flow rate is reduced, until it reaches zero at $d_p = 25$ mm.

Another interesting graph is that obtained by plotting flow rate in terms of orifice size.
expressed as $E_p/d_p$, and this is shown below.

![Graph showing flow rate vs. throat/disc ratio](image)

**C4-11(8) Flow rate vs. throat/disc ratio from equation 8.C4**

It was recognised that keeping the volume flow rate of air to a minimum would be a critical factor in achieving an acceptable device. This is due to the power and cost implications of large air flows, associated noise levels, and the possibility of high air velocities damaging the surface of the yarn. The previous graph shows the importance of keeping the $E_p/d_p$ ratio as small as possible to minimise air flow requirements.

A number of more detailed calculations were carried out in an attempt to gain a wider "feel" for the characteristics of the proposed device. With $\alpha_p$ set at 2.5 degrees, $Q_p$ was calculated for various values of float position and yarn tension. A set of compatible units for equation 8.C4 are shown below:

$$Q_p = C_p \left( E_p + 2x_p \tan(\alpha_p) \right) \left\{ \frac{4T}{d_p \rho_p \left[ \frac{(E_p + 2x_p \tan(\alpha_p))^2 t_p^2}{(E_p + 2x_p \tan(\alpha_p) - d_p)^2 t_p^2 - 1} \right]^{0.5} \right\}$$  

...(8.C4)

$Q_p$ (l/min); $E_p$ (m); $x_p$ (m); $d_p$ (m); $t_p$ (m); $\rho_p$ (kg/m$^3$); $\alpha_p$ (degrees); $C_p$ (dimensionless)
If the float is to be formed from a circular disc, then the periphery of the disc requires some form of location for the yarn. This is to ensure that the yarn does not slip off the disc periphery during operation of the device. It was eventually decided that the disc groove in the prototype pneumatic compensator should be of the easily machined form shown on the diagram below.

![Diagram showing the disc groove](image)

**C4-12(9) Geometry of the disc groove**

For a disc with a thickness nominally equal to the width of the nozzle ($t_p$) of (say) 2 mm, the total projected groove cross-sectional area (top and bottom) would be 2 mm². Therefore, if the outside diameter of the disc was chose to be 25 mm, then in this particular case, the projected area of the disc into the air flow would be given by:

\[
\text{Projected area of disc} = d_p \times t_p - 2 \text{ mm}^2
\]

\[= 48 \text{ mm}^2 = 4.8 \times 10^{-5} \text{ m}^2\]

From the necessity to keep air volume flow rate to a minimum, the height of the nozzle at the throat ($E_p$) will need to be kept as small as possible. In practice, if the disc is to be introduced to the nozzle via the throat, then the throat height cannot be smaller than the disc diameter. For this reason, investigatory calculations fixed $E_p$ at the same value
as $d_p$.

Using these initial trial values of the device parameters, the following is a sample calculation of $Q_p$ based on equation 8.C4.

For $x_p=20$ mm, $t_p=2$ mm, $\alpha_p=2.5$ degrees, $T=40$ g (0.392 N), $d_p=25$ mm, $\rho_p=1.225$ kg/m$^3$.

\[
Q_p = (4.97 \times 10^{-5}) \left[ \frac{32666.7}{1.225 \left( \frac{2.475 \times 10^{-9}}{3.049 \times 10^{-12}} \right)} \right]^{0.5}
\]

\[= 17.11 \text{ l/min}\]

$Q_p$ was calculated for a range of values of $x_p$ and $T$, and the results are plotted in the following four graphs.

C4-13(10) **Air flow rate against float displacement from throat**
for a projected float area of $4.8 \times 10^{-5}$ mm
C4-14(11) Air flow rate against float displacement from throat
for a projected float area of $9.6 \times 10^{-5}$ mm

C4-14(12) Air flow rate against float displacement from throat
for a projected float area of $1.44 \times 10^{-4}$ mm
These graphs clearly indicate a number of important features about the device. Firstly, for a particular constant value of yarn tension, the air flow increases linearly as the float moves away from the nozzle throat. Secondly, for given values of tension and float displacement, the required air flow rate decreases with increasing float diameter. If specific specific values of $Q_p$ and $x_p$ are extracted from these graphs then the relationship between $Q_p$ and $d_p$ can be found for specific values of $x_p$. The next graph shows this relationship when $x_p=30$ mm, and the nozzle half angle ($\alpha_p$)=$2.5$ degrees.
There is however, a "trade off" between reducing air flow requirements by increasing the float size, and the disadvantages which arise from the corresponding increase in mass. This trade off is difficult to predict or analyse, since it seems possible that a "light weight" disc may be produced by manufacturing it in the form of an annulus. This however would be bound to affect the flow equation, if only by changing the value of the discharge coefficient. In fact, the value of the discharge coefficient is the parameter which prevents extensive theoretical predictions being made about the pneumatic compensator's performance characteristics. For all of the previous calculations, \( C_p \) has been set at an arbitrary value of unity. Not only will this not be correct, but the true value is likely to be a function of \( Q_p \), and \( x_p \), and will therefore change as the float moves along the nozzle. With these observations in mind, it was decided not to pursue the analysis further, but to construct a prototype device which could be used to test certain aspects of the mathematical model.

Finally for this section a graph of theoretical air flow rate against tension for different values of \( x_p \) is presented. This graph will be used in section 4.4.1 to evaluate the value of the discharge coefficient against experimental data.
4.3 Prototype design.

On the basis of trial and error, the first prototype pneumatic compensator was constructed with a disc 25 mm in diameter, and 2 mm thick. The internal wall angle of the orifice was made variable by means of pivoting plastic inserts which could be set at any included angle between zero and 10 degrees, by rotating them about the front fixing screws. The prototype, which is shown in the following photographs had a BSP pipe fitting at the back of a side wall. This provided a connection via nylon tubing to a venturi suction head fitted to the 80 psi compressed air supply.
C4-18(17) **Prototype compensator disassembled**

As can be seen, the construction was a sandwich, with the side arms being clamped between two transparent plastic side walls. The thickness of the side arm was the same as the thickness of the disc, but disc side clearance was achieved by the fitting of paper gaskets between the side arms and side walls. Also, because of the abrasive quality of the yarn, a metal shroud with ceramic inserts was fitted over the throat end of the device, to guide the yarn into the nozzle. The general arrangement is shown in the following diagram.
Because of the importance of minimising the throat size/disc diameter ratio to keep air consumption small, the throat height was made to be such that the disc/throat clearance was less than 0.2 mm. However, the pressure drop across the disc would certainly not be as small as that indicated on the graph because of the necessity of machining a peripheral groove to provide a register for the yarn.

The device was mounted onto a bracket which was secured to the front of the machine by screws picking up the holes normally used to secure the bollard compensator. The
bollard compensator was removed for the following tests.

4.4 Experiments on the first prototype.

The experimental facility for testing the prototype compensator is represented diagrammatically below.

**C4-20(19) Test facility for prototype pneumatic compensator**

The tensiometer shown in the above diagram was a Rothschild capacitative probe which was connected to a calibrated bridge circuit. A Bruel and Kerr chart recorder was connected to the bridge. Also included, but not shown on the diagram was a 'U' tube mercury manometer, which was connected in the air pipeline close to its exit from the nozzle. A GEC Elliot 1100 Rotameter (max flow rate: 40 l/min) was used to measure air flow rate. To facilitate the measurement of disc movement, a measuring scale graduated in cm and mm was secured to the side wall of the device. To provide suction, the laboratory 80 psi compressed air supply was connected to a venturi device in the suction line.

The tensiometer was calibrated by clamping it to a table and threading a piece of filament thread through the the measuring gate. The upper end of the thread was held in a laboratory retort clamp, and a number of weights were secured to the lower end on a weight hanger.
4.4.1 Static tests.

Before experiments were carried out with the pneumatic nozzle on the Masterspinner, a number of 'static' tests were carried out to establish the characteristics of the device, and enable the experimental results to be compared to the theoretical predictions of equation 8.C4. The nozzle was clamped to a bench top, then, with the experimental arrangement as shown in the previous diagram and the suction applied, a loop of yarn was introduced into the throat and the disc inserted. The yarn was clamped at both ends, and the tensiometer inserted in the yarn path below the nozzle. With the venturi set to provide maximum suction, the results shown in the following graph were obtained:

![Graph showing tension and air flow rate against the distance of the float from the throat. From experimental data.](image)

C4-21(20) Tension and air flow rate against the distance of the float from the throat. From experimental data.

The graph shows a predictable relationship between the parameters. As the float moves down the diverging nozzle, the gap between itself and the nozzle walls increases. This results in a reduction in force on the float due to the reduction in the pressure drop. Similarly, because the float creates less of an obstruction as it moves down the nozzle, the air flow increases.

The following graph shows the relationship between suction pressure and float displacement.
C4-22(21) Suction pressure against the distance of the float from the throat. From experimental data.

Again, the general relationship shown in the preceding graph is to be expected, since suction pressure is responding to the corresponding decrease in pressure drop as the float moves down the nozzle.

The next graph shows the comparison between experimental and theoretical predictions for changes in yarn tension for various float positions. The lower (and shallower) of the four curves is the experimental tension variation replotted from the graph on page C4-21. The remaining three curves are drawn on theoretical data from equation 8.4 each for a different value of the discharge coefficient $C_p$. As can be seen, none of the theoretical curves fit the experimental data closely.
C4-23(22) Yarn tension against displacement of the float from the throat. Comparison of experimental and theoretical data.

As pointed out in the previous section, because the configuration of the nozzle changes with float position, it is inconceivable that the discharge coefficient would remain constant over a range of values of $x_p$. The next graph shows the variation in $C_p$ necessary to make the theoretical results correspond with the experimental results. As would be expected, the coefficient is low when $x_p$ is small, since this corresponds to a situation where the pressure drop is large, the air velocities are high, and a more unsteady flow regime can be expected. As $x_p$ increases, air velocity and pressure drop decrease, and a more steady regime might be expected to be present. Corresponding to these changes, the discharge coefficient increases.
4.4.2 Dynamic tests.

The main objectives of the dynamic tests on the pneumatic compensator were:

1. to investigate if good quality packages could be wound at high speed
2. to investigate if the device caused damage to the yarn
3. to investigate if the yarn caused excessive wear to the float
4. to observe any relevant operating characteristics of the device.

Objectives (1) to (3) in the above list could be achieved relatively easily by simply winding packages on the Masterspinner with the pneumatic compensator fitted, and the results of these investigations are described in section 4.4.2.1. However, because of the speed of operation, it was necessary to employ high speed video equipment to achieve objective (4) by observing the device in motion. The results of this part of the investigation are described in section 4.4.2.2.

4.4.2.1 Dynamic testing of the pneumatic compensator.

With the prototype pneumatic compensator fitted to the Masterspinner, a number of conical packages were wound at yarn delivery speed speeds varying from 100 m/min to 480 m/min. Since there was no quantitative measure of package quality available, the subjective measures described in the table on page C1-12 were used as a basis for comparison between pneumatic and bollard compensator wound packages. Two
types of packages were used as a standard of good quality for the test packages. These "standard" packages were those obtained from the Platt Saco Lowell Masterspinner at a yarn delivery speed of 150 m/min, and a Schlafhorst Autocoro machine at 120 m/min. Both "standard" packages were built using 30 Ne cotton yarn as was also the case for the package wound using pneumatic compensation.

At no stage during the construction of conical packages using pneumatic compensation, was the resulting package distinguishable from the standard PSL package using the subjective criteria listed on page C1-12. This was true for all sizes of package from "starter" to full size, and over a range of yarn delivery speeds up to 500 m/min. The Masterspinner was, of course, incapable of spinning yarn of useful strength much above 350 m/min, so for high speed testing, packages were constructed by winding from a completed cone.

One of the most gratifying observations in using the prototype pneumatic compensator was that the disc appeared to rotate with the same peripheral speed as the yarn delivery rate. This conclusion was drawn from an examination of high speed video sequences, and the observation that there was no evidence that a groove was being produced in the periphery of the disc through the abrasive contact of sliding yarn. This ability of the disc to remain in contact with the yarn is, of course, another indication of the advantage of an ultra low mass device. Since the previously unknown effect of the disc on the yarn was one of most probable sources of yarn damage, it was subsequently not surprising that the pneumatic compensator made no observable changes to the yarn appearance, either with the naked eye or at low level magnification. Tests of yarn strength for pneumatic and bollard compensated yarn also failed to find any significant variations between the two methods.

Although the packages constructed at 500 m/min with the assistance of the pneumatic compensator were clearly of a mechandisable quality, several problems were identified with the prototype:

1. Because the device worked from the existing machine suction line, rather than from a compressed air supply used to "blow" the disc down the tube, there was a tendency for fly (small fibres) to be sucked into the throat. After a while, these tended to cause problems by accumulating on the side walls where they provided resistance to the free motion of the
disc, thereby causing it to stick from time to time. It was considered likely that this particular problem would be reduced if the entire compensator was constructed from a material with a reduced propensity to develop static charge on its surface. However, it was also recognised that this might merely move the fly collection problem elsewhere, such as on the disc or in the air outlet duct.

There were also the rare occasions recorded where a larger object (such as a fly of the winged variety) was sucked into the nozzle, and blocked the movement of the disc so that the yarn broke.

2. The most serious problems were associated with the "threading" of the nozzle. After piecing the end, the yarn was passed over the nozzle throat where the suction caused a shallow loop to be pulled 1mm or 2mm into the nozzle. The disc was then carefully introduced by hand. On occasions, even though the gap between the disc and the side walls of the nozzle were small, the disc missed the yarn and was sucked up against the outlet duct at the far end. In such cases, the air supply had to be manually cut off, the disc retrieved and re-introduced. A similar process was necessary whenever the yarn broke.

3. On occasions where a yarn irregularity passed through the compensator (eg a slub), the disturbance to tension sometimes caused the disc to lurch towards the throat. If the disturbance was sufficiently large, the disc could be completely ejected from the nozzle.

4.4.2.2 Operating characteristics.

The topics covered in this section are the high speed behaviour of the disc in the nozzle, and the tension control achieved by the device.

High speed disc behaviour was studied by means of a high speed (200 frames / second) video camera loaned from the SERC. The still sequences shown in the following photographs do little justice to the understanding that developed from watching the video, but illustrate several important points.
Depending upon the volume air flow, the disc could be made to operate in any regime of the nozzle for a given yarn tension. Photograph C4-27(24) shows the compensator operating with a yarn delivery speed of just under 500 m/s and a mean tension of 50g. The disc is operating in a regime near to the throat of the nozzle.

C4-27(24) Pneumatic compensation 500 m/s, 50 g tension

C4-27(25) Same as C4-27(24) after 0.03 second elapsed time
C4-27(24) shows the disc at a position just before it has stopped moving to the right. In this position, the disc is undergoing maximum deceleration, and there is the greatest tendency for the disc and yarn to lose contact, but the photograph shows no indication of loose loops forming. This observation was made throughout the video sequences. Photograph C4-27(25) shows the disc 0.03 seconds after photograph C4-27(24). In this position, the disc is near to maximum velocity and travelling to the left. Again, the yarn is seen to be held taught by the disc.

Photograph C4-29(26) shows the disc operating under the same conditions as in the previous two photographs except that the air flow has been increased, thereby causing the disc to operate in a regime further from the throat. Under this and similar conditions, the float was seen to "bounce" between the tapered walls of the nozzle. C4-29(26) shows the disc against the lower wall, whilst C4-29(27) illustrates how, 0.075 seconds later, the disc is against the top wall. It was observed that the frequency of "bouncing" increased with the spinning speed, and it was thought that the movement could be due to out of balance. The two photographs show however that this is not true, since the disc is in the same rotational position in both cases. It is thought that the movement may be due to either the "Magnus" effect whereby the spinning disc has a downwards force imparted to it by the air flow, or by vortex shedding.

It was thought possible that vortex shedding might be observed by introducing a smoke trace into the throat. This trace can be seen in C4-29(26) and C4-29(27), both of which are computer enhanced scans from still photographs of high speed video sequences. Although the original photographs showed little visual suggestion that vortex shedding exists, the enhanced scans indicate that vortices may be present at the side of the nozzle with which the disc is in contact.
C4-29(26) Disc against lower wall with smoke trace

C4-29(27) Disc against upper wall with smoke trace
The ability of the pneumatic compensator to limit variations in yarn tension is an important factor in assessing its potential as a solution. The winding tests have already indicated that the device exerts "adequate" control, since the finished packages compare favourably with those produced by traditional methods. However, to ascertain the precise nature of the variations, a yarn tensiometer was fitted throughout the tests. The result of these tests was encouraging for all sizes of packages at speeds up to, and including 500 m/min of yarn delivery. The diagram shown below (C4-30(28)) is a scan of the experimental trace with a minimum size package at 500 m/min (ie the worst possible condition in terms of acceleration of the yarn loop. The mean value of tension is 40 grams with a maximum of 47 gram, and a minimum of 32 grams. This compares favourably with the variation in tension recorded with the spring bollard compensator, and the steadiness of the trace supports the observations made on photographs C4-27(24), C4-27(25), C4-29(26) and C4-29(27) that the disc keeps in continuous contact with the yarn even at high speeds.
C4-30(28) clearly shows the effect of the anti-patterning device in the low frequency tension change cycle which occurs approximately every 40 cycles of the yarn guide.

4.5 The second prototype

Following the experiments on the first prototype, a second was designed to overcome some of the problems observed with the first. This improved prototype was constructed with a closed nozzle as shown in C4-31(29).

![Diagrammatic representation of the second prototype]

In addition to the suction at the end of the nozzle, a secondary (balance) supply is now provided to give positive pressure on the other side of the disc. This secondary supply is to ensure that there is a small outflow of air from the yarn access slot, thereby preventing fly from being sucked into the device. A "parking zone" is also provided for the disc so that it no longer needs to be introduced by hand. On yarn breakage, the disc might be parked by either a reversal of the air supply, or by gravity if there was sufficient slope for the disc to rotate under its own mass once the supply had been turned off. Photographs C4-32(30) and C4-32(31) show the second prototype prior to assembly and during static testing respectively.
C4-32(30) **Second prototype before assembly**

C4-32(31) **Static tests on the second prototype**
4.5.1 Results with the second prototype.

The results obtained from the second prototype fully justified the time taken in its design and development. The use of a disc of twice the diameter of that used previously enabled the device to maintain a tension force 85% higher than with the smaller disc under the same air flow conditions. The second prototype also successfully overcame all of the problems associated with fly build up on the disc and nozzle side walls.

The automatic introduction of the disc on start up was found to be successful in slightly over 90% of the tests undertaken. In the remaining 10%, the disc missed the yarn and was drawn to the end of the nozzle. Although this might be improved by providing better guides for the yarn at the entrance to the nozzle, it was felt that this success rate was similar to that achieved by "patrolling robots" when undertaking automatic piecing of broken ends, and could therefore be considered acceptable.

At yarn delivery speeds close to 500 m/min the device displayed excellent control over the yarn loop, with none of the problems of slack yarn formation observed with the bollard compensator at much lower speeds. The following sequence of 27 still frames (C4-34(30) to C4-36(32)), was taken from a 200 frames per second video, show the position of the disc in the nozzle during 5 millisecond intervals. The sequence starts at photograph a1 and proceeds through a2, a3, b1, b2 etc. to i2, and i3. A series of marks are visible on the centre line of the perspex side wall of the nozzle. These marks are 1 cm apart, and may be used to establish disc displacement and velocity. The sequence from a1 to a3 record that part of the disc motion corresponding to it's maximum leftward acceleration. This is associated with the motion of the yarn guide as it moves away from the small end of the package, ie that part of the motion over which the bollard compensator has been shown to perform badly. In comparison, the second pneumatic compensator prototype exhibits no such loss of control, with a taught yarn being maintained at all times. The photographic sequence were recorded with the nozzle wall included angle set at 2°. The amplitude of motion of the disc can be measured by reference to the markings along the nozzle wall, and graph C4-37(33) shows how the device was mounted on the machine frame.
Second prototype undergoing tests on the machine

The final sequence of photographs (C4-38(34) to C4-40(36)) from the high speed video further illustrate the ability of the compensator to maintain a tight yarn loop at speeds close to 500 m/min. This sequence is a composite showing the pneumatic compensator and the yarn passing to the package over a curved distribution bar. The sequence suggests that good yarn control is retained at all stages of the cycle, even during the critical phase represented in photographs g1 to h3.
Sequence 1 to 9 of yarn motion at 45° wall angle.
4.6 Conclusions

The pneumatic compensator has good potential as a high speed compensator at speeds of (and probably in excess of) 500 m/min yarn delivery speed. The variations in yarn tension at 500 m/min compare favourably with the tension variations which occur when winding similar packages at 250 m/min. At mean tensions up to 50 gram, the suction available at the doffing tube was capable of operating the compensator. At mean tensions above this level, an auxilliary suction was required.

High speed video studies of the disc showed that it kept in contact with the yarn at high speeds, and showed none of the problems associated with the spring bollard compensator of losing contact at speeds above 300 m/min. There were no signs of wear between the yarn and the disc, which supported the video observation that the disc rotated with the same peripheral speed as the yarn velocity.
Chapter 5

THE BEHAVIOUR OF THE FREE YARN LOOP
Chapter 5

Behaviour of the free yarn loop

5.1 Outline
The analysis of Appendix 3 develops an equation (7.A3) which gives the velocity of the yarn through the guide as a function of machine and package parameters such as guide velocity, drive roller diameter and speed, package size and angle, and guide position. The difference in yarn velocity between that occurring at the point where it emerges from the delivery roller nip, and that occurring at the yarn guide is a major factor influencing yarn tension. An analysis of the effect of this velocity differential, and the yarn path length (free yarn length) between the delivery rollers and guide, is developed in appendix 2.

It will readily be seen that a means of maintaining yarn tension at a substantially constant value may be envisaged as being devised in the form of a variable size yarn loop. Since the delivery rollers rotate at a constant speed, then for a particular value of yarn velocity through the guide which is in excess of the delivery speed, the yarn will attain a steady state tension which may be calculated from equation 17.A2. Provided that the total yarn length is the same as that used in equation 17.A2, the formation of a loop of yarn by the introduction of three rollers into the yarn path as indicated in C5-2(1), will not change the steady state tension value.

If the velocity of the yarn through the guide is now changed by a modification to one or more of the parameters represented on the right hand side of equation 7.A3, there will be a resulting change in yarn tension. In reality, at a particular setting of the machine, the variable parameters are $y$ (the distance of the guide from the small end of the package), $\delta L / \delta t$ (the velocity of the guide) and, over a longer time period, the growth of package size. If initially package size is considered to be constant, and only changes to $y$ and $\delta L / \delta t$ are made (this is reasonable, since the periodic time of the cyclical variations in $y$ and $\delta L / \delta t$ is approximately 0.1 seconds at a yarn delivery speed of 500 m/min, whilst it takes approximately 4 hours to build up to a full size package at the same speed) then the resulting change in tension can be nullified (or compensated) by
a movement of roller 'A' in the following diagram, thereby adding or subtracting from the amount of yarn in the loop.

![Diagram](image)

C5-2(1) **Diagrammatic representation of the yarn loop caused by roller 'A'**

If the **new** values of y and $\delta L / \delta t$ were to be maintained constant for a protracted period of time, a tension control device of the type just described would not be able to maintain tension at the **previous** value since it would need to continue releasing or taking-in yarn, and would therefore reach the end of acceptable motion quite quickly.

However, the changes to y and $\delta L / \delta t$ are themselves cyclical with a short periodic time, and this demands that the motion of roller 'A' is also cyclical with the same periodic time.

By an appropriate movement of roller 'A' it should therefore be possible to maintain a constant value of yarn tension regardless of variations in the values of y and $\delta L / \delta t$. A knowledge of the variation in the length of the yarn loop is therefore essential if a compensator with a satisfactory motion is to be designed. On the basis of equation 17.A2, a computer program called "YARVEL" was written to calculate the nett change
in length of the length of yarn in the loop over successive small periods of time. The program also provides the velocity and acceleration of roller 'A'. A detailed description of the mathematical basis and structure of "YARVEL" and an associated data conditioning program called "DATAPREP", as well as the program coding are presented in Appendix 4. The output from the program is presented in the following section.

5.2 "YARVEL" output

The output from "YARVEL" predicts the displacement, velocity, and acceleration of a yarn loop which would achieve constant yarn tension when spinning onto conical packages. The range of tension drafts normally available on the Masterspinner is 0.996 to 1.05, with the lower numbers being more applicable to the production of cotton packages, and the higher numbers being used for man-made fibres and mixtures. Most of the investigations into yarn loop size using the "YARVEL" program had tension draft set at 1.00.

Since the primary aim of the investigation was to determine the requirements for a compensator which could cope with a yarn delivery speed of 500 m/min, this value was chosen for input to both "YARVEL" and "DATAPREP". In fact, the variation in loop size is independent of delivery speed, but loop velocity and acceleration are, of course, affected, and these measures are important in the final design. Other important parameters were set as follows:

- drive roller radius over the tyre = 38.2 mm
- ratio of drive roller to cam shaft speed (RROL) = 4/7
- package width across the surface = 150 mm

Entering these values, and package included angle = 9°, into "DATAPREP" and "YARVEL" produces the results shown on the following graphs. Graph C5-4(2) shows how the yarn loop must vary in size to maintain tension constant. The plot shows the results for a range of package small diameter sizes from 38 mm (which corresponds to the minimum package size) to one of 200 mm.
C5-4(2)  Yarn loop size variations for a range of package sizes.

Graph C5-4(2) displays a number of interesting features. The most important of these is that yarn loop amplitude will decrease as the package grows in size. This effect is clearly seen when observing the passive spring mass compensator currently fitted, and is accounted for in the design of positive compensators of the type fitted to Schlaflhorst machines. C5-4(2) shows loop size variations over one complete cycle of the guide cam, which corresponds to two complete passes of the package by the guide. The deliberate variation in guide motion between two adjacent passes is clearly seen in C5-4(2) by observing that the two positive peaks are at slightly differing values (as in fact are the two negative peaks). The small irregularities in the curves which occur in close proximity to the positive and negative peaks has been found to be due to the way that "YARVEL" calculates the numerical integration of the loop velocity data at these extreme points. In total, there are 769 coordinates describing the profile of the guide cam track. All of these were used by "YARVEL" in the calculation of a corresponding number of yarn loop size, velocity, and acceleration values. In C5-4(2), only 10% of these values are plotted as an aid to clarity, and other than for the errant extreme points the program produces a smooth continuous curve. Ignoring the extreme data points in the design study of a mechanical compensator in chapter 6 is therefore considered as being thoroughly acceptable.
Graph C5-4(2) provides an overall impression of loop size variation, but contains too much information to be clear. The following five graphs are separate plots of the loop size variations for each of the package sizes included in this analysis. Note that the axis scales are different on each of the following five graphs.

C5-5(3)  **Yarn loop size variation on 38 mm package.**

C5-5(4)  **Yarn loop size variation on 80 mm package.**
C5-6(5)  Yarn loop size variation on 120 mm package.

C5-6(6)  Yarn loop size variation on 160 mm package.
C5-7(7) **Yarn loop size variation on 200 mm package.**

The way in which loop amplitude varies according to package size is an important feature in the design of a positive compensator, since it describes the function of the control or feedback "mechanism" that will be required. The following graph (C5-7(8)) shows the nature of this variation. It should be noted that in the case of a compensator with the geometry shown in diagram C5-2(1) and roller 'A' moving in a straight line, the amplitude of motion of the roller will be half of that of the yarn loop. This relationship is also plotted on C5-7(8).

\[
y = 1338.3044 \times x^{-0.8709} \quad R = 1.00
\]

Equation of loop amplitude change

\[
y = 669.1522 \times x^{-0.8709} \quad R = 1.00
\]

Equation of compensator amplitude change

C5-7(8) **Variation in loop and compensator amplitude for varying package size.**
The important point revealed by C5-7(8) is that compensator roller amplitude decreases by a factor of approximately 4 as the package small diameter increases from 38 mm to 200 mm.

5.3 Comparison of "YARVEL" output
There were two methods available for comparing the output of "YARVEL" to experimental data. Firstly, the motion of the bollard compensator could be recorded and used to provide information about the variation in the loop length. The collection of bollard motion data using high speed video techniques has been described in chapter 2, and this data can be converted into loop length information using the equation 17.A9 developed in appendix 9. The second method was to carry out quasi-static tests, turning the machine over by hand, and measuring the change in length of a yarn loop which had been previously formed. Neither of these methods provide a precise comparison of results with the "YARVEL" program because of the following limitations:

Bollard motion recording
• equation 17.A9 does not precisely model the geometry of the yarn path length because it was defined for a case where there were two fixed points through which the yarn passed. One before and one after the compensator.

• the bollard compensator does not operate under constant yarn tension conditions as is the basic assumption for the positive compensator modelled in "YARVEL".

Quasi-static loop size measurement
• Yarn tension in the loop is zero and the effect of variations in mass flow rate will not be present

5.3.1 Bollard motion recording
Chapter 2 (section 2.3) explains that any passive compensator is incapable of maintaining constant yarn tension, and for this reason the amplitude of the yarn loop in such a device will not be the same as that calculated by "YARVEL" for a positive
compensator. In addition to this, at high speed, the passive compensator will exhibit
dynamic effects in its motion which will not occur in a positive device. However, it was
considered that the bollard compensator offered an available means of carrying out a
rough check on the calculated values of amplitude and amplitude variation with size,
especially if the tests were carried out at low speed.

The experimental procedure described in chapter 2 (section 2.3.2) using a graduated
compensator bezel and a high speed video camera was used to collect data relating
compensator rotation to yarn guide position, as typified by graphs in that section. This
was carried out with a package of 40mm small diameter. Equation 17.A9, developed in
appendix 9 for the length of yarn in the bollard compensator at various angles of
rotation was then used to calculate yarn loop length.

Table C2-13(12) showed the variation in bollard rotational position at different guide
positions. Using equation 17.A9 to convert these values to yarn path length gives the
values presented in the following table.

<table>
<thead>
<tr>
<th>Guide position</th>
<th>0</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
<th>15</th>
<th>16</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of yarn path (mm)</td>
<td>370</td>
<td>308</td>
<td>319</td>
<td>313</td>
<td>380</td>
<td>373</td>
<td>370</td>
<td>368</td>
<td>360</td>
<td>389</td>
<td>312</td>
<td>310</td>
<td>380</td>
<td>281</td>
<td>373</td>
<td>372</td>
<td>370</td>
</tr>
</tbody>
</table>

**C5-9(9) Table of yarn path length and guide position**

If the smallest of these values (ie 281) is subtracted from the values of yarn path length,
the resulting numbers may be used to represent loop length. Graph C5-10(10) uses
these figures to show how loop length varies with guide position.
This graph should be compared with C5-5(3) which shows the "YARVEL" predictions for this size of package. The differences are clearly observable. "YARVEL" predicts a loop length variation of slightly under 60mm whilst C5-10(10) shows a variation of around 80mm. This additional experimental variation may be explained by the stretching effect of the straight guide bar which is not included in the "YARVEL" analysis. The straight guide bar also introduces more violent movements to the compensator than would be present if a curved distribution bar were fitted.
5.3.2 Quasi-static loop size measurement

The Masterspinner was set up with a 170mm package and a straight distribution bar. A small loop was formed in the free yarn length, and the machine was turned over by rotating the delivery shaft by hand. For each 10mm movement of the yarn guide, the length of the yarn loop was measured. Tension draft was set at 1.00 during these tests.

The variation in loop size during this test is shown below in graph C5-11(10), together with plots for a 40mm diameter package with a straight distribution bar, and a 40mm diameter package with a curved distribution bar.

![Graph C5-11(10) Quasi-static yarn loop lengths at zero tension (experimental)](image)

C5-11(10) Quasi-static yarn loop lengths at zero tension (experimental)

The tension in the yarn at all stages of these tests was zero. Although loop length increased and decreased as would be expected, there was a general tendency for the mean size of the loop to decrease. This was because yarn draft was greater than unity, and average demand was therefore greater than supply. This effect would not be observed in the normal running of the machine because a rise in tension would result in a reduction in yarn diameter and a corresponding drop in mass flow rate onto the package, thereby stabilising the mean loop size.

Graph C5-11(10) can be corrected for this effect by multiplying each of the loop length
values by linearly increasing factors which will increase the last value to the value of the first. The result of this transformation is shown in the three graphs starting with C5-12(11).

**C5-12(11)**  
Quasi-static yarn loop lengths corrected for draft (experimental). Straight guide bar, 170mm dia package.

**C5-12(12)**  
Quasi-static yarn loop lengths corrected for draft (experimental). Straight guide bar, 40mm dia package.
The loop size variation for the 40mm and 170mm diameter packages (straight bar) are 57mm and 25mm, both of which correspond closely to the "YARVEL" predictions in graph C5-5(3) and C5-6(6) which give 50mm and 14mm respectively. The experimental loop size variation for the 40mm diameter package with the curved distribution bar as shown on graph C5-13(13) is 50mm. Comparison of the loop size variation for the two 40mm diameter packages shows that the loop size variation decreases by 7mm (12.3%) when the curved bar is fitted.

Output from "YARVEL" as typified in C5-5(3) shows irregularities in the curve at the peaks. This effect is almost certainly due to the way that the program deals with the numerical iteration at the reversal point. Smoothing the curve over the points leading to and from the peak gives peak values of + and - 23mm and a resulting loop size variation of 46mm which is very close to the experimental value of 50mm.
Chapter 6

A MECHANISM FOR POSITIVE COMPENSATION
Chapter 6
A mechanism for positive compensation

Chapter 5 has described the benefits of a system of positive compensation for yarn tension control. This chapter and appendix 6 describe the design and testing of a mechanical compensator to work at speeds in excess of those currently attained with positive devices. Section 3.2 described a number of mechanisms which might yield a satisfactory solution to the problem of tension control, but the one adopted here has 7 bars.

6.2 A variable amplitude mechanism

Chapter 5 showed that a mechanism output could be used to approximate a motion of the type required of the compensator arm, but could not provide a perfect match. In general, the analogy of a Fourier Series is valid, in that increasing the number of links allows a closer approximation to the ideal motion in much the same way that an increase in the number of Fourier terms allows a closer approximation to a particular function or set of data points. However, there is a trade off between the desirability of the output motion and the cost and reliability of the mechanism.

Several factors are of course involved. In terms of reliability, an increase in the number of links will bring a proportional increase in the number of pivots. Since the pivots are the source of mechanism backlash through bearing clearances, the total mechanism backlash will increase with the number of links. This will result in an increase in the percentage error present in a particular arrangement, with a probability of increased vibration due to lower rigidity. The total mechanism mass will also increase, with resulting penalties for the life of bearings, and deflection of links due to inertia forces. This effect is particularly important at high speeds. The manufacturing cost of the mechanism will also increase proportionally to the number of links and bearings.

Section 3.6 suggests that the simplest mechanism with the variable output amplitude necessary to allow for increasing package size had 4 moving links and one fixed (or ground) link. However, it was considered that a more useful approximation to the ideal motion, without excessive penalties of cost, unreliability, mass, or flexibility, might be achieved from a 7-bar mechanism. This decision was quite arbitrary in the sense that there was no available experience of such mechanisms being applied to similar problems, and no useful published information could be identified. However, one
important influence on the decision was the existence of a separate research project in the author's Department in which a Chinese postgraduate student (Mr Sen Yang) was attempting to carry out the mathematical optimisation of a 5-bar mechanism using a computer based hill climbing technique. The problems of defining a suitable objective function, and setting realistic constraints on fixed pivot coordinates, link lengths, transmission angles etc., was making the optimisation difficult. This made it unlikely that the method could be used for the optimisation of a 7-bar mechanism within an acceptable time scale for the project. However, it was considered that a manual iteration of alternative 7-bar mechanism configurations might yield a satisfactory (although not optimum) solution. It was therefore decided to leave the synthesis of the 5-bar mechanism to the mathematical optimisation approach, and concentrate on the iteration of the 7-bar mechanism solution in this project. It was also felt that the complexity of the 7-bar mechanism design problem was such that a purely mathematical approach was not only difficult but probably not even the best, and that engineering skill applied in a creative way might lead to a solution.

As a reminder of the 7-bar mechanism configuration defined in chapter 3, the following drawing is presented.

![7-bar mechanism](image)

**KEY**
- Pivot
- Link with 3 pivots
- Link with 2 pivots

**C6-2(1) The basic 7-bar mechanism**

Of the two triangular links, one represents the machine frame (or "ground"). The second, moving, triangular link is connected to ground via three routes. Two of the routes are comprised of two links, whilst the third has a single link. One of the routes will constitute the **input** drive, a second will be the **output**, and a third will be the **control** to adjust output amplitude with changing package size. The configuration
selected for the compensator is shown below.

![Diagram of 7-bar mechanism](image)

- Fixed pivot
- Moving pivot
- Point around which the yarn passes

C6-3(2) **The 7-bar mechanism adapted for the compensator.**

Pivots 1, 2, and 3 are the centres of rotation of the input crank, control crank, and output link. They are located on ground on the machine frame.

In the chosen configuration, the single link route between ground and the triangular link is that adopted by the input crank. The control crank and control link adopt one of the two-link routes, whilst the connecting link and output link adopt the other. The recognition that fixed pivot 3, and points e and f do not need to be placed upon a straight line is indicated by the representation of a triangular output link.

The choice of the selected configuration was based upon the following factors:

- a dual link route for the mechanism output stage gives simple control over output amplitude by allowing the position of pivot e to be changed in regard to 3 and f.

- the control crank moves slowly in comparison to the input crank. This is a strong argument for making the input stage a single link route, thereby
reducing the number of bearings rotating at high speed, and reducing the mass of the higher speed stage.

It was recognised of course, that a machine capable of spinning yarn at 500 m/min would represent a total redesign of the existing machine. Current space constraints on a compensator mechanism would be therefore not necessarily present on the redesigned machine. However, it seemed unrealistic to set about the design process without any restrictions upon space, and no better solution could be identified than using the current machine as a guide. An additional factor in coming to this decision was that the mechanism, once manufactured, would perhaps require testing, and the current Masterspinner would provide an ideal test bed for this purpose.

Initially, there was a need to develop a feel for the various ways in which a mechanism of the type described could be fitted onto the existing Masterspinner. Even before any analysis or optimisation was carried out, it was decided to "explore" the available space by fitting "promising" looking configurations upon a cross sectional drawing of the appropriate part of the Masterspinner.

The cross section, showing important existing machine elements such as the drive and delivery rollers and shafts, and the cam shaft and it's 'U' section housing, was drawn on the University's DOGS computer based 2d-draughting system. The section was defined as a symbol (called "MAP") so that it could be rapidly recalled from memory as a basis for trying various mechanism configurations. Arbitrary choices were then made for the positions of the fixed pivots and the lengths of the various links. Each link was represented in DOGS by a symbol with key points at the pivot points. These simple facilities then allowed an initial exploration of the various possibilities for assembling the mechanism within the the envelope of space provided by the existing machine elements. The following photograph (C6-5(3)) provides an example of some of the configurations considered during the investigation.
An attempt to place the mechanism in the available space.

The photograph shows an early attempt at placing the 7-bar mechanism within the available space. The input shaft (in yellow) is shown in the centre of the eccentric (magenta) which forms the input crank. Directly below, are the control crank and link (magenta and yellow respectively. In the configuration shown, the input link shares a common pivot point with the connecting link (dark blue) on the input member (light blue). The connecting link provides the final drive to the output link (green). Experience with this design method persuaded the author that a more promising solution would be to locate the input crank and link below the control crank, because of the relative ease of routing a mechanical linkage system from the package support arms to the control crank.

Although these investigations were useful in obtaining an improved understanding of the various mounting options within the available space, the use of a fixed symbolic representation of the mechanism components made it far too slow to be used as a method of determining if the output motion of a particular configuration was sufficiently close to the ideal. This was because output link motion could only be observed by
redrawing the mechanism with the input crank at various degrees of rotation, and manually assembling the remainder of the mechanism components on the screen.

What was required, was an "automatic" method of drawing a promising mechanism at various positions of input crank rotation. The DOGS parametric symbol capability offered an ideal way of achieving this aim.

DOGS parametrics permit the designer to write macro routines in a special parametric language which, when called, complete a series of predetermined calculations and, if required, produce appropriate drawings on the workstation display. To enable a macro to be written to draw a mechanism to the display, the equations of motion needed to be determined. These equations, which defined the positions of all the pivot points in the mechanism as functions of fixed pivot coordinates, link lengths, and control and input crank rotational positions, are derived in appendix 6.

The parametric macro (program) based upon these equations is called "MECHANISM", and is described in detail in appendix 7.

"MECHANISM" has been written so that the designer describes the 7-bar mechanism under consideration by responding to a series of screen prompts. These prompts request the following information:

• the x,y coordinates of the centre of rotation of the input crank
• the x,y coordinates of the centre of rotation of the control crank
• the x,y coordinates of the centre of rotation of the output link
• the lengths of:
  • the input crank
  • the control crank
  • the control link
  • the connecting link
• the lengths:
  • ab
  • ac
  • cb
  • 3e
  • 3f
  • ef
the angular setting of the control crank

The parametric then automatically produced overlaid drawings of the mechanism for each 20° of input crank rotation. To simplify these drawings, the mechanism components were represented in skeleton form (i.e., two pivot links were represented by a single straight line, and three pivot links by a triangle. The overlaid drawings were produced on top of the machine cross section drawing, thereby enabling an evaluation to be made of the extent to which the mechanism envelope of motion would infringe upon the existing machine components.

Although this visual representation of the mechanism envelope is important in evaluating possible configurations, it would be of no use unless the parametric provided details of the amount of yarn that the mechanism would accumulate throughout a full cycle. A second parametric (called GRAPH1) was therefore used to calculate the amount of yarn held in the compensator at any given time, and plot a graph of the variation during a cycle. The mathematical analysis and resulting parametric program are also described in detail in appendix 7.

To enable a comparison to be made between the amount of yarn taken in by the mechanism and the ideal amount required for precise tension control as described in chapter 5, a second parametric called "GRAPH2" was written. The yarn loop length data presented in chapter 5 was entered into a data file on the Apollo. GRAPH2 (which is described in appendix 7) was then used to draw a screen graph of the ideal variation of the yarn loop length against that achieved by a particular mechanism configuration.

The following photograph (C6-8(4)) shows the screen with the output from MECHANISM, GRAPH1, and GRAPH2 displayed. The cross section through the existing machine components (drawn from symbol MAP) is shown mainly in white, although a sheet metal (non-structural) bulkhead is shown in red chain dot. The colour coding for the mechanism components is as follows:

- input link - pale blue (cyan)
- control crank - green
- control link - dark blue
- connecting link - pink (magenta)
- output link - yellow
The 16 mechanism parameters identified on pages 6 and 7 of this section, are listed below the drawing of the mechanism.

C6-8(4) Output from MECHANISM, GRAPH1, GRAPH2 parametrics superimposed on MAP.

Underneath the printout of mechanism parameter values is the output from GRAPH1 (in yellow) which is the variation in the yarn loop length for the mechanism, and the output from GRAPH2 which is the ideal variation. In the above photograph, the two graphs have been synchronised so that they correspond at the bottom of the first part of the cycle.

The following photograph (C6-9(5)) shows the combined output from GRAPH1 and GRAPH2 for a mechanism which gives a much closer match between actual and ideal yarn loop length than was the case in the previous photograph. The photograph also
illustrates how the ERROR parametric can be used to plot the difference (or error) between the two graphs. The vertical blue ordinates on the actual/ideal graph are used to identify the points of interest where the difference is to be calculated and plotted on the bottom graph. The smaller the perturbations on the lower graph, the closer is the mechanism to providing the ideal output motion.

C6-9(5) Parametric output with the addition of the ERROR parametric showing the effect of positional error on yarn loop length.

6.3 Mechanism optimisation
The selection of a suitable mechanism by the method developed in appendices 6 and 7, and described in the previous section, is an iterative process. It cannot therefore be expected to yield the absolute optimum mechanism design. Although this admission suggests that the title of section 6.3 may be incorrect, the author argues that the design thereby achieved is sufficiently superior to anything that could have been synthesised by long hand techniques to warrant its use.

The process of mechanism design took between 3 and 4 weeks of continuous work. The main objectives being to:
1) ensure that the envelope of motion did not interfere with immovable machine parts
2) ensure that the fixed pivot points (1, 2, and 3) were situated in a position where they could be adequately located, and that drives could be secured to the input and control crank shafts
3) minimise the error between the actual and ideal yarn loop lengths over the full range of package size, but particularly at the smallest package size where the values of compensator output amplitude and acceleration are at a maximum.

The first two items on the preceding list are subjective and essentially impossible to quantify. For this reason, even though they played an important role in the design process, they are not detailed in the following presentation of results, which is limited to the variation and error in yarn loop length with different values of the mechanism parameters.

The first configuration of mechanism which was explored was the one represented diagrammatically in C6-10(6).

![Diagram of mechanism configuration](image)

**C6-10(6) Initial configuration for the optimisation study**

In this configuration, the yarn path from point 'a' to point 'b' would be a straight line if it was not diverted around the mechanism output roller. In other words, the fixed compensator rollers are positioned so that the points on their periphery which contact...
the yarn lie on the straight line drawn from 'a' to 'b'.

There are of course other options available for the positioning of the fixed rollers. Diagram C6-11(7) shows some of these.

C6-11(7) Some alternative compensator configurations

In C6-11(7)i and C6-11(7)ii, the fixed rollers are inside (closer to the longitudinal centre line through the machine) the line connecting points 'a' and 'b' in C6-10(6). There seemed to be no advantage in these configurations over that shown in C6-10(6).

Diagram C6-11(7)iv shows a case where the fixed rollers are moved outside the straight path.

Although it was observed at the start of the optimisation process that the various configurations proposed in C6-11(7) might be useful when attempting to fit a particular
mechanism into the available space, there seemed to be no reason to adopt any of these at the outset. The configuration of C6-10(6), which is identified from now on as the SYP (straight yarn path) configuration, was therefore chosen for investigation.

One of the major problems of starting any design optimisation process is the choice of the seed configuration, which is the first guess at a solution. The parameter values (in this case, fixed pivot coordinates and link lengths) of the seed configuration are entered into the computer program so that the first analysis step can be carried out. If the seed configuration is not a feasible solution, then the computation will fail, and a meaningless output will be delivered. It will not, therefore, be possible to continue to the second iteration.

In the case of the compensator mechanism, it was the difficulties in choosing the initial values for the link lengths and fixed pivot coordinates which presented considerable problems in the definition of a suitable seed. For the mechanism shown in C6-3(2), there are two coordinates for each of the three fixed pivots, plus ten parameters to fully describe all of the links. The choice of initial values for these sixteen parameters turned out to be a considerable problem, especially since the resulting mechanism had to fit within the area defined by the MAP symbol. Numerous attempts at defining a seed were made, but none of the parameter value groups which were suitable from an overall size point of view, resulted in working mechanisms, and all resulted in program crash. Recourse was eventually made to physical modelling, using the standard designer's modelling materials (cardboard and drawing pins). The result was the rapid definition of a workable seed mechanism to be used as the first stage of the optimisation process.

The optimisation process made possible by the four parametric programs produced a number of interesting possibilities. However, the SYP configuration was characterised by a difficulty in obtaining sufficient variation in yarn loop amplitude corresponding to package size and control crank angle. One of the best SYP configurations had the parameter values shown in the following table (C6-13(8)).
Parameter values are in mm.

### Input crank pivot coordinates: X=170, Y=50
### Control crank pivot coordinates: X=111, Y=109
### Output crank pivot coordinates: X=40, Y=0
### Input crank length: 5
### Control crank length: 23
### Control link length: 35

![Parameter values for a promising SYP mechanism](image)

**C6-13(8) Parameter values for a promising SYP mechanism**

The screen plots for this mechanism are shown in photographs C6-14(9), C6-14(10), and C6-15(11) for a control crank angle of zero degrees.
C6-14(9) General layout of the SYP mechanism showing the relationship to existing machine elements.

C6-14(10) Enlarged view of the SYP mechanism envelope.
It was found that a good measure of the potential suitability of a mechanism was the variation in the yarn loop length which it would collect and release over a range of movement of the control crank. This information provides two useful measures:

a. the variation in the absolute yarn loop length gathered by the mechanism throughout its range of motion

b. the yarn loop length at various stages of the mechanisms range of motion relative to the length at some datum position

Measure 'a' is useful in that it enables an immediate comparison to be made between the performance of the mechanism and the ideal yarn loop length variations which are defined in appendix 3. However, increasing the absolute yarn loop length gathered and released by a particular mechanism configuration is simply a matter of increasing or decreasing the length of the output link. A much better measure of potential suitability is therefore 'b'. In particular, the ratio of maximum to minimum yarn loop

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length immediately gives an indication of whether a particular configuration is capable of providing the range of compensator amplitude required from minimum to maximum package size.

For the mechanism with the parameter values defined in table C6-13(8), the variation in accumulated yarn length over a range of control crank positions is shown in graph C6-16(9).

![Graph of yarn loop length variation against control crank position for the SYP mechanism defined in table C6-13(8)](image)

C6-16(12) **Graph of yarn loop length variation against control crank position for the SYP mechanism defined in table C6-13(8)**

Although this configuration provided a good output motion which closely matched the ideal curve, the ratio of maximum to minimum yarn loop length was only 95:78 (=1.22) whilst that required from the predictions of the YARVEL program (see appendix 3) was 3.59. This limitation on the range of motion of the output link was a recurring problem in all of the 7-bar mechanisms which were designed to fit into the available space on the current Masterspinner. Indeed, at one stage of the investigation, it was considered unlikely that an acceptable solution could be found. A further investigation of possible solutions was carried out on the configuration shown in C6-11(7)iv. The particular configuration investigated is shown in the upper left drawing in photograph C6-17(13)
C6-17(13) Various arrangements of 7-bar mechanism.

Although this configuration provided a minor improvement in the range of available amplitudes (ie ratio of max to min yarn loop length = 1.25), the change was still insufficient to provide the ratio required. Graph C6-17(14) shows the variation achieved.

C6-17(14) Graph of yarn loop length variation against control crank position for the mechanism shown in C6-17(13)

At this stage, it was observed that an increase in the range of output amplitudes could
be obtained by changing the position of one of the two fixed compensator rollers. This effect is explained in diagram C6-18(15).

Diagram C6-18(15)i shows the SYP configuration described earlier. For the case shown, the moving roller is assumed to follow a straight line motion with an amplitude of 'A'. If the gap between the two fixed rollers is the same as the diameter of the moving roller, then the amount of yarn collected and released during one cycle will be 2A.

C6-18(15)ii shows a configuration which is the same as C6-18(15)i except that the lower fixed roller has been positioned some distance 'c' below the compensator roller centre. The amount of yarn collected and released during one cycle of this compensator is A+(b-a), but b-a is calculated by:

\[
b-a = \left[ \left( A+d \right)^2 + c^2 \right]^{0.5} - \left[ d^2 + c^2 \right]^{0.5}
\]

from which it can be seen that b-a is always less than A. Therefore, for a given amplitude A, the compensator in C6-18(15)ii will collect and release less yarn than that shown in C6-18(15)i, and this effect increases as d gets smaller and/or c gets larger.

To check the effect of moving the lower roller, a new series of investigations were carried out using the MECHANISM parametric. It seemed reasonable to envisage a mechanism in which the lower roller swings about a fixed pivot point. An examination of the available space indicated that it might be possible to locate such a pivot point on the under side of the 'U' section channel containing the guide cam. Such a mechanism is shown on all four configurations represented in C6-17(13), where the lower roller
has been shown in four positions.

With the lower roller rotated anti-clockwise through $90^\circ$ from it's SYP position, the amount of yarn taken by the compensator decreased. Table C6-18(16) shows the yarn accumulated by the SYP compensator with the lower roller alternatively at it's normal position and it's $90^\circ$ position.

<table>
<thead>
<tr>
<th>INPUT CRANK ANGLE (DEG)</th>
<th>ACCUMULATED YARN (mm) WITH SWINGING LOWER ROLLER</th>
<th>ACCUMULATED YARN (mm) WITH FIXED LOWER ROLLER</th>
</tr>
</thead>
<tbody>
<tr>
<td>+10</td>
<td>33</td>
<td>78</td>
</tr>
<tr>
<td>0</td>
<td>38</td>
<td>84</td>
</tr>
<tr>
<td>-20</td>
<td>46</td>
<td>93</td>
</tr>
<tr>
<td>-40</td>
<td>32</td>
<td>95</td>
</tr>
</tbody>
</table>

C6-19(16) Comparison between the yarn accumulated by the SYP compensator and the compensator with the swinging lower roller.

Table C6-20(17) shows the result of attaching a swinging lower roller to a compensator with an indirect yarn path.
<table>
<thead>
<tr>
<th>INPUT CRANK ANGLE (DEG)</th>
<th>ACCUMULATED YARN (mm) WITH SWINGING LOWER ROLLER</th>
<th>ACCUMULATED YARN (mm) WITH FIXED LOWER ROLLER</th>
</tr>
</thead>
<tbody>
<tr>
<td>+10</td>
<td>54</td>
<td>77</td>
</tr>
<tr>
<td>0</td>
<td>60</td>
<td>82</td>
</tr>
<tr>
<td>-20</td>
<td>70</td>
<td>93</td>
</tr>
<tr>
<td>-40</td>
<td>74</td>
<td>97</td>
</tr>
</tbody>
</table>

C6-20(17) **Comparison of yarn accumulated by an indirect yarn path compensator fitted with fixed and swinging lower roller.**

It can be seen from tables C6-19(16) and C6-20(17) that the addition of a swinging lower roller causes a marked increase in the range of accumulated yarn which the compensator can collect. Graphs C6-21(18) and C6-21(19) show the effect of the swinging roller. The SYP and indirect yarn path compensators fitted with the swinging lower roller are now capable of producing an accumulated yarn ratio of 1.4 which is still insufficient for a standard package. At this stage it was decided to undertake some experimental investigations on the 7-bar mechanism in an attempt to determine if a suitable amplitude range could be obtained by providing movable guide rollers of the type indicated diagrammatically in sketches xx11 and xx111 on page C3-7.
C6-21(18) Accumulated yarn against input crank angle for the
SYP compensator.

C6-21(19) Accumulated yarn against input crank angle for the
indirect yarn path compensator.

6.4 An engineering solution
To keep link masses low, small needle bearings were selected for the pivots and eccentric. These bearings were overlaid onto the "MAP" cross section, and links were added to provide what seemed to be reasonable housing thicknesses. The final outcome of this initial design process is shown in the following photograph (C6-22(20)). This photograph also shows the swinging lower yarn guide roller, which
was introduced in the previous section. As can be seen, although part of the mechanism extends behind the bulkhead, there is no interference with the machine structure or major components.

C6-22(20) First layout of the proposed mechanism into the machine cross section.
Detail design of the mechanism components were carried out using the 3D-modeller to check assembly and location. Photographs C6-24(21), C6-24(22), and C6-25(23) on the following two pages show a 3D-model of the input link, part of the fully detailed component, and the final mechanism assembly drawing. It should be noted that this first prototype was designed to have a locking control crank which could be secured in position by tightening a clamping nut. Consideration of a control link from the package support arm was to be considered later.
C6-24(21) Details of the compensator input link.

C6-24(22) Partial 3D-model of the input link for assembly checking.
C6-25(23) Details of compensator assembly.

It is obviously important to the high speed performance of the mechanism that the mass of the moving parts are kept to a minimum. At a yarn delivery speed of 500 m/min, the mechanism cycles at just under 10 Hz, so inertia forces will be high unless links are designed to be as light as possible. The detailed manufacturing drawings presented in appendix 10 show that considerable attention was given to minimising mass.

Since the control crank moves slowly during package build, it is relatively unimportant from the point of view of mass reduction. The control link and connecting link are both of low mass. However, the output link, and three pivot input link are both high speed and potentially high mass members. The output link oscillates about the pivot at $X_1, Y_1$ and can therefore be balanced to reduce the forces likely to cause vibration. The centre of rotation of the input link is, however, continually changing, so effective balancing is not possible. The input link is therefore a potential source of vibration and inertia forces, and requires special attention in order that these are minimised. Drawing MC.1 in appendix 10 shows the approach adopted in the design of this component. The three housings are machined to take low profile needle bearings, and are
separated by narrow webs to minimise weight. A production version of the mechanism might have high speed links made from injection moulded thermoplastic, which would be acceptable with quantities in excess of 10,000 per year. For the prototype, a low density aluminium alloy was chosen for the input link.

The mechanism was provided with rigid side plates so that bearing alignments and fixed pivot locations could be maintained. The side plates (visible in MA.1 - General Assembly in appendix 10) are separated by pillars which also provide the location necessary for the machining of the housings.

6.4.1 Manufacture

Photograph C6-26(24) shows the mechanism prior to the fitting of the second side plate, whilst C6-26(25) shows the mechanism with the second plate in place.
The mechanism as manufactured was close to the details defined previously, and shown on drawings in appendix 10, although a necessary modification was made to the output crank (drawing MC.14) by removing metal to reduce weight. A small amount of metal was also removed from the control link to allow the full range of movement required without fouling occurring on the control crank.

6.5 Experimental evaluation

The first phase of testing the mechanism was carried out on a bench with the input shaft turned by hand. The mechanism turned freely, and with minimal backlash. The control crank was capable of rotation through the range predicted as being necessary, and the mechanism for clamping the control crank worked correctly.

A loop of yarn was passed around the end of the output link, and guided around two fixed rollers as shown in C6-28(26):
The mechanism input shaft was turned slowly, and the length of the yarn loop measured for a series of shaft positions. The test was then repeated for a number of different control crank positions. The results are plotted on graphs C6-28(27) and C6-29(28).
C6-29(28) **Polar plot of experimental loop length measurements**

<table>
<thead>
<tr>
<th>Yarn loop amplitude (mm)</th>
<th>Control crank angle (degrees)</th>
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<tr>
<td>Test #2: 65</td>
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<tr>
<td>Test #3: 54</td>
<td>70</td>
</tr>
<tr>
<td>Test #4: 44</td>
<td>100</td>
</tr>
</tbody>
</table>

Dead zone.

C6-29(29) **Limits of control crank rotation on the experimental linkage**
Because of interference between the input link and the control link the useful rotation of the control crank was limited to within 0° and 100°, and the amplitude of the accumulated loop at various crank angles within this range is shown in the table below C6-29(28). It can be seen that the ratio of maximum to minimum is 65/44 = 1.47 which compares well with the 1.4 which was predicted theoretically on page C6-20.

It is obvious that the theoretical predictions of yarn accumulation closely matches the measured accumulations. Although gratifying in one respect, it also confirms that neither the basic 7-bar mechanism, nor such a mechanism fitted with a moving lower roller, can be made to provide a suitably large range of loop size variation to meet the requirements of a full range of package sizes.

6.6 An alternative mechanical solution.

It now seems unlikely that a 7-bar mechanical compensator can be designed which will fit within a 'reasonable' space envelope. However, the work which has been undertaken in attempting to provide the necessary variation in motion amplitude has developed an understanding of the potential for the use of variable position yarn runners as a solution to this problem. It might reasonably be argued that, since the entire purpose of using 5 or 7-bar mechanisms was to make use of their variable amplitude capabilities, it is pointless to consider using them if an additional amplitude variation mechanism must be added. This argument persuaded the author to consider using the variable runner system of amplitude variation with the much simpler and cheaper compensation device outlined in diagram C6-31(30).
Diagrammatic representation of the circular arc compensator with adjustable runners.

In this device, yarn is caused to run through or around runners 1 and 2, prior to and after passing around pin 'p' which is following a circular path of radius 'R'. Runners 1 and 2 are each distance 'C' from the horizontal centre line through the centre of rotation of 'p'. 'C' is a variable, and is increased to reduce the amount of compensated yarn as described by equation 2.C6 below.

\[
\text{Yarn loop length} = A + B = \left[ x^2 + \left( C + Y \right)^2 \right]^{0.5} + \left[ x^2 + \left( C - Y \right)^2 \right]^{0.5}
\]

................................. (2.C6)

where:

\[ x = XC + \left( R \cdot \cos \theta \right) \]

and:

\[ Y = R \cdot \sin \theta \]

A short computer program was written to investigate the effect of parameter changes on the yarn loop length as described by equation 2.C6, and the result of this is typified by the curves shown in graphs C6-32(31) and C6-32(32).
Graph C6-32(31) shows how the yarn loop length varies as point 'p' rotates along a path of radius 'R'. The graph also shows the variation in yarn loop length which can be obtained by changing the value of 'C'. For the parameter values chosen, the ratio of maximum to minimum yarn loop amplitude can be seen to be $54/13 = 4.15$, which is sufficient to meet the needs of the process.

The variation in 'C' which is required to attain particular values of yarn loop amplitude
are shown in graph C6-33(33).

C6-33(33)  Variation of yarn loop amplitude with dimension 'C'

It is convenient to compare the above graph to C5-7(8) which shows the ideal (theoretical) variation in yarn loop amplitude required for various sizes of package. This is reproduced from chapter 5 below.

C5-7(8)  Variation in loop and compensator amplitude for varying package size.

Combining data from C6-33(33) and C5-7(8) provides information for graph C6-34(34), which shows how 'C' must be varied to provide the necessary variation in yarn loop amplitude as the package grows in size.
\[ y = -14.5695 + 0.9548x - 0.0011x^2 \quad R = 1.00 \]

**C6-34(34) Variation of dimension 'C' with package size.**

The graph shows the mathematical relationship between 'C' and package diameter in an equation where the vertical axis (C) is represented by 'y', and the horizontal axis (package diameter) is represented by 'x'.

It now remains to check the shape of the curves on graph C6-32(31) against the ideal yarn loop variations predicted by "YARVEL" in graph C5-4(2) (reproduced below).

**C5-4(2) Yarn loop size variations for a range of package sizes.**

Once again, the key comparison is that which corresponds to the smallest diameter.
package where acceleration forces are at a maximum. Using a yarn stiffness of 8.3 grams/mm, graph C6-35(33) shows that the induced tension "errors" due to imperfect yarn loop size control are considerable.

C6-35(33) Compensator generated loop length and associated tension fluctuation.

With predicted tension errors of around 75 grams, this device is obviously unsuitable for development as a compensator.
Chapter 7

GENERAL CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK
Chapter 7
General conclusions and suggestions for further work

7.1 General conclusions.
This section provides general conclusions for the work described in this thesis. Several chapters and appendices also contain sections which draw specific conclusions about the work which they describe. These earlier concluding sections are not repeated in this section, but reference is made to them.

Designing a compensator which will operate at yarn delivery speeds of 500 m/min is only one of the problems to be overcome in enabling the friction spinner to work at the speeds considered to be essential for commercial success in the mid 1990's. It is, however a key problem if package quality is to be maintained at a level which is at least no worse than that which is achievable today. For reasons described in chapter 3, the work undertaken in preparation of this thesis was limited to two avenues of investigation. These were (a) a completely novel pneumatic passive compensator and (b) a mechanical positive compensator.

In terms of the specification which was defined at the commencement of the work, and which is stated in sections 3.1 to 3.5, the pneumatic compensator fully satisfies all points. Using volumetric air flows and pressures which are within the capacity of the existing machine services, the device which is described in detail in chapter 4 is capable of providing adequate (in terms of the effect on package quality) control of yarn tension provided that a correctly profiled curved distribution bar (described in appendix 8) is fitted. The device caused no observable detrimental changes to yarn structure in terms of either strength or texture. In addition, the only moving part (the disc) appeared not to suffer from any measurable wear during many days of operation.

After proving the concept of pneumatic compensation with the first prototype, the second prototype successfully removed the problems which were encountered at first. By increasing the size of the disc, it was possible to control a higher yarn tension with the same air flow and pressure drop. The second prototype also overcame the problem of disc insertion on start-up, in addition to the avoidance of fly build-up on the disc and nozzle walls.

The work on a mechanical positive compensator which is described in chapter 6 and
it's supporting analytical appendices was undertaken as an attempt to find a mechanical device which would function at high speed in much the same way as the mechanical compensators fitted to German, Swiss, and Italian open end spinning machines. When working at yarn delivery speeds of 500 m/min, the high velocity sliding motion existing in the cams of current systems exhibit life expectancies which are too short. The work described in chapter 6 shows that although 7-bar a mechanism can be designed which fits into the available machine framework, and which has an output motion which is close to the ideal predicted by analysis, there is insufficient variation available in the motion amplitude to be acceptable for the full range of package sizes.

Chapter 6 also shows that a mechanism which utilises only rotational pivots can provide sufficient amplitude variation if it is used in conjunction with variable position runners. It is shown however, that the yarn loop cannot be made to change in size sufficiently closely to that theoretically required, and the resulting tension variations are excessive. The general conclusion from this chapter is that none of the positive mechanical compensator concepts introduced in chapter 3 are capable of providing adequate control at the desired speed.

The work described in appendices 1, 2, 3, and 4 develop a theory for the explanation of yarn tension variations in conical package construction, and provide a means by which compensator behaviour may be mathematically modelled during the design activity. The conclusions on the way that machine parameters influence tension are given in section A1.8.

The fitting of a properly profiled curved distribution bar has been shown to make a significant reduction in the demands placed by yarn path length variations on a compensator. Even a circular arc approximation to the ideal profile carries significant tension variation penalties.

7.2 Suggestions for further work.

Much remains to be done in the definition and design of a satisfactory cone packaging system for high speed spinning. To recap on the major problem areas, these are:

- control of tension variations due to differences in demand between the
small and large ends of a conical package

- control of "long term" tension drop associated with increased package diameter

- control of "package bounce" under certain critical conditions

- the need to increase anti-patterning for the purpose of further improving package quality, without adversely affecting tension variation

A major impediment to progress is the absence of a widely acceptable measure of package quality. This, effectively, prevents the writing of an engineering specification based upon customer need. The result, is that the value of important parameters such as package density, end shape, and even tension itself are "quantified" by means of vague definitions such as:

- density should be as constant as possible throughout the package

- corners should be firm and well defined

- tension variations should be as small as possible

The problem with such definitions is that the machine designer never knows if he has succeeded or failed in the task, and there is a tendency to believe that what is necessary is always just a little better than what has been achieved. This of course is dangerous, since "perfection is the enemy of good enough", and it is possible that an adequate and cost effective solution might be rejected for a functionally superior but expensive alternative. The main purpose of these observations is to stress the urgent requirement for a carefully considered and quantified specification for an acceptable package. This is best undertaken within the industry by those involved in selling, buying, and using the end product.

Regardless of the problems brought about by the lack of a detailed specification, there are a number of engineering tasks which can be identified as being necessary before a suitable system can be designed.
Work on the novel pneumatic compensator is still in its infancy. The present prototypes have by no means been optimised, and owe as much to good judgement as to mathematical synthesis. A further prototype is required to test the hypotheses regarding the minimisation of volumetric air flow without unduly sacrificing the advantages of small inertia. Experimentation is also required to determine the best shape of the disc and wall profiles.

One of the most promising avenues for the development of this device is in the use of fluidics to control air flow rate. The quick response of the system to flow rate changes makes it conceivable that the pneumatic compensator could, instead of being a purely passive device, be turned into a positive device with the potential for precise tension control. Figure C7-4(1) illustrates in outline how such a system might work.

C7-4(1) Closed loop control of the pneumatic compensator.

C7-4(1) shows the disc in the nozzle with the addition of two pilot holes machined into the nozzle wall. In the condition where yarn tension was fluctuating between acceptable upper and lower limits, the air flow control valve would be be open at a constant setting, and the disc would oscillate over a range between, but not over, the two pilot holes. If tension decreased below an acceptable value, the disc would move
down the air flow away from the nozzle throat, and would start, at some point, to intermittently cover one of the pilot holes. This "blocking" of the pilot hole would be made to cause the flipping of a fluidic switch which would reset the main air control valve so that flow was increased and tension restored. Alternatively, a high pressure would tend to move the disc towards the nozzle throat, which might be made to cause a decrease in flow and a resulting drop in tension. An interesting alternative might be to use opto-electric devices to sense the position of the disc and hence provide feedback to the air control valve. In practice, an array of sensing devices would be required to provide sufficient sensitivity.

The pneumatic compensator also offers potential for the controlled storage of large quantities of yarn during a splicing operation.

Finally, an interesting concept is the union of a pneumatic compensator with a linkage based positive compensator of the type described in chapter 6. Since mechanisms can only approximate the ideal motion of a yarn accumulation device, there will always be an inbuilt error which will manifest itself in the form of tension variations. These variations can be reduced by the fitting of a compliant element in the yarn path. This element may be in the form of a spring loaded member like the current bollard compensator, or could be a pneumatic device similar to that described in this thesis. The advantage of using the pneumatic device as a compliant element for a mechanical compensator is the same as that which may be put forward when comparing the pneumatic and bollard devices as stand alone passive compensators, ie the minimisation of induced loading due to inertia forces.

7.2.2 Mechanism compensation.

The mechanical compensator of chapter 6 has been shown to be capable of providing a degree of yarn accumulation and release which substantially reduces the cyclical tension variations which are present when no form of compensator is fitted. The next stage in the development of this device is the optimisation of design parameters, to provide optimum motion within the constraints of space, mass, forces, and stress. This aspect is being pursued within the Engineering Design Institute and Mechanical Engineering Department at Loughborough University of Technology by the author and his colleagues.
7.2.3 Long term tension reduction.

The reason for the reduction in the mean level of yarn tension as package size increases, remains scientifically unproven, although the effect is clearly visible from the graphs plotted from "TENSION" in appendix 2. The general consensus of opinion (and one which is shared by the author) is that the mean tension drop experienced as the package increases in size can be corrected by a proportional increase in the peripheral speed of the drive tyre. It has also been suggested however, that some aspects of tension reduction could be due to the changing drive geometry as the tyre/package contact line changes, and slip between the tyre and the package surface. Neither of these possibilities have been rigorously investigated in this programme of work, and remain to be considered at a later date.

7.2.4 Package bounce.

A study of package bounce was not an original objective of this SERC project. However, a number of experimental investigations were carried out for Platt Saco Lowell in an attempt to determine the cause of high amplitude package bounce on some machines in service. The reason(s) for bounce were not identified during these tests, because of a complete failure to reproduce the effect on either the Loughborough Spin Tester, or at the Accrington works.

The latest PSL machines do not seem to suffer from bounce, probably due to the inclusion of a new spring / damper device on the package support arm. However, many machines in service, which are fitted with the older package arm assembly can be susceptible to bounce in rare circumstances. A full theoretical and experimental analysis of package arm dynamics is therefore desirable to provide adequate design information for future designs and modifications.
References
10. References

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(10) British patent application. GB 2024269. Filed 2 April 1979. Barber-Coleman Company, USA. "Self threading tension compensators".
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Appendix 1

AN ANALYSIS OF STEADY STATE TENSION AND THE RESPONSE TO STEP OUTPUT CHANGE
Appendix 1

An analysis of steady state tension and the response to step output change.

A1.1 Outline

The package winding process on the Platt Saco Lowell Masterspinner generates tension in the yarn prior to it being wound on to the package. This tension is vital to the construction of stable packages which will not suffer from undue deformation during normal usage. It is also important that tension variations are controlled within "acceptable" limits to ensure that yarn density is "relatively" constant at all points in the package. This control over density is necessary to ensure that yarn which is dyed on the package is of even shade. Tension control is also required to ensure that the yarn is placed upon the package surface in a precise manner, and hence prevent the occurrence of "stitching" as described in Chapter 1.

This appendix provides the basic analysis necessary for an understanding of the factors influencing yarn tension. Although the analysis is directed specifically to the generation of thread tension during winding, parts are equally applicable to other processes such as wire winding and steel rolling. For this reason, the analysis makes use of the word "strand" rather than "yarn" to make the generality clear.

The basis of tension generation is indicated in the diagram on the next page. An untensioned strand (ie axial tension = zero) is passed between a pair of contra-rotating rollers which grip the yarn so that no relative slip occurs. In the case of yarn winding on a spinning machine, the strand is pulled by the rollers out of the yarn manufacturing area (ie the friction rollers or the rotor). Between the manufacturing area and this first pair of rollers, the yarn tension is very low in comparison to the tension generated after the first pair of rollers. Experimental assessment of pre-roller tension suggests a figure significantly less than 2 gram (21). For the purpose of this analysis, pre-roller yarn tension is assumed to be zero.

The peripheral velocity of the first pair of rollers (marked "m" on the diagram, and represented by the delivery rollers on the Masterspinner) is set at the required yarn delivery rate. In the case of yarn winding, the second pair of rollers on the diagram (marked "q") represent the package and the drive roller.
Tension is generated in the strand by ensuring that rollers "q" rotate with a higher surface velocity than rollers "m", thereby making the strand velocity at "q" greater than that at "m". Since, in the steady state, the volume flow rate entering "m" must be the same as the volume flow rate entering "q" if the strand is not to break, the diameter of the strand entering "q" must be less than the diameter entering "m". A steady state will therefore be achieved when the strand tension is at a level where the stretched strand diameter results in equal flow rates past the two pairs of rollers.

The analysis in this appendix considers the effect on tension of setting the peripheral velocities of the two pairs of rollers at certain specified values. The response of the tension to step changes in roller velocity is also considered.

Since, in fact, the peripheral velocity of a conical package varies from one end to the other, the effect on yarn tension is the same as if the peripheral velocity of rollers "q" was constantly changing. The analysis of this effect is the subject of Appendix 2.

A1.2 Generation of tension in a strand passing between two pairs of driven rollers
Consider an elastic strand passing between two pairs of rotating rollers as shown in the following diagram.

**Assumptions:**
- there is no slip between the strand and the rollers
- the strand retains a circular cross section at all times
• the strand is elastic at all times and retains a constant stiffness per unit length
• the strand is weightless

Nomenclature:

\[ \begin{align*}
\text{do} & \quad = \text{diameter of strand after passing through second set of rollers} \\
\text{ds} & \quad = \text{diameter of strand prior to entering first pair of rollers} \\
\text{Pm} & \quad = \text{peripheral speed of rollers m} \\
\text{Pq} & \quad = \text{peripheral speed of rollers q} \\
\text{Z} & \quad = \text{distance over which the strand is stretched} \\
\Delta Z & \quad = \text{the extension in the strand held between the two pairs of rollers}
\end{align*} \]

Consider a case where the tension (T) in the strand held between the two pairs of rollers was zero. If there was no slack in the strand (ie the path followed by the strand between the rollers was straight), then the volume (Vol_u) of the strand held between the two pairs of rollers would be given by:

\[ V_{ol_u} = \frac{\pi}{4} (ds)^2 Z \]  \hspace{1cm} (1.A1)

But if the strand between the rollers is under tension and, as a result, has a reduced diameter (do), then the volume (Vol_s) of stretched strand between the two pairs of rollers would be given by:

\[ V_{ol_s} = \frac{\pi}{4} (do)^2 Z \]  \hspace{1cm} (2.A1)

If the length of stretched strand between p and q were released, it would "spring back" to the length Z-\(\Delta Z\), and this unstretched strand will have the same volume as \(V_{ol_s}\).

Therefore, from equation 2.A1:

\[ \frac{\pi}{4} (do)^2 Z = \frac{\pi}{4} (ds)^2 (Z - \Delta Z) \]  \hspace{1cm} (3.A1)
and:

\[
\left( \frac{do}{ds} \right)^2 = 1 - \frac{\Delta Z}{Z} \tag{4.A1}
\]

or:

\[
\Delta Z = Z \left[ 1 - \left( \frac{do}{ds} \right)^2 \right] \tag{5.A1}
\]

Under steady state conditions (ie over a time period during which tension is constant):

Volume flow rate of strand past point m = Volume flow rate of strand past point q

ie, in time interval \( \delta t \):

\[
V_{fm} \delta t \frac{\pi}{4} (ds)^2 = V_{fq} \delta t \frac{\pi}{4} (do)^2
\]

where \( V_{fm} \) is the yarn velocity approaching the first pair of rollers
and \( V_{fq} \) is the yarn velocity approaching the second pair of rollers

hence:

\[
\frac{V_{fm}}{V_{fq}} = \left( \frac{do}{ds} \right)^2 \tag{6.A1}
\]

Substituting 6.A1 into 5.A1:

\[
\Delta Z = Z \left[ 1 - \frac{V_{fm}}{V_{fq}} \right] \tag{7.A1}
\]

If \( K_n \) is the force constant (stiffness) of a strand of length \( n \):

\[
\text{Tension (T)} = K_n \Delta Z \tag{8.A1}
\]

and substituting 8.A1 into 7.A1:

\[
T = K_n \Delta n Z \left[ 1 - \frac{V_{fm}}{V_{fq}} \right] \tag{9.A1}
\]
where, in this case, $\Delta n = \Delta Z$

### A1.3 Analysis of tension with a corrected value of $K_n$

$K_n$ can be determined experimentally by taking a length ($n$) of the strand and measuring deflection against load. An experiment to determine $K_n$ is described in Appendix 2, where it is shown that for a particular type of yarn $K_{300} = 83$ gram/cm. Hence, it is possible to express the relationship:

$$T = K_{300} \Delta Z$$

It is obvious that $K_n$ will depend upon the test length ($n$), and that, for example:

$$K_{150} = 2 K_{300}$$

ie for a 150 mm long test length, twice the force will be required to induce a particular yarn extension than will be the case with a 300 mm long test length. Now, even when yarn is being wound on to a package with the aid of a curved yarn distribution bar (resulting in a constant yarn path length), the fact that the tension changes during the guide cam cycle, means that the length of unstretched yarn (ie the length of yarn if tension were to be removed) also changes. Therefore, $K_n$ also changes. The following analysis relates strand extension to tension, with an "inbuilt correction" for $K_n$.

---

**A1-5(2) Yarn extension parameters**

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Page: A1 - 5  Appendix 1
Unstretched strand length \( (n) = Z - \Delta Z \)

But:

\[
K_n = K_{300} \frac{300}{n}
\]

ie

\[
K_n = K_{300} \frac{300}{(Z - \Delta Z)}
\]

But:

Tension \( (T) = K_n \Delta Z \)

Therefore:

\[
T = K_{300} \frac{300}{(Z - \Delta Z)} \Delta Z
\]

and, dividing by \( \Delta Z \):

\[
T = \frac{300K_{300}}{(\frac{Z}{\Delta Z} - 1)}
\]

Substituting \( K_{300} = 83 \text{ gram/cm} \) into the above equation:

\[
T = \frac{2490}{\left(\frac{Z}{\Delta Z} - 1\right)}
\]

and finally:

\[
\Delta Z = \frac{Z}{\left(\frac{2490}{T} + 1\right)}
\]

where \( \Delta Z \) and \( Z \) are in cm and \( T \) is in grams

**A1.4 Relationship between tension \((T)\) and the volume of strand in the free length.**

From the previous section:

\[
\text{Volume (V) of free strand} = \frac{\pi ds^2 (Z - \Delta Z)}{4}
\]
therefore:

\[ \Delta Z = Z - \frac{4V}{\pi ds^2} \] .................................................. (12.A1)

combining 10.A1 and 12.A1:

\[
T = \frac{300 K_{300}}{\left( \frac{Z}{Z - \frac{4V}{\pi ds^2}} \right) - 1}
\]

ie:

\[
T = \frac{300 K_{300}}{\left( \frac{1}{1 - \frac{4V}{\pi ds^2 Z}} \right) - 1}
\]

and finally:

\[
T = \frac{300 K_{300}}{\left( \frac{1}{1 - \frac{V_s}{V_u}} \right) - 1} \] .................................................. (13.A1)

where \( V_u \) is the volume of a strand of length \( Z - \Delta Z \) with tension = zero, and \( V_s \) is the volume of a strand in tension with a stretched length = \( Z \).

A1.5 A computer model of tension variation due to step changes in roller velocity.

Equations 1.A1, 2.A1, 3.A1, 11.A1, and 13.A1 were written into a computer model to enable an investigation to be made of the effect on tension of step changes in roller velocity. The flow chart for this program follows.
Set values of:
- unstretched strand diameter
- peripheral velocity of input rollers
- peripheral velocity of output rollers
- strand path length between rollers
- initial strand tension
- size of time increment used during the iteration

Calculate the unstretched strand volume from equation 1.A1

Calculate strand extension from eqn 11.1A

Calculate the stretched strand diameter from equation 3.A1

Calculate the stretched strand volume from equation 2.A1

Calculate the volume change in the strand between the two pairs of rollers in the selected time increment

Calculate the free strand length at the end of the selected time increment

Calculate strand tension at the end of the selected time increment from equation 13.A1

Output the value of tension calculated

Set start tension to last value calculated

Increase time?

Yes

No

Stop

A1.6 Computer model coding for the calculation of strand tension

The following coding was written in BASIC for a Sharp PC-1211 hand held pocket computer:

10 PRINT "ENTER UNSTRETCHED STRAND DIAMETER"
20 INPUT D
30 PRINT "ENTER START TENSION"
40 INPUT T
50 PRINT "ENTER INPUT ROLLER PERIPHERAL VELOCITY"
60 INPUT P
70 PRINT "ENTER OUTPUT ROLLER PERIPHERAL VELOCITY"
80 INPUT Q
90 PRINT "ENTER STRAND LENGTH BETWEEN ROLLERS"
100 INPUT L
102 PRINT "ENTER INCREMENTAL TIME STEP SIZE"
104 INPUT H
105 REM CALC EQN 1.A1
106 V= 3.1415 *(D^2) * L / 4
109 REM CALC EQN 11.A1
110 E=L / ((2490 / T) + 1)
129 REM CALC EQN 3.A1
130 F=((D^2) * (L - E) / L)^0.5
131 REM CALC EQN 2.A1
132 W=3.1415 * (F^2) * L / 4
139 REM CALC CHANGE IN STRAND VOLUME IN TIME H
140 G = (3.1415*(D^2)*P*H/4)-(3.1415*(F^2)*Q*H/4)
149 REM CALC STRETCHED STRAND VOLUME AT THE END ON TIME INCREMENT
150 X=W+G
159 REM CALC EQN 13.A1
160 U=2490/((1/(1-(X/V)))-1)
169 REM OUTPUT VALUE OF TENSION
170 PRINT U
179 REM RESET START TENSION TO CALCULATED VALUE
180 T=U
189 LOOP BACK FOR NEXT ITERATION
190 GOTO 110

A1.7 Output from the tension program of section A1.6
As expected, the changes in tension predicted by the model were independent of the
unstretched yarn diameter. The results tabulated on the following pages illustrate the
program output, and these are plotted and discussed in section A1.8.
### Input values

Unstretched yarn diameter = 0.1 mm

Start tension = 100 grams

Input roller velocity = 3333 mm/sec

Output roller velocity = 3333 mm/sec

Length of yarn path between rollers = 300 mm

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**A1-10(3) Predicted tension variation with various iteration step sizes**
**Input values**
Unstretched yarn diameter = 0.1 mm
Start tension = 100 grams
Input roller velocity = 8333 mm/sec
Output roller velocity = 8333 mm/sec
Length of yarn path between rollers = 300 mm
Time increment = 0.0025
Yarn path length 300 mm 320 mm 340 mm

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**A1-11(4) Predicted tension variation with various iteration step sizes**
**Input values**

Unstretched yarn diameter = 0.1 mm
Input roller velocity = 8333 mm/sec
Length of yarn path between rollers = 300 mm
Time increment = 0.0025

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*Output roller velocity (mm/sec) 8416 8499 8499*

**A1-12(5) Predicted tension variation with various iteration step sizes**

The following graph (A1-13(6)) is drawn from the data in table A1-10(3), and illustrates the effect of altering the size of the time increment on the results. The effect is small, and gets less as the increment reduces. The difference between tension after 0.24 seconds, calculated with increments of 0.02 seconds and 0.01 seconds is 20%, whilst the difference calculated with increments of 0.01 seconds and 0.0025 seconds has reduced to 13%. However, the difference between calculated values of tension for increments of 0.005 seconds and 0.0025 seconds is only 4%.

For this reason, it was decided that all subsequent calculations using the model would be carried out with a time increment of 0.0025 seconds.
A1-13(6) Tension against time. The effect of program increment size

The graph plotted below is from the data in table A-12(5), and shows how, for given peripheral velocities of input and output rollers, the tension in the yarn will approach the same steady state value. The graph also illustrates how the rate of approach to the steady state is greater with correspondingly greater differences between the initial (transient) value and the steady state value.

A1-13(7) Tension against time. Variation of tension with respect to time for different values of initial tension

The following graph (A1-14(8)) is also from the data of table A1-12(5), and shows how the steady state value of yarn tension depends upon the relative velocities of the input and output roller velocities. For both of the curves which are plotted, the input roller peripheral velocity was set at 8333 mm/second. With the output roller peripheral
velocity at 8416 mm/sec the steady state tension is approximately 35 grams, whilst with the output roller peripheral velocity set at 8499 mm/sec the steady state tension has increased to approximately 50 grams.

![Graph showing tension against time for two values of output roller velocity](image1)

**A1-14(8) Tension against time. Variation of tension with time for two values of output roller velocity**

The graph plotted below is from the data in table A1-11(4), and illustrates the effect on yarn tension of changing the free yarn length (ie the yarn path length between the input and output rollers). Although the steady state tension is independent of the free yarn length, the time duration of the transient increases as the free yarn length increases.

![Graph showing tension against time for three values of free yarn length](image2)

**A1-14(9) Tension against time. Variations in tension with time for three values of free yarn length**
Conclusions on appendix 1

- the steady state tension induced in a strand gripped by and fed between two pairs of driven rollers depends upon the relative peripheral velocity of the rollers.

- the same steady state will be achieved, regardless of the initial value of tension in the strand.

- the rate at which the initial tension approaches the steady state is proportional to:
  
  - the difference between the initial tension and the steady state tension
  
  - the peripheral velocity of the rollers

- the rate at which the initial tension approaches the steady state is inversely proportional to:
  
  - the free strand length
Appendix 2

TIME DEPENDENT TENSION VARIATIONS DUE TO CHANGES IN YARN DEMAND ON A CONICAL PACKAGE
Appendix 2
Time dependent tension variations due to step changes in yarn demand.

A2.1 Outline
Referring to the diagram shown below, the parameters for this analysis are:

![Diagram of machine elements](image)

A2-1(1) Diagrammatic representation of machine elements

Constants:

- $P_m$ = peripheral speed of delivery rollers
- $V_{fm}$ = volume flow rate of yarn just prior to the delivery rollers
- $ds$ = diameter of the unstretched yarn

Variables:

- $P_q$ = peripheral speed of package at the nip point
- $T$ = tension in the yarn
- $V_{fq}$ = volume flow rate of yarn through the guide
- $V$ = volume of yarn between points $m$ and $q$
- $Z$ = the length of the yarn path between the delivery rollers and the point where the yarn is being wound on to the package. This length changes as the yarn guide traverses the package if a straight distribution bar is fitted.

The tension in the stretched yarn will be shown to be proportional to both the volume of yarn between $m$ and $q$, and the distance between $m$ and $q$.
This relationship is established in section A2.4 and will look similar to that shown below:

\[ T = f(V, Z) \]

If, therefore, we consider a number of time periods each of duration \( \Delta t \), and calculate the volume of stretched yarn and distance \( Z \) for each period, we can also determine the tension \( T \).

Assume in the first instance that \( P_m = P_q \) and that the stretched yarn tension is zero. This will remain the case as long as \( P_m, P_q, \) and \( Z \) remain constant.

Now consider what happens if \( P_q \) changes.

If \( P_q \) increases, the volume flow rate of yarn passing point \( q \) will also momentarily increase.

\[ V_{fm} < V_{fq} \]

and the volume of yarn between points \( m \) and \( q \) will decrease. If \( Z \) has remained constant, this will cause an increase in the tension of the yarn between \( m \) and \( q \). The result will be a rise in yarn tension.
A2.2 Numerical evaluation

Consider a number 'n' of consecutive time intervals. During each time interval, the value of $P_m$ and $P_q$ will remain constant, but may vary from one interval to the next. The time intervals are numbered 1, 2, 3, 4,..., n, and during the $i$th time interval the value of $P_q$ is $P_{qi}$ and similarly for $P_m$.

**During the 1st time interval.**

Let the yarn be unstretched so the volume of yarn between points $m$ and $q$ will be given by:

$$V = \frac{Z\pi(ds)^2}{4} \quad \text{(1.A2)}$$

From the notation established previously:

$$P_m = P_{m1}$$

and

$$P_q = P_{q1}$$

At the end of this time interval, the volume of yarn between $m$ and $q$ will have changed by the difference in volumes passing the delivery rollers and the nip point. From equation 1.A2

$$\text{Volume (} V_{m1} \text{) passing } m = P_{m1} \frac{\pi(ds)^2}{4} \quad \text{(2.A2)}$$

$$\text{Volume (} V_{q1} \text{) passing } q = P_{q1} \frac{\pi(ds)^2}{4} \quad \text{(3.A2)}$$

(We assume that a change in tension during a particular time interval only affects yarn diameter during the next time interval).
Change in stretched yarn volume during the 1st interval \((\Delta V_1) = V_{m1} - V_{q1}\)

Therefore, from 2.A2 and 3.A2:

\[
\Delta V_1 = P_{m1} \frac{\partial t \pi (ds)^2}{4} - P_{q1} \frac{\partial t \pi (ds)^2}{4} \tag{4.A2}
\]

ie

\[
\Delta V_1 = \frac{\pi (ds)^2}{4} \partial t (P_{m1} - P_{q1}) \tag{5.A2}
\]

At the end of the 1st time interval, the stretched yarn volume \((V_1)\) will be:

\[
V_1 = V + \Delta V_1
\]

If we know \(Z\) at the end of this time period, we can calculate \(T\) from \(T = f(V, Z)\), and the new diameter \((d_1)\) of the stretched yarn from \(d = f(T)\). We can then take \(d_1\) as the stretched yarn diameter into the next time interval.

During the 2nd time interval

During the 2nd time interval, the volume of yarn passing point \(m\) is given by:

\[
V_{m2} = P_{m2} \frac{\partial t \pi (ds)^2}{4}
\]

But since \(P_{m2} = P_{m1} = P_m\), it follows that \(V_{m2} = V_{m1} = V_m\) where

\[
V_m = P_m \frac{\partial t \pi (ds)^2}{4}
\]

The volume of stretched yarn passing \(q\) is given by:

\[
V_{q2} = P_{q2} \frac{\partial t \pi (d1)^2}{4}
\]
The change in the stretched yarn volume during the second interval is given by:

\[ \Delta V_2 = V_{m2} - V_{q2} \]
\[ \Delta V_2 = P \frac{\pi(ds)^2}{4} - P_{q2} \frac{\pi(d_1)^2}{4} \]
\[ \Delta V_2 = \frac{\pi ds}{4} (P_{m} d_2 - P_{q2} d_1^2) \]

And the stretched yarn volume at the end of the 2nd time interval is:

\[ V_2 = V_1 + \Delta V_2 \]

Again, tension can be calculated from \( T = f(V, Z) \), and \( d_2 \) from \( d = f(T) \).

**During the nth time interval**

During time interval \( n \), the volume of yarn passing point \( m \) is given by:

\[ V_{mn} = P \frac{\pi(ds)^2}{4} \]

The volume of stretched yarn passing \( q \) is given by:

\[ V_{qn} = P \frac{\pi d_{(n-1)}^2}{4} \]

The change in stretched yarn volume during the nth interval is given by:

\[ \Delta V_n = V_{mn} - V_{qn} \]

Therefore, from equations 6.A2, 7.A2, and 8.A2:

\[ \Delta V_n = P \frac{\pi(ds)^2}{4} - P \frac{\pi d_{(n-1)}^2}{4} \]
hence,
\[
\Delta V_n = \frac{\pi \, \partial t}{4} \left[ P_m (dS)^2 - P_{qn} \, d_{(n-1)}^2 \right]
\] ..................................... (9.A2)

The stretched yarn volume by the end of interval \( n \) is given by:

\[
V_n = V_{(n-1)} + \Delta V_n
\]
\[
V_n = V_{(n-1)} + \frac{\pi \, \partial t}{4} \left[ P_m (dS)^2 - P_{qn} \, d_{(n-1)}^2 \right]
\] ..................................... (10.A2)

To solve this last equation, we need an analysis of the velocity of the yarn at point \( q \) so that \( P_{qn} \) can be calculated at each time interval. This analysis is presented in the next section of this appendix.

**A2.3 Calculation of parameter \( (P_{qn}) \)**

The previous equation (10.A2) expresses the volume \( (V_n) \) between points \( p \) and \( q \) during time interval \( n \), in terms of \( V_{(n-1)} \), \( t \), \( P_m \), \( ds \), \( P_{qn} \) and \( d_{(n-1)} \). However, relating this analysis to cone winding, the parameter \( P_{qn} \) depends not only on the surface speed of the package at different points across its width, but also on the velocity of the guide which adds a component of velocity at right angles to the rotation of the package. In fact, we need to replace \( P_{qn} \) in the above equation by the velocity \( (V_y) \) of the yarn through the guide.

Consider that the package rotates in a number of steps, each step producing an equal angular rotation \( (\psi) \) of the package when measured about its developed surface. During each step, the guide will move distance \( w \) across the surface of the package. See figure A2-7(3).
Let velocity of yarn guide = \( U \)
Let width of package measured on the surface = \( M \)
Let the distance from the surface of the package at the small end, to the package focus = \( L \)
Let the number of segments across width \( M = N \)
Let the width of any segment = \( w \)

A2-7(3) Movement of yarn path over package surface

The length of each segment (\( w \)) is given by:

\[
\frac{M}{N} \]

(11.A2)

This of course assumes that the ratio between package rotational speed and guide velocity is constant. This is not the case but the effect will be considered in appendix 4.

Consider what happens whilst the yarn is being wound onto a number of segments. First, consider that the yarn is being wound onto segment 1 in diagram A2-8(4), and then segment 2. Then consider the general case for the nth segment.
A2-8(4) Segments on the package surface

Geometry of segment 1
Consider the geometry of segment 1, and the length of yarn laid down within it during one pass of the guide.

Let the length of yarn laid down in segment 1 = \( a_1 \)

Let the angle subtended by the segment = \( \psi \)
From the diagram:

\[ 2\sin\left(\frac{\psi}{2}\right) = \frac{h_1}{L + w} \]

and

\[ h_1 = 2(L + w)\sin\left(\frac{\psi}{2}\right) \]

where

\[ \alpha = 90 - \frac{\psi}{2} = \frac{1}{2}(\pi - \psi) \]

By the cosine rule, the length of yarn \(a_1\) deposited in segment 1 is given by:

\[ a_1 = \left( h_1^2 + w^2 - 2h_1w\cos\alpha \right)^{0.5} \]

If the time taken for the yarn guide to traverse distance \(w\) is \(\delta t\), then the average velocity \(V_y\) of the yarn demand during this time is given by:

\[ V_y = \frac{a_1}{\delta t} = \frac{1}{\delta t} \left( h_1^2 + w^2 - 2h_1w\cos\alpha \right)^{0.5} \]
**Geometry of segment 2**

\[ 2 \sin \left( \frac{\psi}{2} \right) = \frac{h_2}{L + 2w} \]

\[ h_2 = 2(L + 2w) \sin \left( \frac{\psi}{2} \right) \]

\[ a_2 = \left( h_2^2 + w^2 - 2h_2w \cos \alpha \right)^{0.5} \]

**Geometry of segment n**

\[ 2 \sin \left( \frac{\psi}{2} \right) = \frac{h_n}{L + n.w} \]  \hspace{1cm} (12.A2)

\[ h_n = 2(L + n.w) \sin \left( \frac{\psi}{2} \right) \]  \hspace{1cm} (13.A2)
\( a_n = \left( h_n^2 + w^2 - 2h_nw\cos\alpha \right)^{0.5} \) ................................ (14.A2)

Velocity of yarn \((V_y)\) during \(n\)th interval = \( \frac{a_n}{\partial t} \)

Therefore:

\[ V_y = \frac{P}{\pi n} = \frac{1}{\partial t} \left( h_n^2 + w^2 - 2h_nw\cos\alpha \right)^{0.5} \] ................................ (15.A2)

The equation for \(h_n\) is not in a convenient form because angle \(\psi\) is measured on the plane of the developed surface of the cone, and cannot immediately be related to the angular rotation of the package about it's axis of rotation. This relationship will now be established. The diagram on the left below represents the full developed surface of the package.

---

**A2-11(7) Package surface development**

Let \(\Omega\) = the rotational speed of the package (rad/sec)

Let \(\psi\) = the angle of the developed cone which is passed through during each time interval (\(\partial t\))

Let \(D\) = the small diameter of the package

Let \(\beta\) = the angle spanned by the developed cone.

Let \(s\) = the circumference of the small end of the package
From diagram A2-11(7):

\[ \beta = \frac{s}{L} \]

but

\[ s = \pi D \]

therefore:

\[ \beta = \frac{\pi D}{L} \]

In each incremental time unit (\(\partial t\)), the number of radians (\(R\)) through which the package will rotate is given by:

\[ R = \Omega \partial t \]

And the proportion (\(P\)) of a full revolution made in each time interval (\(\partial t\)) is given by:

\[ P = \frac{\Omega \partial t}{2\pi} \]

Therefore, the angle (\(\psi\)) of the developed cone which is passed through during each time interval (\(\partial t\)) is given by:

\[ \psi = \frac{\Omega \partial t \beta}{2\pi} \]

Therefore:

\[ \psi = \frac{\Omega D \partial t}{2L} \]

and equation 13.A2 can be rewritten as:

\[ h_n = 2(L + nw) \sin \left( \frac{D \Omega \partial t}{4L} \right) \]

We now have a set of equations describing the way in which the free yarn volume (\(V\)) changes in response to variations in the peripheral speed of the package. We now need to establish the expression \(T = f(V,Z)\) before the changes can be computed. The next section of this appendix is concerned with this expression.

**A2.4 Experimental determination of \(T = f(V,Z)\).**

For this experimental determination of \(T = f(V,Z)\), it was decided to use 30s count cotton
yarn. Besides being the yarn for which the University Masterspinner had been set by Platt Saco Lowell, it accounts for the largest production volume of the most widely used counts. Different equations for $T=f(V,Z)$ encountered with other materials and counts are not difficult to calculate as will be demonstrated later. All of the following tests and discussions assume that the yarn behaves elastically.

Experimental method.

A 300mm length of 30s count cotton yarn was secured in clamps on an Instron Tensile testing machine. The rate of loading was set at 50mm strain per minute, with the load scale at 200g full scale. Nine tests were carried out to determine the force constant. The results are shown below:

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<td>7.6</td>
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<tr>
<td>3</td>
<td>6.8</td>
</tr>
<tr>
<td>4</td>
<td>8.4</td>
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<tr>
<td>5</td>
<td>8.8</td>
</tr>
<tr>
<td>6</td>
<td>8.4</td>
</tr>
<tr>
<td>7</td>
<td>9.6</td>
</tr>
<tr>
<td>8</td>
<td>8.0</td>
</tr>
</tbody>
</table>

A2-13(8) Table of test results for yarn force constant (stiffness)

The average of these constants is 8.3 g/mm, and this value will be used in the subsequent calculations. Graph A2-14(9) indicates the load extension relationship for a 300mm length of yarn assuming the force constant derived above.
A2-14(9) Load v yarn length for a 300mm test length.

It will be obvious that a piece of yarn 600mm long will extend twice as much for a particular load, than a piece 300mm long. Similarly, a piece of yarn 150mm long will only extend half as much as a piece 300mm long under the same load.

Therefore, the force constant depends not only on the physical properties of the yarn, but also on its length. Therefore if the force constant for a 300mm length is 8.3 g/mm, the following table indicates the force constant for other test lengths.

<table>
<thead>
<tr>
<th>TEST LENGTH (mm)</th>
<th>FORCE CONSTANT (gm/mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>150</td>
<td>16.6</td>
</tr>
<tr>
<td>300</td>
<td>8.3</td>
</tr>
<tr>
<td>600</td>
<td>4.15</td>
</tr>
</tbody>
</table>

A2-14(10) Table showing the relationship between force constant and test length

Therefore, if the force constant for a test length of n mm is designated $K_n$, then:

$$ K_n = \frac{300}{n} K_{300} $$

ie

$$ K_n = \frac{2490}{n} $$

where the units of $K_n$ are g/mm.
The equation for the line in graph A2-14(9) is:

\[ \text{Load} = -2490 + 8.3 \text{ Yarn length} \]

or, using the symbols introduced earlier in this appendix:

\[ T = -2490 + 8.3Z \]

Replotting the graph for test lengths of 150mm and 600mm, produces A2-15(11) and A2-15(12).

![Graph A2-15(11) Load v Yarn length for a 150mm test length.]

\[ y = -2490 + 16.6x \quad R = 1.00 \]

![Graph A2-15(12) Load v Yarn length for a 600mm test length.]

\[ y = -2490 + 4.15x \]

The general expression relating tension (T) to K and Z is therefore:
\[ T = -2490 + K_n Z \]

\[ T = -2490 + \frac{2490}{n} Z \]

But the volume \( V \) of the free yarn between delivery and drive rollers is given by:

\[ V = \frac{\pi ds^2}{4} n \]

ie

\[ n = \frac{4}{\pi ds^2} V \]

so,

\[ T = -2490 + \frac{2490}{4V} \pi ds^2 Z \]

and finally,

\[ T = -2490 + 1955.64 \frac{ds^2 Z}{V} \] ................................. (17.A2)

where \( ds \) = the diameter of the unstretched yarn

\( T \) = yarn tension (g)

\( V \) = volume of free yarn

\( Z \) = length of stretched yarn

**A2.5 Effect of tension changes on yarn diameter.**

The equations so far developed in this appendix, have been derived to express yarn tension as function of the volume of free yarn and the length between the delivery and drive rollers. The equations show that tension is related to the volume flow rate entering the free length through the delivery rollers, and the volume flow rate leaving the free length between the package and the drive roller. However, the volume flow rate leaving the free length depends not only on the peripheral velocity of the drive...
roller, but also the diameter of the yarn in the free length. Obviously, for a given drive roller peripheral velocity, the flow rate leaving will be greater when the yarn diameter is large than when it is small. It is this effect which allows the tension to reach a "steady state", subject of course to short term variations due to the variations in the package peripheral velocity from large to small ends. This section of appendix 2 establishes the relationship between the diameter of the yarn under tension, to the diameter of the unstretched yarn and the tension magnitude. This relationship, together with the equation \( T = f(V,Z) \) is then used in the computer model of yarn tension which follows.

\[
\text{Volume of unstretched yarn} = \frac{\pi ds^2}{4} n
\]

\[
\text{Volume of stretched yarn} = \frac{\pi do^2}{4} Z
\]

where \( do \) = the diameter of the stretched yarn.

But,

\[
Z = n + \frac{T}{K_n}
\]

Therefore,

\[
\text{Volume of stretched yarn (VN)} = \frac{\pi do^2}{4} \left[ n + \frac{T}{K_n} \right]
\]

and finally,

\[
do = \left( \frac{4 \ (VN)}{\pi n + \frac{T}{K_n}} \right)^{0.5} \text{ ........................................................................................................ (18.A2)}
\]

A2.6 A computer model of tension variation

The following flow chart describes the computer model constructed to reflect the equations established in this appendix, and show how yarn tension variations occur as
the guide traverses the surface of the package. The validity of the model is discussed in section A2.9 by comparing it with the experimental values obtained by the method described in section A2.8.
A2.7 Computer program listing

The program was written in BASICA for a Victor V286 (IBM compatible) pc.

```
10 REM THIS PROGRAM IS CALLED TENSION AND CALCULATES THE VARIATION
20 REM IN YARN TENSION AS THE YARN GUIDE TRAVERSES ACROSS THE
30 REM SURFACE OF A CONICAL PACKAGE. THE PROGRAM ASSUMES A
40 REM CONSTANT VELOCITY MOTION OF THE YARN GUIDE.
50 REM
60 REM * THE PROGRAM WAS WRITTEN BY I.C.WRIGHT AS PART
70 REM * OF A COOPERATIVE RESEARCH PROGRAMME BETWEEN
80 REM * LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY AND PLATT
```

© I.C.Wright 1990
90 REM SACO LOWELL LTD.
100 REM

110 PRINT "DO YOU WANT OUTPUT TO THE PRINTER? (Y/N) ";
120 INPUT PRINTER$
130 IF PRINTER$ = "N" GOTO 150
140 LPRTINT "PASS N D1 Z1 VN PQN DS TENSION"
150 PRINT "DIAMETER OF UNSTRETCHED YARN (MM) = ";
160 INPUT DS
170 PRINT "PERIPHERAL SPEED OF DELIVERY ROLLER (M/MIN) = ";
180 INPUT PM
190 REM CHANGE UNITS OF PM TO MM/SECOND
200 PM = PM * 1000 / 60
210 PRINT "PERIPHERAL SPEED OF DRIVE ROLLER = ";
220 INPUT PQ
230 PRINT "DIAMETER OF PACKAGE ROLLER = ";
240 INPUT DP
250 REM CALC SPEED OF DRIVE ROLLER IN RAD / SECOND
260 PQ = PQ * 1000 * 2 / 60 / DP
270 PRINT "DIAMETER OF SMALL END OF PACKAGE = ";
280 INPUT D
290 PRINT "WIDTH OF PACKAGE MEASURED ON SURFACE = ";
300 INPUT M
310 PRINT "HOW MANY SEGMENTS ACROSS THE PACKAGE WIDTH (EVEN NUMBER) = ";
320 INPUT N1
330 PRINT "POINT ON SURFACE WHERE RUBBER TYRE DRIVES PACKAGE. GIVE A NUMBER BETWEEN 0 AND 1 (0 FOR SMALL END, 1 FOR LARGE END) = ";
340 INPUT XL
350 PRINT "PACKAGE INCLUDED ANGLE (RADIANS) = ";
360 INPUT GAMMA
370 L = D / 2 / SIN(GAMMA / 2)
380 REM DRIVEDIA = DIA OF PACKAGE AT THE TYRE
390 DRIVEDIA = D + XL * 2 * M * SIN(GAMMA / 2)
400 REM CAPTHETA = ROTATIONAL SPEED OF PACKAGE (RAD / SECOND)
410 CAPTHETA = DP * PQ / DRIVEDIA
420 PRINT "TIME TAKEN FOR YARN GUIDE TO MAKE ONE PASS OF PACKAGE = ";
430 INPUT TP
435 REM T = TIME FOR YARN TO BE LAID IN ONE SEGMENT
440 T = TP / N1
450 PRINT "HOW MANY PASSES DO YOU WANT TO ANALYSE? : ";
460 INPUT PASSES
470 PRINT "DISTANCE BETWEEN DELIVERY AND DRIVE ROLLERS = ";
490 \pi = 3.14159
500 \text{REM } \theta = \text{ANGLE OF DEVELOPED CONE PASSED THROUGH IN TIME INTERVAL T}
510 \theta = \text{CAP} \cdot \text{D} \cdot \text{T} / 2 \cdot \text{L}
520 \text{REM } w = \text{WIDTH OF EACH SEGMENT}
530 w = m / n1
540 \alpha = (\pi / 2) - (\theta / 2)
550 \text{REM } v = \text{UNSTRETCHED YARN VOLUME}
560 v = z \cdot \pi \cdot (ds^2) / 4
570 \text{REM ASSIGN FIRST YARN DIA = UNSTRETCHED DIA.}
580 \text{D}1 = \text{DS}
590 \text{DIM } a(\text{PASSES},n1)
600 \text{DIM } pion(n1)
610 \text{DIM } z1(n1)
620 \text{REM CALC DEMAND (PON) AND LENGTH (Z1) FOR EACH SEGMENT}
630 \text{IF PRINTER$ = "N" GOTO 790}
640 \text{LPRINT "DIAMETER OF UNSTRETCHED YARN = ";} ds
650 \text{LPRINT "PERIPHERAL SPEED OF DELIVERY ROLLER (MM/MIN) = ";} pm
660 \text{LPRINT "PERIPHERAL SPEED OF DRIVE ROLLER = ";} pq
670 \text{LPRINT "DIAMETER OF PACKAGE ROLLER (MM) = ";} dp
680 \text{LPRINT "DIAMETER OF SMALL END OF PACKAGE = ";} d
690 \text{LPRINT "WIDTH OF PACKAGE MEASURED ON SURFACE = ";} m
700 \text{LPRINT "FRACTION OF PACKAGE WIDTH FROM SMALL END WHERE TYRE CONTACTS = ";} xl
710 \text{LPRINT "DIAMETER OF PACKAGE AT THE TYRE = ";} drivenia
720 \text{LPRINT "ROTATIONAL SPEED OF PACKAGE (RAD/SEC) = ";} capptheta
730 \text{LPRINT "TIME FOR YARN TO MAKE ONE PASS = ";} tp
740 \text{LPRINT "TIME FOR GUIDE TO TRAVERSE ONE SEGMENT = ";} t
750 \text{LPRINT "DISTANCE BETWEEN DELIVERY AND DRIVE ROLLERS AT CENTRE = ";} z
760 \text{LPRINT "ANGLE OF DEVELOPED CONE PASSED THROUGH IN TIME T (RAD) = ";} theta
770 \text{LPRINT "ANGLE ALPHA (RAD) = ";} alpha
780 \text{LPRINT "UNSTRETCHED YARN VOLUME = ";} v
790 \text{FOR PASS = 1 TO PASSES}
800 \text{FOR SEGMENT = 1 TO N1}
810 \text{REM IF PASS IS ODD GO TO 870}
820 \text{IF PASS/2 - INT(PASS/2) > 0.1 THEN 870}
830 \text{REM SET N FOR PASSES FROM SMALL TO LARGE END}
840 n = n1 - segment + 1
850 goto 880
860 \text{REM SET N FOR PASSES FROM LARGE TO SMALL END}
870 n = segment
880 \text{HN} = 2 \cdot (l + n \cdot w) \cdot \sin \left( \theta / 2 \right)
890 \text{REM CALC YARN VELOCITY IN THIS TIME INTERVAL}
900 \text{PQ}(n) = (hn^2 + w^2 - 2 \cdot hn \cdot w \cdot \cos (alpha)) ^ 0.5 / t
The computer model described in the previous section was run for a number of different situations. Firstly, the way that the model described the effect on tension of variations in package diameter had to be established. The effect on tension of tension draft and cone angle was also of interest. Finally, the effect of varying the position of contact of the drive tyre on the package was of fundamental interest to machine design.
As will be seen later in this section, the shape of the distribution bar had an effect upon tension, and this persuaded the author to undertake an analysis of what would happen if a **curved** distribution bar were fitted. It was subsequently shown that a correctly profiled curved distribution bar might remove a worthwhile amount of tension variation. The program "Tension" was then modified to predict what the results of this modification would be.

### A2.8.1 "Tension" predictions for a straight distribution bar

Graph A2-24(13) shows how the model predicts tension variations during the winding of a package with a straight distribution bar. If the path length of the yarn was always the same regardless of the position of the yarn guide; tension would be lowest as the guide passed the mid point of traverse travelling from the small end to the large end (from right to left). Similarly, it would be expected that the point of highest tension would occur when the guide was at mid traverse moving from the large end to the small end (from right to left). Graph A2-24(13) however shows a tension variation which does not follow this pattern. For a package with a 40 mm diameter small end, the point of highest tension is predicted to be halfway between the large end and the mid point when the guide is moving from the large end towards the small end. It is proposed that this is due to the effect of the straight distribution bar which stretches the yarn towards the ends of the package. In moving from the large end towards the mid point, the yarn path length is reducing, thereby reducing the amount of stretch and tension. This effect is most notable with small diameter packages where the ratio of end diameters and end peripheral speeds is largest. In moving from the small end to the mid point, the yarn path length decreases and the package peripheral velocity is always lower than the yarn delivery speed. Both of these factors contribute to a reduction in tension, and this effect can also be seen in A2-24(13). From the mid point to the large end, the yarn path length increases and the package peripheral velocity is higher than the yarn delivery speed. Both of these factors work to cause an increase in tension as shown in the graph.
Graph A2-24(13) is the result of running the model with tension draft = 1, a yarn stiffness of 8 gram/mm (corresponding to KCW cotton 30 Ne), and 70 segments across the package width. For packages of the width produced on the Masterspinner, it was found that the improved accuracy obtained by increasing the number of segments decreased rapidly above 40 segments. Increasing the number of segments from 70 to 140 resulted in a change in calculated values of less than 0.5%. For this reason all runs of "Tension" were made with the number of segments set at 70.

Graph A2-25(14) shows the prediction of "TENSION" in regard to the effect of changes in tension draft. Again, the graph shows the predicted levels of tension with the guide moving from the small end to the large end (left to right), and from the large end to the small end (right to left). An increase in tension draft increases the yarn demand with respect to delivery, and would therefore be expected to increase yarn tension. This effect is shown clearly in A2-25(14) which is plotted for a package with a 40 mm diameter small end.
A2-25(14)  Calculated tension variations for 0.97, 1.0, and 1.05 tension draft

Graph A2-26(15) shows how "TENSION" predicts the changes in tension resulting from different included cone angles. As would be expected, the graph shows that maximum tension increases with increasing cone angle whilst tension at the small end of the package decreases. In the extreme, tension variations in yarn being wound onto a package with zero cone angle (a cylindrical package), will be due only to the stretching which takes place because of variations in the yarn path length. This, of course, is because the peripheral velocity (and hence yarn demand rate) of a cylindrical package is constant at all points along it's width. As the cone angle increases, the peripheral velocity increases in the large half of the package, and decreases in the small half. Maximum peripheral speed and maximum tension therefore increase, whilst minimum peripheral speed and minimum tension therefore decrease.
Graph A2-27(16) shows the effect of moving the position at which the drive tyre makes contact with the package. The offset of the tyre is measured from the small end of the package and lies between 0 and 1. If the tyre was to make contact on the smallest package diameter, the offset would be 0. If contact was at the largest diameter then offset would be 1. Contact two thirds of the way from the small to the large ends would represent an offset of 0.66.

The closer the contact point to the large end, the slower the package rotates for a given value of tyre peripheral speed. This reduction in package speed gives rise to a corresponding reduction in yarn demand and tension. Graph A2-27(16) plots the tensions predicted by "TENSION" for offsets of 0.5, 0.6, and 0.7.
Calculated tension variations for 0.5, 0.6, and 0.7 offset

On rotor spinning machines fitted to wind cylindrical packages, Platt Saco Lowell do not incorporate spring compensators because of the absence of package peripheral speed variation from one end to the other. These machines are, however, fitted with a curved distribution bar to reduce the amount of yarn stretching as the reciprocating yarn guide moves towards the ends of the package. The profile of these curved distribution bars approximate to part of a circular arc.

On the basis of the "Tension" output, it was decided to analyse the precise profile of a curved distribution bar to remove all of the tension introduced as a result of yarn stretch over the straight bar. This decision has important implications. A machine winding conical packages might be fitted with either a positive or passive compensator, and both have their own particular engineering problems. However, a problem common to both is the acceleration required of the device when operating at high speed. This problem is examined more closely later, but it will be obvious that a cheap and entirely passive device such as the curved bar, which can reduce the amplitude and acceleration of any compensator, must be worthy of investigation. This analysis of a curved distribution bar is shown in appendix 8.

Experimental investigation to evaluate the "Tension" model

One position on the Masterspinner was set to spin KCW 30 Ne cotton onto conical packages. Because the "Tension" model does not take the compensator into affect, this was removed from the machine. The change gears were set to give tension draft = 1.
A Rothchild yarn tensometer was secured to the front of the machine by means of a bracket fixed to the channel containing the scroll cam. The tensometer was connected to a storage oscilloscope with output to a chart recorder. Graphs A2.28(17) to A2.31(20) show typical plots from the recorder.

A2-28(17) **Experimental plot of tension variation (straight distribution bar, no compensator, 95mm package)**

Graph A2-28(18) shows experimentally recorded tension variations at a yarn delivery speed of 120 m/min, and a small end diameter of 95mm. The two marker pulses are from a proximity probe at the small end of the package which picked up the presence of a small bracket secured to the yarn guide mounting arm. The trace shows the
general trend predicted by "TENSION" in that the tension is low as the guide moves from the small end towards the centre, but increases as the guide approaches the large end. After reaching a peak soon after leaving the large end, there is a gradual reduction in tension until just before the small end when there is an increase due to the stretching effect of the straight distribution bar. Graph A2-29(18) is for a similar experimental arrangement to A2-28(17) except that the small end diameter is 103mm. The tension has shown a small decrease as predicted theoretically.

A2-29(18) Experimental plot of tension variation (straight distribution bar, no compensator, 103mm package)
A2.10 Theoretical prediction of fitting a curved distribution bar

The predictions by the "TENSION" program due to the fitting of a curved distribution bar are shown in graph A2-30(19) for different diameters of the package small end.

A2-30(19) Calculated tension variations when a curved guide bar is fitted

The curves provide a useful theoretical confirmation of the earlier assumptions made of the effect of keeping the yarn path length constant at all times. The shape of the curves now confirms the predictions made in section A2.8.1 on page A2.23 in regard to the causes of tension variation. Because yarn path length is now constant there is a decreased tendency for tension to increase at the ends of the package. Maximum tension now occurs only between the large end and the mid point as the guide is moving towards the small end. Absolute values of tension have correspondingly reduced, with the 40 mm diameter package exhibiting a reduction of over 50% in the maximum tension value.

A2.11 Experimental tension variation with a curved distribution bar

Graph A2-31(20) shows an experimental recording of the variation in tension when a profiled curved distribution bar is fitted. For this graph, the proximity probe which generated pulses from the yarn guide arm was located at the large end of the package. The trace shows general agreement with the trends predicted by "TENSION" in the previous graph in that tension rises to a maximum after the guide has left the large end and is moving towards the middle. There is now no increase in the tension at the small end because the stretching effect of the straight bar has been removed.
A2-31(20) *Experimental recording of tension variation with a profiled distribution bar fitted.*
Appendix 3

ANALYSIS OF YARN VELOCITY THROUGH THE GUIDE WHEN WINDING CONE SHAPED PACKAGES
Appendix 3.

Analysis of yarn velocity through the guide when winding cone shaped packages.

Assumption:

• there is no gap between the point where the yarn leaves the guide and the nip point between the package and the drive roller.

Development of complete package surface.

A3-1(1) Package surface development

Let:

M= the width of the package measured on the surface
P= the distance from the focus of the conical package to a point on the periphery on the small end
L= the distance between the reciprocating yarn guide and the focus of the conical package
y= distance of guide from small end of package
φ= angular measure around the developed surface of the package
\( \Phi \) = included angle of the conical package

\[ \Phi \]

\[ \text{Package} \]

\[ Q \]

\[ \text{Focus} \]

\[ r_m \]

\[ r \]

\[ R \]

\[ P \]

**A3-2(1) Cone geometry**

\[ R = \text{radius of large end of package} \]

\[ r = \text{radius of small end of package} \]

\[ r_m = \text{mean radius of package} = (R + r) / 2 \]

\[ Q = \text{distance from cone focus to mean diameter of package measured along the package surface} \]

Also:

\[ \frac{\delta L}{\delta t} = \text{velocity of the reciprocating guide} \]

and

\[ \frac{\delta \phi}{\delta t} = \text{angular velocity of the nip point around the developed surface of the cone} \]

Consider the surface element shown below in A3-2(2):
A3-2(2) Package surface element geometry

\[ \delta x = \left[ (\delta L)^2 + (L \phi)^2 \right]^{0.5} \]

\[ \frac{\delta x}{\delta t} = \left[ \left( \frac{\delta L}{\delta t} \right)^2 + L^2 \left( \frac{\phi}{\delta t} \right)^2 \right]^{0.5} \] .................................... (1.A3)

But \( L = P + y \)

and, from diagram A3-2(2):

\[ \sin \left( \frac{\Phi}{2} \right) = \frac{r}{P} \]

therefore:

\[ L = \frac{r}{\sin \left( \frac{\Phi}{2} \right)} + y \]

and:

\[ \frac{\delta x}{\delta t} = \left\{ \left( \frac{\delta L}{\delta t} \right)^2 + \left[ \frac{r}{\sin \left( \frac{\Phi}{2} \right)} + y \right]^2 \left( \frac{\phi}{\delta t} \right)^2 \right\}^{0.5} \] ......(2.A3)
If \( C_m \) is the circumference of the developed surface of the conical package measured at the mean diameter, then:

\[
C_m = 2\pi \left( P + \frac{M}{2} \right)
\]

ie

\[
C_m = 2\pi \left[ \frac{r}{\sin\left(\frac{\Phi}{2}\right)} + \frac{M}{2} \right]
\]

(3.A3)

Let \( C_d \) = the circumference of the rubber drive tyre,

If:

\[ \zeta = \text{the distance moved by a point on the drive tyre} \]

then

\[ \frac{\delta \zeta}{\delta t} = \text{the angular velocity of the drive roller} \]

and:

\[
\left( \frac{\delta \Phi}{\delta t} \right) = \frac{C_d}{C_m} \left( \frac{\delta \zeta}{\delta t} \right)
\]

therefore:

\[
\frac{\delta \Phi}{\delta t} = \frac{C_d}{C_m} \frac{\delta \zeta}{\delta t}
\]

(4.A3)

If the radius of the drive tyre = \( r_d \), then:

\[
\frac{\delta \Phi}{\delta t} = \frac{2\pi r_d}{2\pi \left[ \frac{r}{\sin\left(\frac{\Phi}{2}\right)} + \frac{M}{2} \right]} \frac{\delta \zeta}{\delta t}
\]

(5.A3)
\[ \frac{\delta \phi}{\delta t} = \frac{r_d}{\delta x} \frac{\delta \xi}{\delta t} \] ........................................ (6.A3)

\[ \left( \frac{r}{\sin \left( \frac{\Phi}{2} \right)} + \frac{M}{2} \right) \]

and finally:

\[ \frac{\delta x}{\delta t} = \left\{ \left[ \frac{\delta L}{\delta t} \right]^2 + \left[ \frac{r}{\sin \left( \frac{\Phi}{2} \right)} + y \right]^2 \left[ \frac{r_d}{\sin \left( \frac{\Phi}{2} \right)} + \frac{M}{2} \right]^2 \left[ \frac{\delta \xi}{\delta t} \right]^2 \right\} \]

.............................. (7.A3)
Appendix 4

AN ANALYSIS OF THE BEHAVIOUR OF THE FREE YARN LOOP DURING CONE WINDING
Appendix 4

An analysis of the behaviour of the free yarn loop during cone winding.

A4.1 Outline

This appendix provides the equations and computer coding which are necessary to predict the motion of an ideal positive compensator. The analysis assumes that the spinning machine is fitted with a curved distribution bar which maintains a constant free yarn length regardless of the position of the reciprocating yarn guide. This means that the only factors influencing the motion of the ideal positive compensator will be the guide velocity, and the variation in package peripheral speed due to the cone angle.

The appendix is divided into two major parts. The first describes how the time/displacement data for the yarn guide was processed to provide data files containing: (a) elapsed time at various values of guide displacement from the small end of the package, and (b) the velocity of the guide at various values of displacement. The second part describes the method by which these data files were used to calculate the rate of variation in size of the yarn loop in the compensator.

A4.2 Preparation of the data

As can be seen from equation 7.A3, the velocity with which the yarn passes through the guide depends upon geometric factors such as cone angle and drive roller radius, as well as the peripheral speed of the drive roller tyre and the velocity of the reciprocating yarn guide. Although it is generally thought that a guide moving with constant velocity would produce a package with the best (most consistent) density distribution, this is obviously not possible because of the infinitely high acceleration and inertia force at the guide reversal points. The barrel cam which moves the guide via a follower is therefore produced with a track profile which imparts controlled acceleration and deceleration upon the guide as it leaves and approaches the reversal points. In addition to this reversal profile, the cam track is of such a shape as to impart a reduction in guide velocity at that part of its traverse where it passes the drive tyre. This localised reduction in velocity causes proportionately more yarn to be deposited under
the tyre than would otherwise be the case, and this produces a harder less resilient structure for the tyre to press upon. The outcome of this is said to be that the tyre retains a good grip upon the package with the result that slipage is minimised. The cam track can therefore be considered to be composed of three major profile types. At the ends (reversal points), there is an acceleration/deceleration profile; under the tyre there is a velocity reduction profile; and over the remainder of the track the profile imparts a guide motion which is as close to constant velocity as can be achieved bearing in mind that each of the profiles must be interfaced in such a way as to produce no discontinuity to velocity or acceleration.

The cam track is therefore some way from the "ideal" profile which would result in a constant velocity guide motion. It would therefore be unwise to assume constant velocity guide motion in the solution of equation 7.A3.

Information on the exact profile of the cam track was obtained from Platt Saco Lowell in the form of the coordinates of the cutter used to machine the track. These coordinates were given as angle and displacement pairs, as defined in diagram A4-2(1)

![Diagram A4-2(1) Definition of guide cam track coordinates](image)

 Altogether there are 769 sets of coordinates describing the track profile. The following table A4-3(2) gives 77 of these sets (one in ten).
<table>
<thead>
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<th>Angle (degrees)</th>
<th>Displacement (mm)</th>
</tr>
</thead>
<tbody>
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</tr>
<tr>
<td>30.2</td>
<td>8.2</td>
</tr>
<tr>
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<td>11.3</td>
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<td>15.9</td>
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<tr>
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<td>52.5</td>
</tr>
<tr>
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<td>57.7</td>
</tr>
<tr>
<td>398.2</td>
<td>62.6</td>
</tr>
<tr>
<td>432.1</td>
<td>67.7</td>
</tr>
<tr>
<td>465.6</td>
<td>72.3</td>
</tr>
<tr>
<td>498.8</td>
<td>77.5</td>
</tr>
<tr>
<td>532.2</td>
<td>82.7</td>
</tr>
<tr>
<td>565.7</td>
<td>88.0</td>
</tr>
</tbody>
</table>

A4-3(2) Table of cam track coordinates (10% of 769 coordinate points).
The total angular rotation of the cam is 4680 degrees which corresponds to the 13 revolutions of the cam which take place before the follower returns to it's start position. Over this period, the guide makes two passes in each direction over the package surface. Graph A4-4(3) plots the above data, and is an accurate representation of the diagrammatic curve reproduced in C1-29(19).

A4-4(3) Graph of cam cutter (follower) against angle of rotation.

To enable calculations to be made of yarn winding velocity from equation 7.A3, the data in the previous table and above graph must be transformed. The equation requires the velocity of the yarn guide, the position of the guide, and the peripheral speed of the package under the guide. A program called "DATAPREP" was written to take the full set of guide track coordinate data from data files, and produce new data files containing guide position, velocity, and elapsed time, at each of the 769 intervals which the cam track coordinate data covers.

It was decided that "DATAPREP" should read the track coordinate data from data files called "NANGL.DATA" (which contains the angular coordinates of the cam), and "DISP.DATA" (which contains the displacement of the centre of the track from the end of the cam body). The first step in data transformation required of the "DATAPREP" program was to modify the displacement data so that it was measured from the extreme of motion of the guide rather than from the end of the cam body. As can be
seen from A4-3(2) this merely requires a constant value to be subtracted from each data element in "DISP.DATA". This value is rounded to 7.3 mm in table A4-3(2). After calculating the position of the guide relative to its extreme of motion, "DATA.PREP" places the values into a data file called "PSN.DATA" for use by another program called "YARVEL" which will be described in section A4.3.

To calculate the elapsed time and guide velocity, "DATA.PREP" needs to calculate the angular velocity of the cam. This of course varies on the current Masterspinner by ±2%, as a result of pattern breaking. The effect of pattern breaking has however been ignored in the writing of "DATA.PREP", and cam angular velocity has been assumed to be constant. This assumption has been adopted because of the simplification in program coding which results, and the argument which can be made that the effects of pattern breaking at ±2% are small. This is discussed further in chapter 5.

Cam rotational speed is directly linked to that of the package drive shaft so that a constant winding angle is maintained. The variable identifiers and mathematical symbols used in the program coding are used in the following analysis to avoid confusion when the program is being considered.

\[
\begin{align*}
\text{DRA} &= \text{tension draft (=} \text{peripheral speed of tyre} / \text{peripheral speed of delivery roller}) \\
\text{PDR} &= \text{peripheral speed of the delivery roller (m/min)} \\
\text{RD} &= \text{radius of the drive roller tyre (mm)} \\
\text{RROL} &= \text{ratio of drive roller rotational speed (r/min) / cam shaft rotational speed (r/min)} \\
\text{NDGC} &= \text{cam shaft rotational speed}
\end{align*}
\]

As can be seen:

\[
\text{peripheral speed of the tyre} = \text{DRA} \times \text{PDR} \quad \text{........ (1.A4)}
\]

and,

\[
\text{NDGC} = \text{drive roller rotational speed} / \text{RROL} \quad \text{.... (2.A4)}
\]

but,
peripheral speed of the tyre = drive roller rotational speed * 2 * π * RD

.................. (3.A4)

Therefore, substituting 1.A4 and 3.A4 into 2.A4 gives:

NDGC = DRA * PDR / 2 / π / RD / RROL .......... (4.A4)

Rationalising units:
converting PDR from m/min to mm/sec gives

NDGC = DRA * PDR * 1000 / 2 / π / RD / RROL / 60 .......... (5.A4)

where NDGC is now in r/sec.

Converting NDGC into degrees/sec by multiplying by 360 finally gives

NDGC = 360*DRA*PDR*1000/3.14159/RD/RROL/60/2 ...... (6.A4)

and this equation is used in "DATA.PREP"

Ratio RROL is set by a timing belt transmission, and is fixed at the value: RROL = 4 / 7

"DATA.PREP" calculates guide velocity by taking two adjacent values of guide displacement, finding the difference between them, and dividing this by the time which has elapsed between the guide being in the first of these positions and the second.

The elapsed time between two adjacent guide positions (at increments n and n-1 from the initial position at the extreme of movement) is calculated by the equation:

T(N) = T(N-1) + (PHI(N) - PHI(N-1)) / NDGC ...... (7.A4)

where PHI is an array within "DATAPREP" into which the data from NANGL.DATA is written. N is incremented through each of the 769 steps, during which, equation 7.A4 provides a value to array element T(N) which is the elapsed time appropriate to each step. During the first step T(N) is known to be zero so PHI(N-1) need only be accessed during the second step.
The velocity of the guide corresponding to each increment is calculated in "DATAPREP" be equation 8.A4.

\[ DL(N) = \frac{(X(N+1) - X(N))}{(T(N+1) - T(N))} \]  ...... (8.A4)

In equation 8.A4, DL is an array which receives the guide velocity corresponding to each increment N. Array T is as described for equation 7.A4, and array X contains the guide displacement data written to it from DISP.DATA.

Because the velocities in array DL are mid-point velocities associated with two guide incremental positions, they do not correspond exactly with the time calculated by equation 7.A4 and stored in array TN. To correct this small error, the time elapsed between each velocity element in array DL is calculated using equation 9.A4.

\[ T2(N) = \frac{(T(N) + T(N + 1))}{2} \]  ...................... (9.A4)

where T2 is an array containing the corrected elapsed time values.

"DATAPREP" concludes by creating four data files which are accessed by "YARVEL". These files are:

"PSN.DATA" which has previously been described on page A4 - 5
"TIME.DATA" to which is written the contents of array T
"VEL.DATA" to which is written the contents of array DL
"TIM2.DATA" to which is written the contents of array T2

The following flow chart outline the structure of "DATAPREP".
Prepare arrays to receive data from external data files and from internal calculations

Input:
- tension draft
- peripheral speed of delivery rollers

Calculate the rotational speed of the guide cam

Read guide cam rotation / displacement data from data files and place in arrays NANGL.DATA and DISP.DATA

Calculate for each data pair in NANGL and DISP the displacement of guide from the small end of the package, and the elapsed time

Calculate the velocity of the guide, and the elapsed time associated with each guide position

Open data files PSN TIME VEL and TIM2 to receive data ready for use by program YARVEL

Write data to the plotter and to the data files

A4.2.1 "DATAPREP coding"

REM THIS PROGRAM IS CALLED "DATAPREP"
REM IT PRODUCES DATA FILES CONTAINING VELOCITY AND TIME DATA FOR REM INPUT TO PROGRAM "YARVEL" FOR CALCULATION OF LOOP SIZE REM FIRST DIMENSION ARRAYS REM GUIDE DISP' ARRAY ASSUMING 769 DATA POINTS (MM) DIM X(1:769) REM GUIDE CAM ROTATION ASSUMING 769 DATA POINTS (DEGREES) DIM PHI(1:769) REM DIST' OF GUIDE FROM SMALL END OF PACKAGE (MM)
DIM Y(1:769)
REM TIME TO REACH ITERATION 'N' (SECONDS)
DIM T(1:769)
REM LINEAR VELOCITY OF YARN GUIDE (MM/SECOND)
DIM DL(1:769)
PRINT "TENSION DRAFT = ";
INPUT DRA
PRINT "PERIPHERAL SPEED OF DELIVERY ROLLER (M/MIN) = ";
INPUT PDR
REM RD=RADIUS OF DRIVE ROLLER TYRE (MM)
RD=76.4/2
REM RR= RATIO OF DRIVE ROLLER SPEED / CAM SHAFT SPEED
RR=4/7
REM NDGC= ROTATIONAL SPEED OF CAM
NDGC=360*DRA*PDR*1000/3.14159/RD/RR/60/2
REM OPEN CAM TRACK DATA FILES FOR INPUT
OPEN "DISP.DATA" FOR INPUT AS #1
OPEN "NANGL.DATA" FOR INPUT AS #2
FOR N=1 TO 769
REM INPUT THE X COORDINATE OF THE CAM TRACK POSITION
INPUT #1, X(N)
REM INPUT THE ANGULAR COORDINATE OF THE CAM TRACK POSITION
INPUT #2, PHI(N)
REM CALCULATE GUIDE POSITION FROM SMALL END OF PACKAGE
Y(N)=X(N) - 7.25
REM CALC ACCUMULATED TIME (T) ELAPSED AT EACH INCREMENT
IF N=1 THEN
T(N)=0
ELSE
T(N)=T(N-1) + (PHI(N) - PHI(N-1))/NDGC
END IF
NEXT N
CLOSE
FOR N=1 TO 768
REM CALC VELOCITY OF GUIDE AT EACH INCREMENT DL(N)
DL(N)=(X(N+1) - X(N)) / (T(N+1) - T(N))
REM CALC ELAPSED TIME (T2(N)) ASSOCIATED WITH EACH VELOCITY CALC
T2(N)=(T(N) + T(N+1)) / 2
NEXT N
DL(769)=DL(1)
REM OPEN FILES TO STORE GUIDE POSITION, TIME, AND GUIDE VELOCITY DATA
OPEN "PSN.DATA" FOR OUTPUT AS £3
OPEN "TIME.DATA" FOR OUTPUT AS £4
OPEN "VEL.DATA" FOR OUTPUT AS £5
OPEN "TIM2.DAT" FOR OUTPUT AS £6
REM PRINT HEADINGS
LPRINT "TENSION DRAFT = ";DRA
LPRINT "DELIVERY SPEED = ";PDR;"M/MIN"
LPRINT
LPRINT " DISP'T ANGLE GUIDE PS'N AT TIME VELOCITY AT TIME"
FOR N=1 TO 769
REM PRINT DATA UNDER HEADINGS
LPRINT USING " ££.££££AAAA"; X(N);PHI(N);Y(N);T(N);DL(N);T2(N);
LPRINT
REM WRITE DATA TO FILES
WRITE £3, Y(N)
WRITE £4, T(N)
WRITE £5, DL(N)
WRITE £6, T2(N)
NEXT N
CLOSE
END
A4.3 Calculation of yarn loop parameters

This section describes the objectives, structure, and mathematical basis of the program "YARVEL". The main objective of this program is to calculate the size of the yarn loop which will be required when the yarn guide is at each of the 769 known coordinate points of the cam track. This loop size is that which is required to keep the amount of yarn between the delivery rollers and the winding point constant, and therefore also maintain yarn tension at a constant level. The problem is better understood by reference to equation 7.3, which is reproduced below for convenience.

\[
\frac{\delta x}{\delta t} = \left\{ \left[ \frac{\delta L}{\delta t} \right]^2 + \left[ \frac{r}{\sin \left( \frac{\phi}{2} \right)} + y \right]^2 \left[ \frac{r_d}{\sin \left( \frac{\phi}{2} \right)} + \frac{M}{2} \right]^2 \frac{\delta \zeta}{\delta t} \right\}^{0.5}
\]

............... (7.3)

Although yarn passes through the delivery rollers at a constant speed, the velocity at which it is wound onto the package varies accordingly to equation 7.3. Therefore, to maintain the length of yarn between the delivery rollers and the package at a constant value, the compensator must cause its yarn loop to collect and release yarn with a velocity which is equal to the difference in the velocities of the yarn through the delivery rollers and onto the package surface.

The following mathematical development provides the basis for the numerical analysis of "YARVEL". As in the case for "DATAPREP" most of the variable symbols used in the following development are those used in the program coding rather than that of the general nomenclature defined at the beginning of this thesis.

Equation 7.3 requires input of the values of \( \frac{\delta \zeta}{\delta t} \), which is the angular velocity of the drive tyre, and \( r \) the radius of the small end of the package. The program therefore
requires input phases for these two parameters. The equation also requires the included angle of the package \( \Phi \), the radius of the drive tyre \( r_d \), and the width of the package \( M \). Because these values remain the same throughout the package build, they are set to constant values within the program.

Now,
\[
\frac{\delta r}{\delta t} = \frac{1000 \ V_p}{60. r_d} \quad \text{................................................................. (10.A4)}
\]

where \( V_p \) is the peripheral speed of the delivery rollers (i.e., yarn delivery velocity) in m/min, and \( \delta \zeta / \delta t \) is in radians / sec.

In "YARVEL", equation 10.A4 is coded as:

\[
DA = PDR * \frac{1000 \ V_p}{RD / 60}
\]

where
\[
\delta \zeta / \delta t = DA \\
V_p = PDR \\
RD = r_d
\]

After reading the data files created in "DATAPREP", "YARVEL" is in a position to calculate the winding velocity of the yarn via equation 7.A3 for each of the 769 coordinate points. In "YARVEL", equation 7.A3 appears as follows:

\[
.DXT(N)=((\text{VEL}(N)^2)+(((\text{RS}/\text{SIN(THETA/2/57.296)})+\text{PSN}(N))^2)*((\text{RD}/((\text{RS}/\text{SIN(THETA/2/57.296)})+(W/2))))^2)*((\text{DA}^2)))^0.5
\]

\[
\quad \text{.................. (11.A4)}
\]

where

- \( \text{DXT}(N) \) is an array containing each value of \( \delta \zeta / \delta t \)
- \( \text{VEL}(N) \) is an array of guide velocity (\( \delta L / \delta t \)) data read from a "DATAPREP" file
\[
\begin{align*}
\text{RS} &= r \\
\text{THETA} &= \Phi \text{ (converted to radians for trig' manipulation)} \\
\text{PSN}(N) &= \text{an array containing guide displacement (y) data from a} \\
& \quad \text{"DATAPREP" file} \\
W &= M
\end{align*}
\]

Having calculated the yarn winding velocity, "YARVEL" is able to calculate the change in size of the loop with respect to time (called "loop velocity" for brevity). The equation for this is:

\[
V_{\text{loop}} = V_P \frac{1000}{60} - \frac{\Delta x}{\Delta t} \quad \text{.............................................. (12.A4)}
\]

where \(\Delta x / \Delta t\) is calculated from 7.A3 (11.A4)

In "YARVEL", equation 12.A4 is coded as follows:

\[
\text{LOOPVEL}(N) = (\text{PDR}*1000/60) - \text{DXT}(N) \quad \text{......... (13.A4)}
\]

The calculation of loop velocity is important, not only because it provides the basis for the subsequent calculation of relative loop size and thereby compensator arm movement, but also because it clears the way for the calculation of loop acceleration and the analysis of inertia forces in the compensator.

The next phase of "YARVEL" was to calculate loop acceleration by numerical analysis. This was achieved by the following equation as it was coded into the program:

\[
\text{ACCN}(N) = (\text{LOOPVEL}(N+1) - \text{LOOPVEL}(N)) / (\text{TIM2}(N+1) - \text{TIM2}(N)) \quad \text{.... (14.A4)}
\]

where the arrays LOOPVEL contain 769 calculated values of loop velocity from 13.A3, and TIM2 contains elapsed time data from the data file prepared by "DATAPREP".

Finally, "YARVEL" calculates the change in length of the yarn in the loop which, for a compensator and guide roller layout as shown in A4-14(4), approximates to twice the
linear motion of the movable compensator roller.

Fixed guide rollers

Yarn loop

Moving compensator roller

A4-14(4) **Yarn loop configuration**

The change in length of the yarn loop is calculated in "YARVEL" by the following coded equation:

\[
YARNDISP(N) = YARNDISP(N - 1) + (LOOPVEL(N) \times (TIM2(N + 1) - TIM2(N)))
\]

\[ \ldots \ldots (15.A4) \]

where YARNDISP is an array containing the change in length of the loop which occurs at each of the 769 increments, and LOOPVEL and TIM2 have been previously defined in this section.

The following flow chart shows the structure of the "YARVEL" program.
Input
radius of small end of the package
peripheral speed of the delivery rollers

Set the package included angle and the radius of
the drive roller

Calculate the angular velocity of the drive roller

Open the data files created in program DATAPREP
so that they can be read

Dimension arrays so that the data from the data files
can be read to them

Transfer data from the data files to the arrays

Dimension array DXT to receive the data describing
the velocity of yarn demand

Calculate the velocity of yarn demand using
equation 7.A3

Open data file and write yarn demand velocity to it.

Prepare arrays to receive data on yarn loop acceleration,
velocity, and change in size

Calculate yarn loop velocity, acceleration, and change
in size, and write to appropriate arrays

Write data to the printer
A4.3.2 "YARVEL" coding

REM THIS PROGRAM IS CALLED "YARVEL"
REM INPUT THE PACKAGE SIZE AND DELIVERY ROLLER SPEED
PRINT "RADIUS OF SMALL END OF PACKAGE (MM) = ";
INPUT RS
LPRINT "RADIUS OF SMALL END OF PACKAGE = ";RS
PRINT "PERIPHERAL SPEED OF DELIVERY ROLLER (M/MIN) = ";
INPUT PDR
LPRINT "PERIPHERAL SPEED OF DELIVERY ROLLER = ";PDR
LPRINT
LPRINT
REM SET INCLUDED ANGLE OF PACKAGE
THETA=9
REM SET RADIUS OF DRIVE ROLLER TYRE
RD=76.4/2
SET WIDTH OF PACKAGE SURFACE
W=150
REM CALC ANGULAR VEL' OF DRIVE ROLLER TYRE
DA = PDR*1000 / RD / 60
REM DEFINE THE NUMBER OF COORDINATE PAIRS
NA = 769
REM OPEN THE DATA FILES FOR READING
OPEN "DISP.DAT" FOR INPUT AS £1
OPEN "NANGL.DAT" FOR INPUT AS £2
OPEN "PSN.DAT" FOR INPUT AS £3
OPEN "TIM2.DAT" FOR INPUT AS £4
OPEN "VEL.DAT" FOR INPUT AS £5
REM DIMENSION ARRAYS TO CONTAIN 769 ITEMS OF DATA
DIM DISP(1:NA)
DIM NANGL(1:NA)
DIM PSN(1:NA)
DIM TIM2(1:NA)
DIM VEL(1:NA)
REM TRANSFER DATA FROM THE DATA FILES TO THE ARRAYS
FOR N=1 TO NA
INPUT £1, DISP(N)
INPUT £2, NANGL(N)
INPUT £3, PSN(N)
INPUT £4, TIM2(N)
INPUT £5, VEL(N)
NEXT N
REM CLOSE ALL DATA FILES
CLOSE
REM DIMENSION ARRAY DXT READY TO RECEIVE YARN VELOCITY DATA
DIM DXT(1:NA)
REM CALCULATE VELOCITY OF YARN DEMAND FROM EQUATION 7.A3
FOR N=1 TO 769
DXT(N)=((VEL(N)^2)+(((RS/SIN(THETA/2/57.296))+PSN(N))^2)*((RD/((RS/SIN(THETA/2/57.296))+(W/2)))^2)*(DA^2))^0.5
NEXT N
REM OPEN DATA FILE TO RECEIVE YARN VELOCITY DATA FROM ARRAY DXT
OPEN "DXT.DATA" FOR OUTPUT AS £6
REM WRITE DATA TO FILE
FOR N=1 TO 769
WRITE £6, DXT(N)
NEXT N
REM CLOSE DATA FILE
CLOSE
REM PREPARE ARRAYS TO RECEIVE DATA ON LOOP ACCELERATION, LENGTH
REM OF YARN IN THE LOOP, AND LOOP VELOCITY
DIM ACCN(1:NA)
DIM YARNDISP(1:NA)
DIM LOOPVEL(1:NA)
REM CALCULATE THE VELOCITY OF THE LOOP
FOR N=1 TO 769
LOOPVEL(N) = (PDR*1000/60) - DXT(N)
NEXT N
REM CALCULATE THE ACCELERATION OF THE LOOP
FOR N=1 TO 768
ACCN(N) = (LOOPVEL(N+1) - LOOPVEL(N)) / (TIM2(N+1) - TIM2(N))
NEXT N
REM CALCULATE THE CHANGE IN LENGTH OF THE LOOP OVER THE FIRST REM INCREMENT
YARNDISP(1) = LOOPVEL(1) * (TIM2(2) - TIM2(1))
REM CALCULATE THE CHANGE IN LENGTH OF THE LOOP FOR ALL OTHER REM INCREMENTS
FOR N=2 TO 768
YARNDISP(N) = YARNDISP(N - 1) + (LOOPVEL(N) * (TIM2(N + 1) - TIM2(N)))
NEXT N
REM WRITE OUTPUT COLUMN HEADINGS ON THE PRINTER
LPRINT "GUIDE POS'N TIME YARN VEL LOOP VEL ACCELERATION YARN ERROR"
REM "YARN ERROR" IS THE CHANGE IN LOOP SIZE REQUIRED TO MAINTAIN REM CONSTANT TENSION
REM WRITE THE CONTENTS OF THE ARRAYS TO THE PRINTER
FOR N=1 TO 769
LPRINT USING "££.££££AAAA;Y(N);TIM2(N);LOOPVEL(N);ACCN(N);YARNDISP(N);LPRINTN NEXT N END

Chapter 5 discusses the output of from "YARVEL" and its relevance to the requirements of compensator motion.
Appendix 5

THE CONTROL OF YARN LOOP SIZE VARIATION USING ADJUSTABLE RUNNERS
Appendix 5.
The control of yarn loop size variation using adjustable yarn runners.

A5.1 Runner geometry.

Analysis of the mechanical positive compensator was based upon the assumption that the points (runners) above and below the compensator through which the yarn runs are fixed in space. This assumption is acceptable if the mechanism is capable of providing the necessary amplitude control by means of adjustment of the control crank. If this is not the case, then an alternative or additional way in which loop amplitude can be modified is the movement of the runners. Consider diagram A5-1(1).

![Diagram of yarn compensator and runners](image)

A5-1(1) Diagrammatic representation of compensator and runners

Diagram A5-1(1) shows the yarn travelling in the direction indicated by the arrows, through the first runner, around the compensator at, and through the second runner. The horizontal distance from the compensator to the runners is 'c', and the vertical distance between the runners is 'H'. The amplitude of the compensator is designated 'A'. If the diameter of the compensator roller is the same as 'H' then the length of the yarn loop (ignoring the wrap around the roller) is 2X. This will also be approximately correct if X is large in respect to the compensator roller diameter even in the case where H=0.
If the vertical separation between the runners is increased, then the length of yarn in one leg of the loop can be expressed by:

\[ b - a = \left\{ \left( A + d \right)^2 + c^2 \right\}^{0.5} - \left\{ d^2 + c^2 \right\}^{0.5} \] ........................ (1.A5)

The variables of 1.A5 are shown in diagram A5-2(2)

Therefore, assuming that the runners are equidistant above and below the horizontal centre line through the compensator:

\[ \text{Loop length } (L_A) = 2 \left( b - a \right) \]

therefore:

\[ L_A = 2 \left\{ \left[ \left( A + d \right)^2 + c^2 \right]^{0.5} - \left[ d^2 + c^2 \right]^{0.5} \right\} \] ........................ (2.A5)

In chapter 5 (section 5.2) the maximum and minimum yarn loop lengths predicted by "YARVEL" are 50mm and 14mm respectively. Setting \( A=25\text{mm} \), and \( d=100\text{mm} \) (a reasonable space in view of the available space) then equation 2.A5 yields the following loop size variations at different values of \( c \):
For $A=25\text{mm}$ and $d=0$ the equation yields the following values:

<table>
<thead>
<tr>
<th>$c$</th>
<th>1</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>40</th>
<th>50</th>
<th>60</th>
<th>80</th>
<th>100</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_A$</td>
<td>24</td>
<td>16.9</td>
<td>12.0</td>
<td>9.1</td>
<td>7.2</td>
<td>5.9</td>
<td>5.0</td>
<td>3.8</td>
<td>3.0</td>
</tr>
</tbody>
</table>

It is obvious that $d$ must be kept as small as possible if the necessary variation in loop size is to be obtained without excessive movement of the runners. Therefore, placing $d=0$ into 2.A5:

$$L_A = 2 \left[ \left( A^2 + c^2 \right)^{0.5} - c \right]$$  \hspace{1cm} (3.A5)

### A5.2 Compensator with a guide roller moving in a circular arc.

In the case of a compensator roller which is rotating in a circular motion rather than following a locus of the form followed by the output link of the 7-bar mechanism, the relationship between the roller and the runners will be as shown in A5-4(3).
A5-4(3) Diagrammatic representation of a compensator with a rotary motion guide roller and variable width runners.

From A5-4(3):

\[
X = X_c + x = X_c + r \cos \theta \\
Y = r \sin \theta \\
a^2 = X^2 + (c + Y)^2 \\
b^2 = X^2 + (c - Y)^2
\]

A computer program was written on the basis of the previous equations to explore the feasibility of using adjustable runners to provide yarn loop size variation on a compensator with a guide roller moving in a circular path. The output from this program is plotted in the following graphs.
A5-5(4)  Yarn loop amplitude against runner spacing for $X_c=25\text{mm}$ and $R=25\text{mm}$.

A5-5(5)  Yarn loop amplitude against runner spacing for $X_c=20$ and $R=20$
A5-6(6) Yarn loop amplitude against runner spacing for $X_c=22$ and $R=22$

The graphs show that with $X_c=22\text{mm}$ and $R=22\text{mm}$ a movement of each runner by 140 mm away from the mid-line of the compensator will provide the variation in yarn loop amplitude predicted as being necessary by "YARVEL".

The program used to provide the output plotted in the previous graphs was also used to provide plots of the variation in yarn loop size during one full rotation of the guide roller. This information is plotted on the following graph.
A5-7(7)  **Yarn loop size variation against angle of rotation of compensator roller**

As always, the key factor is the accuracy with which the curve for the smallest diameter package fits the ideal requirement. This corresponds to the curve for \(c=22\) mm in the above graph, and this is reproduced on its own below.

A5-7(8)  **Yarn loop size variation against angle of rotation of compensator roller for \(c=22\) mm (ie for the smallest package)**
The application of the circular arc compensator and the variable runner amplitude control are dealt with in chapter 6 where particular consideration is given to the effect of yarn loop size error on tension control.
Appendix 6

RELATIVE POSITION ANALYSIS OF A 7-BAR MECHANISM
Appendix 6

Relative position analysis of a 7-bar mechanism.

A6.1 Outline

The purpose of this analysis is to provide the mathematical basis for the parametric program developed in Appendix 7.

The mechanism described in Chapter 6 (section 6.1) consists of six moving links and a fixed link (or ground). To enable an iterative optimisation program to be written, it is necessary that the motion of the output link can be defined relative to the positions of the input and control cranks. The diagram presented below shows, and names, the links in the mechanism.

The purpose of the relative position analysis is (as the name implies) to develop a number of expressions, each of which describes the x,y coordinate position on one link in respect to the position of a point on an adjacent link. Eventually, by combining the expressions, the position of point f (on the output link) can be determined from a knowledge of the positions of the input and control cranks.
A6.2 Analysis

INPUT CRANK:

\[ \text{length of crank} = r_1 \]

From drawing A6-2(2):

\[ x_b = r_1 \cos(\theta) + x_1 \]  \hspace{1cm} (1.6A) \\
\[ y_b = r_1 \sin(\theta) + y_1 \]  \hspace{1cm} (2.6A)

where \( x_b, y_b, x_1, \) and \( y_1 \) are the coordinates of points b and 1 respectively.

CONTROL CRANK:

From diagram A6-2(3):

\[ x_d = r_2 \cos(\tau) + x_2 \]  \hspace{1cm} (3.6A) \\
\[ y_d = r_2 \sin(\tau) + y_2 \]  \hspace{1cm} (4.6A)

where \( x_d, y_d, x_2, \) and \( y_2 \) are the coordinates of points d and 2 respectively.
CONTROL LINK:

From diagram A6-3(4):

\[
\tan(\gamma) = \frac{x_b - x_d}{y_d - y_b} \tag{5.A6}
\]

so,

\[
\gamma = \tan^{-1} \left( \frac{x_b - x_d}{y_d - y_b} \right) \tag{6.A6}
\]

Also,

\[
s_1 = \left[ \left( x_b - x_d \right)^2 + \left( y_d - y_b \right)^2 \right]^{0.5} \tag{7.A6}
\]

and,

\[
\left( l_1 \right)^2 = \left( s_1 \right)^2 + \left( l_2 \right)^2 - 2s_1 l_2 \cos(\epsilon) \tag{8.A6}
\]
from (8.A6),

\[ \varepsilon = \cos^{-1}\left[ \frac{(s_1)^2 + (l_2)^2 - (l_1)^2}{2s_1l_2} \right] \]  \hspace{1cm} (9A6)

\[ \chi = \varepsilon + \gamma + \frac{\pi}{2} \]  \hspace{1cm} (10A6)

Also,

\[ y_a = l_2 \sin(\chi) + y_b \]  \hspace{1cm} (11A6)

\[ x_a = l_2 \cos(\chi) + x_b \]  \hspace{1cm} (12A6)

INPUT LINK:

From diagram A6-5(6):

\[ (l_4)^2 = (l_2)^2 + (l_3)^2 - 2(l_2)(l_3)\cos(\nu) \]  \hspace{1cm} (13A6)

From (12.A6),

\[ \nu = \cos^{-1}\left[ \frac{(l_2)^2 + (l_3)^2 - (l_4)^2}{2(l_2)(l_3)} \right] \]  \hspace{1cm} (14A6)
Also,
\[ \kappa = \chi + \upsilon \]  
(15A6)

and,
\[ y_c = l_3 \sin(\kappa) + y_b \]  
(16A6)

and
\[ x_c = l_3 \cos(\kappa) + x_b \]  
(17A6)

Where \( x_c, y_c \), are the coordinates of point c.

CONNECTING LINK:

\[ A6-5(7) \text{ Connecting link - output link geometry.} \]
From diagram A6-5(7):

\[ s_2 = \left[ (y_c - y_3)^2 + (x_c - x_3)^2 \right]^{0.5} \]  \hspace{2cm} (18A6)

and,

\[ \rho = \tan^{-1} \left[ \frac{x_c - x_3}{y_c - y_3} \right] \]  \hspace{2cm} (19A6)

and,

\[ (r_3)^2 = (s_2)^2 + (l_5)^2 - 2(s_2)(l_5)\cos(\eta) \]  \hspace{2cm} (20A6)

and,

\[ \eta = \cos^{-1} \left[ \frac{(s_2)^2 + (l_5)^2 - (r_3)^2}{2(s_2)(l_5)} \right] \]  \hspace{2cm} (21A6)

and,

\[ \lambda = \frac{3\pi}{2} - \eta - \rho \]  \hspace{2cm} (22A6)

The coordinates of point e are given by:

\[ x_e = x_c + l_5\cos(\lambda) \]  \hspace{2cm} (23A6)

and

\[ y_e = y_c + l_5\sin(\lambda) \]  \hspace{2cm} (24A6)

where \( \lambda \) is defined in equation (22.A6).
From diagram A6-7(8):

\[ \phi = \tan^{-1} \left[ \frac{y_e - y_3}{x_e - x_3} \right] \]  \hspace{1cm} (25.A6)

and,

\[ (l_9)^2 = (l_7)^2 + (r_3)^2 - 2(l_7)(r_3)\cos(\Gamma) \]  \hspace{1cm} (26.A6)

but,

\[ \Gamma = \cos^{-1} \left[ \frac{(l_7)^2 + (r_3)^2 - (l_6)^2}{2(l_7)(r_3)} \right] \]  \hspace{1cm} (27.A6)

Also,

\[ \zeta = \Gamma + \phi \]  \hspace{1cm} (28.A6)
and finally, the coordinates of the output point are given by:

\[
\begin{align*}
Y_1 &= y_3 + l_y \sin(\gamma) \quad \ldots \quad (29.A6) \\
X_1 &= x_3 + l_y \cos(\gamma) \quad \ldots \quad (30.A6)
\end{align*}
\]

A6.3 Conclusions on appendix 6

We now have a series of linked equations which yield the x,y coordinate of the point on the mechanism about which the yarn will pass. This point \((x_f, y_f)\) is expressed in terms of link lengths, fixed pivot coordinates, and the angular positions of the input and control cranks. Appendix 7 describes a parametric program on PAFEC DOGS* which uses these equations and draws the mechanism.

* Registered trade mark of PAFEC Ltd, Strelley, Nottingham.
Appendix 7

A PARAMETRIC SYMBOL FOR MECHANISM SYNTHESIS
Appendix 7
A parametric symbol for mechanism synthesis.

A7.1 Outline
The relative position analysis derived in appendix 6 forms the basis for this parametric symbol. The symbol was written for PAFEC DOGS version 3.2. The purpose of the symbol was twofold:

1. to enable the author to try various embodiments of the 7-bar compensator mechanism outlined in Chapter 6 and Appendix 6, and draw the resulting mechanism to see whether it would fit into the space available in the Masterspinner

2. to calculate the amount of yarn held by the compensator as the input crank rotates through 360°, so that this can be compared to the amount that the compensator would need to hold if the tension in the yarn were to remain constant throughout the full cycle of the reciprocating yarn guide

The appendix is divided into eleven major sections. Section A7.2 contains a flow chart describing the parametric symbol, and section A7.3 contains the coding for the symbol. Section A7.4 develops equations for the calculation of the amount of yarn held in the compensator, and section A7.5 develops a global coordinate system relative to a fixed point on the machine frame. Section A7.6 and A7.7 list the final parametric coding, whilst sections A7.8 and A7.9 develop programs to plot ideal and actual yarn loop variations to the screen. Section 10 describes a parametric to provide information on the difference between the actual and ideal yarn loop size, and section 11 outlines how the parametrics might be used.

A7.2 Program flow chart
All variables are as identified in Appendix 6.
A7.3 Symbol coding

This section contains the coding for the parametric symbol called "MECHANISM"

START/3,2

PROMPT ENTER X,Y COORDINATES OF INPUT ROTATION
STORE/C X1,Y1
PROMPT ENTER X,Y COORDINATES OF CONTROL ROTATION
STORE/C X2,Y2
PROMPT ENTER X,Y COORDINATES OF OUTPUT ROTATION
STORE/C X3,Y3
PROMPT TYPE RADIUS (ECCENTRICITY) OF INPUT CRANK (R1)
STORE/N R1
PROMPT TYPE RADIUS OF CONTROL CRANK (R2)
STORE/N R2
PROMPT TYPE LENGTH OF CONTROL LINK (L1)
STORE/N L1
PROMPT TYPE DIMENSIONS OF INPUT LINK (L2,L3,L4)
STORE/N L2,L3,L4
PROMPT TYPE LENGTH OF CONNECTING LINK (L5)
STORE/N L5
PROMPT TYPE DIMENSIONS OF OUTPUT LINK (R3,L6,L7)
STORE/N R3,L6,L7
PROMPT TYPE ANGULAR POSITION OF CONTROL CRANK
STORE/N T
REM SET THE ANGULAR POSITION OF THE INPUT CRANK TO ZERO
LET THETA=0
REM PROVIDE A LABEL TO JUMP TO AFTER THETA HAS BEEN INCREASED
*LABEL1
REM THIS IS THE START OF THE COORDINATE CALCULATION ROUTINE
REM CALCULATE THE COORDINATES OF POINT B
LET XB=(R1*COS(THETA))+X1
LET YB=(R1*SIN(THETA))+Y1
REM CALCULATE THE COORDINATES OF POINT D
LET XD=(R2*COS(T))+X2
LET YD=(R2*SIN(T))+Y2
REM CALCULATE ANGLE LOWER CASE GAMMA (SGAMMA)
LET SGAMMA=ARCTAN((XB-XD)/(YD-YB))
REM CALCULATE DISTANCE S1
LET S1=(XB-XD)**2+(YD-YB)**2**0.5
REM CALCULATE ANGLE EPSILON
LET EPSILON=ARCCOS(((S1)**2+L2**2-L1**2)/(2*S1*L2))
REM CALCULATE ANGLE CHI
LET CHI=90+EPSILON+SGAMMA
REM CALCULATE COORDINATES OF POINT A
LET YA=YB+(L2*SIN(CHI))
LETXA=XB+(L2*COS(CHI))
REM CALCULATE ANGLE U
LET U=ARCCOS(((L2)**2+(L3)**2-(L4)**2)/(2*L2*L3))
REM CALCULATE ANGLE KAPPA
LET KAPPA=CHI+U
REM CALCULATE COORDINATES OF POINT C
LETYC=YB+L3*SIN(KAPPA)
LETXC=XB+L3*COS(KAPPA)
REM CALCULATE DISTANCE S2
LET S2=((YC-Y3)**2+((XC-X3)**2))**0.5
REM CALCULATE ANGLE RHO
LET RHO=ARCTAN((XC-X3)/(YC-Y3))
REM CALCULATE ANGLE NEETA
LET NEETA=ARCCOS(((S2)**2+(L5)**2-(R3)**2)/(2*S2*L5))
REM CALCULATE ANGLE LAMBDA
LET LAMBDA=270-NEETA-RHO
REM CALCULATE COORDINATES OF POINT E
LET XE=XC+(L5*COS(LAMBDA))
LET YE=YC+(L5*SIN(LAMBDA))
REM CALCULATE ANGLE J (θ)
LET J=ARCTAN((YE-Y3)/(XE-X3))
REM CALCULATE ANGLE UPPER CASE GAMMA (Γ)
LET BGAMMA=ARCCOS(((L7**2)+(R3**2)-(L6**2))/(2*L7*R3))
REM CALCULATE ANGLE W (ω)
LET W=BGAMMA+J
REM CALCULATE COORDINATES OF POINT F
LET YF=Y3+(L7*SIN(W))
LET XF=X3+(L7*COS(W))
REM THIS IS THE END OF THE COORDINATE CALCULATION ROUTINE
REM THE NEXT ROUTINE DRAWS THE MECHANISM ON THE SCREEN
LN2
X=X1,Y=Y1
X=XB,Y=YB
LN2
X=X2,Y=Y2
X=XD,Y=YD
LN2
X=XD,Y=YD
X=XA,Y=YA
LN1
X=XA,Y=YA
X=XB,Y=YB
X=XC,Y=YC
X=XA,Y=YA
LN2
X=XC,Y=YC
X=XE,Y=YE
LN1
X=XE,Y=YE
X=X3,Y=Y3
X=XF,Y=YF
X=XE,Y=YE
REM THIS IS THE END OF THE ROUTINE TO DRAW THE MECHANISM TO THE SCREEN
REM THE NEXT ROUTINE INCREASES THETA, CALCULATES IF IT HAS REACHED 360°, AND IF IT
REM HAS NOT, RETURNS THE PROGRAM TO LABEL1
LET THETA=THETA+20
A7.4 The amount of yarn held in the compensator

The parametric program presented in section A7.3 provides a means of making repetitive drawings of a mechanism as its input crank rotates through $360^\circ$. This enables the envelope of movement to be checked to determine that it falls within the available space on the machine. However, this is only one aspect of the search for a suitable mechanism.

Of prime importance is the amount of yarn held in the compensator at every stage during its rotation. This amount must approximate the ideal amount calculated in Appendix 4 and discussed in Chapter 4. If, as is bound to be the case with a mechanism of the type under consideration, there is a difference between the ideal and actual amount of yarn in the compensator, the result will be a change in yarn tension. This change must be within acceptable limits if the mechanism is to be considered suitable.

There are several ways in which the mechanism described in Appendix 6 can be made to accumulate and release yarn during a guide cycle. However, the most obvious, and the one which seems to offer the simplest engineering solution is the one shown in the following diagram.
A7-6(1) **Geometry of the swinging arm compensator.**

In this compensator, the yarn passes first around fixed roller $d_3$, and then around the swinging roller $d_2$. The centre of roller $d_2$ corresponds to point $f$ on the output link of the 7-bar mechanism shown in section A6.1. The pivot for the link upon which roller $d_2$ is mounted is at $X_3,Y_3$, which again corresponds with the nomenclature used in Appendix 6. Finally, the yarn passes around fixed roller $d_1$ on its way to the package.

Having developed the program "MECHANISM" to draw the envelope of movement, we now need to develop an expression which will give the amount of yarn stored in the compensator for any given position of the swinging roller at point $f$. This expression is developed below.

**Calculation of tangential yarn length $(H+F)$ between rollers $d_1$ and $d_2$**

A7-6(2) **The relationship of yarn length $E+G$ to parameters $A$, $B$, and $C$.**
From diagram A7-6(2):

\[ E+G = \left[ (C-A)^2 + B^2 \right]^{0.5} \]  \hspace{1cm} (1A7)

From diagram A7-6(1):

\[ E^2 = \left( \frac{d_1}{2} \right)^2 + H^2 \]

and,

\[ G^2 = \left( \frac{d_2}{2} \right)^2 + F^2 \]

therefore:

\[ H = \left[ E^2 - \left( \frac{d_1}{2} \right)^2 \right]^{0.5} \] \hspace{1cm} (2A7)

and:

\[ F = \left[ G^2 - \left( \frac{d_2}{2} \right)^2 \right]^{0.5} \] \hspace{1cm} (3A7)

Now:

\[ \sin(\alpha) = \left( \frac{d_1}{2} \right) / E = \left( \frac{d_2}{2} \right) / G \]

and,

\[ E = \frac{d_1 G 2}{2 d_2} = \frac{G d_1}{d_2} \] \hspace{1cm} (4A7)
from (1.A7):
\[ G \frac{d_1}{d_2} + G = \left[ (C - A)^2 + B^2 \right]^{0.5} \]

hence,
\[ G \left( \frac{d_1}{d_2} + 1 \right) = \left[ (C - A)^2 + B^2 \right]^{0.5} \]

and,
\[ G = \left[ \frac{(C - A)^2 + B^2}{\left( \frac{d_1}{d_2} + 1 \right)^2} \right]^{0.5} \]

From (3.A7):
\[ F = \left[ \left( \frac{(C - A)^2 + B^2}{\left( \frac{d_1}{d_2} + 1 \right)} \right)^2 - \left( \frac{d_2}{2} \right)^2 \right]^{0.5} \]

From (4.A7):
\[ G = E \frac{d_2}{d_1} \]

hence,
\[ E \left( 1 + \frac{d_2}{d_1} \right) = \left[ (C - A)^2 + B^2 \right]^{0.5} \]
and,

$$\begin{align*}
E &= \left( \frac{(C - A)^2 + B^2}{1 + \frac{d_2}{d_1}} \right)^{0.5} \\
\text{From (2.A7):} \\
H &= \left[ \frac{(C - A)^2 + B^2}{\left(1 + \frac{d_2}{d_1}\right)^2} - \left( \frac{d_1}{2} \right)^2 \right]^{0.5} \\
\text{........................................................................ (6A7)}
\end{align*}$$

Adding (5.A7) and (6.A7):

$$\begin{align*}
H + F &= \left[ \frac{(C - A)^2 + B^2}{\left(1 + \frac{d_2}{d_1}\right)^2} - \left( \frac{d_1}{2} \right)^2 \right]^{0.5} + \left[ \frac{(C - A)^2 + B^2}{\left(\frac{d_1}{d_2} + 1\right)^2} - \left( \frac{d_2}{2} \right)^2 \right]^{0.5} \\
\text{........................................................................ (7A7)}
\end{align*}$$

**Calculation of length of lap around roller** ,

$$S = \tan^{-1} \left( \frac{C - A}{B} \right)$$
and, 

\[ \kappa = \sin^{-1} \left( \frac{\frac{d_1}{2}}{E} \right) = \sin^{-1} \left\{ \frac{\left(1 + \frac{d_2}{d_1}\right) \left(\frac{d_1}{2}\right)}{\left[(C - A)^2 + B^2\right]^{0.5}} \right\} \]

and, 

\[ \sigma = 3 - \kappa = \tan^{-1} \left( \frac{C - A}{B} \right) \sin^{-1} \left\{ \frac{\left(1 + \frac{d_2}{d_1}\right) \left(\frac{d_1}{2}\right)}{\left[(C - A)^2 + B^2\right]^{0.5}} \right\} \]

Therefore:

Angle of lap \( \gamma \) = 0.5 \( \pi - \sigma \) = \tan^{-1} \left( \frac{C - A}{B} \right) \sin^{-1} \left\{ \frac{\left(1 + \frac{d_2}{d_1}\right) \left(\frac{d_1}{2}\right)}{\left[(C - A)^2 + B^2\right]^{0.5}} \right\}

and, 

Length of yarn lap on roller 1 \( (s_1) \) = \( \frac{d_1}{2} \)  

**Calculation of tangential yarn length \((N + J)\) between rollers 2 and 3.**

A7-10(3) **The relationship of yarn length \( K + Q \) to parameters A and B.**
From diagram A7-10(3),

\[ K + Q = \left( A^2 + B^2 \right)^{0.5} \]  
\[ (10A7) \]

From diagram A7-6(1):

\[ K^2 = \left( \frac{d_3}{2} \right)^2 + N^2 \]

and:

\[ Q^2 = \left( \frac{d_2}{2} \right)^2 + J^2 \]

from which:

\[ N = \left[ K^2 - \left( \frac{d_3}{2} \right)^2 \right]^{0.5} \]  
\[ (11A7) \]

and:

\[ J = \left[ Q^2 - \left( \frac{d_2}{2} \right)^2 \right]^{0.5} \]  
\[ (12A7) \]

Now:

\[ \sin (\Theta) = \frac{\left( \frac{d_2}{2} \right)}{Q} = \frac{\left( \frac{d_3}{2} \right)}{K} \]

ie \[ Q = K \frac{d_2}{d_3} \]  
\[ (13A7) \]
from (8.A7):

\[ K \left( 1 + \frac{d_2}{d_3} \right) = \left( A^2 + B^2 \right)^{0.5} \]

\[ K = \frac{\left( A^2 + B^2 \right)^{0.5}}{\left( 1 + \frac{d_2}{d_3} \right)} \]

from (11.A7):

\[ N = \left[ \frac{\left( A^2 + B^2 \right)}{\left( 1 + \frac{d_2}{d_3} \right)} \right]^2 - \left( \frac{d_3}{2} \right)^2 \]^{0.5}

\[ \text{.................................................. (14A7)} \]

from (13.A7):

\[ K = Q \frac{d_3}{d_2} \]

from (10.A7):

\[ Q \left( \frac{d_3}{d_2} + 1 \right) = \left( A^2 + B^2 \right)^{0.5} \]

\[ Q = \frac{\left( A^2 + B^2 \right)^{0.5}}{\left( \frac{d_3}{d_2} + 1 \right)} \]
substituting into (12.47):

\[
J = \left[ \frac{(A^2 + B^2)}{\left( \frac{d_3}{d_2} + 1 \right)} \right]^2 - \left( \frac{d_2}{2} \right)^2 \right]^{0.5} \tag{15A7}
\]

adding (14.47) and (15.47):

\[
N + J = \left[ \frac{(A^2 + B^2)}{\left( 1 + \frac{d_2}{d_3} \right)} \right]^2 - \left( \frac{d_3}{2} \right)^2 + \left[ \frac{(A^2 + B^2)}{\left( \frac{d_3}{d_2} + 1 \right)} \right]^2 - \left( \frac{d_2}{2} \right)^2 \right]^{0.5} \tag{16A7}
\]

Calculation of length of lap around roller 3

From diagram A7-6(1),

\[
\nu = \tan^{-1} \left( \frac{A}{B} \right)
\]

and,

\[
\Theta = \sin^{-1} \left( \frac{\left( \frac{d_3}{2} \right)}{K} \right) = \sin^{-1} \left( \frac{\left( 1 + \frac{d_2}{d_3} \right) \left( \frac{d_3}{2} \right)}{\left( A^2 + B^2 \right)^{0.5}} \right)
\]
also,

\[ A = \sqrt{B^2 + (\frac{d_2}{d_3})^2} \cdot \sin^{-1}\left(\frac{1 + \frac{d_2}{d_3}}{(A^2 + B^2)^{0.5}}\right) \]

................................. (17A7)

But,

\[ \text{Angle of lap (} \mu \text{)} = \frac{\pi}{2} - A \]

and,

\[ \mu = 0.5 \pi - \tan^{-1}\left(\frac{A}{B}\right) + \sin^{-1}\left(\frac{1 + \frac{d_2}{d_3}}{(A^2 + B^2)^{0.5}}\right) \]

Therefore,

\[ \text{Length of lap on roller 3 (} s_3 \text{)} = \mu \cdot \frac{d_3}{2} \] ................................. (18A7)

**Calculation of length of lap around roller 2**

\[ \text{Angle of lap on roller 2} = \pi - (\sigma + A) \]

\[ \text{Length of lap on roller 2 (} s_2 \text{)} = \frac{d_2}{2} \left[\pi - (\sigma + A)\right] \] ................................. (19A7)

**Total length of yarn in the swinging arm compensator**

\[ \text{Accumulated yarn length (} A \text{)} = s_1 + (h+f) + s_2 + (n+j) + s_3 \] ..................(20A7)
A7.5 Calculation of global coordinates for position of compensator rollers.

Examination of diagram A7-6(1), and the ensuing calculations, will show that the datum for the measurement of coordinate points is the centre of the lower fixed roller $d_3$. However, in relation to the frame and major elements of the 4-position spin tester, upon which the mechanism is to be fitted, this datum is not fixed. ie the position of the lower fixed roller is variable, and may be positioned anywhere in an attempt to define an "optimum" mechanism. Before computer drawings can be made of proposed mechanisms, it will therefore be necessary to define the coordinates of key mechanism elements in general (and output point "f" in particular) with reference to some fixed point on the machine. For the purpose of this analysis, the coordinate system with it's origin at the centre of the lower roller is called the "local coordinate system", and the coordinate system with it's origin fixed to a point on the machine is called the "global coordinate system". The point chosen for the origin of the global system is the centre of the delivery roller shaft. This is a fairly arbitrary choice since any fixed point would suffice, however, the proximity of the roller to the pivot point of the output link makes it particularly suitable. The centre lines of the two fixed compensator rollers do not, of course, need to lie in a vertical plane. The local coordinate system, and the global system are, therefore, as shown on the following diagram.

**Local and global coordinate systems and their relationship to each other.**
Consider a global coordinate system with axes Y and X and an origin 'O' at the centre of the delivery shaft. Also consider a coordinate system local to the compensator with axes y and x and an origin 'A' at the centre of the lower compensator roller.

Compensator roller d1 will lie on the y axis. The origin of the x,y system is shifted from the origin of the X,Y system by xshift and yshift, and the x axis is rotated through angle $\theta_g$ anti clockwise measured relative to the global system.

A point with coordinates $X_f,Y_f$ relative to the global system, will have coordinates $x_f,y_f$ relative to the local system. For the case when $\theta_g = 0$, the transformations between the two systems are:

\[
x_f = X_f - x_{\text{shift}} \quad \text{(1A7)}
\]

\[
y_f = Y_f - y_{\text{shift}} \quad \text{(2A7)}
\]

For the unrotated local coordinate system shown below:

\[
r = \left( x_f^2 + y_f^2 \right)^{0.5} \quad \text{(3A7)}
\]

\[
r = \left[ (X_f - x_{\text{shift}})^2 + (Y_f - y_{\text{shift}})^2 \right]^{0.5} \quad \text{(4A7)}
\]

A7-16(5) Unrotated local coordinate system.
Now consider the rotated local coordinate system shown below:

\[ \begin{align*}
\beta_g &= \alpha_g - \theta_g = \tan^{-1} \left( \frac{(Y_f - y_{shift})}{(X_f - x_{shift})} \right) - \theta_g \quad \text{......... (5.A7)} \\
\cos \beta_g &= \frac{x_f}{r_g} ; \quad x_f = r_g \cos \beta_g \\
\sin \beta_g &= \frac{y_f}{r_g} ; \quad y_f = r_g \sin \beta_g
\end{align*} \]

Similarly for \( y_f \):

\[ \begin{align*}
X_f &= \left[ \left( X_f - x_{shift} \right)^2 + \left( Y_f - y_{shift} \right)^2 \right]^{0.5} \cdot \cos \left[ \tan^{-1} \left( \frac{(Y_f - y_{shift})}{(X_f - x_{shift})} \right) - \theta_g \right] \\
\text{......... (6.A7)}
\end{align*} \]
\[
y_f = \left[ (X_f - x_{\text{shift}})^2 + (Y_f - y_{\text{shift}})^2 \right]^{0.5} \cdot \sin \left[ \tan^{-1} \left( \frac{y_f - y_{\text{shift}}}{X_f - x_{\text{shift}}} \right) - \theta_g \right]
\]

\[\text{................. (7.}A7\text{)}\]

**A7.6 Modifications to the parametric program to include yarn length calculations.**

Having completed the analysis of sections A7.4 and A7.5, it remains to modify the parametric symbol of section A7.3 so that it incorporates these equations. The flow chart, presented below, outlines the basis of the coding modifications, which are listed in section A7.7.

1. **Define the diameters, and positions of rollers \(d_1\) and \(d_2\)**
2. **Calculate the position of all the links using "MECHANISM".**
3. **Calculate \(A, B,\) and \(C\) in terms of a global coordinate origin located at the centre of the delivery roller shaft.**
4. **Calculate the amount of yarn held in the compensator.**
5. **Write the amount of yarn in the compensator, and the associated value of \(\theta\) to a file called "YARN\(\text{DATA}".**
6. **Output to the screen the values of the important mechanism parameters.**
7. **Repeat calculations for next value of \(\theta\) unless \(\theta = 360^\circ\), in which case STOP.**
The coding of section A7.3 is here repeated together with the additions outlined in the previous section. Additions are highlighted in bold text.

```
START/3.2
REM OPEN FILE "YARN DATA" TO RECEIVE LENGTH INFORMATION
OPEN/W/YARNDATA/
REM DEFINE SIZE OF COMPENSATOR ROLLERS
LET D1=20
LET D2=20
LETD3=20
REM LOCATE THE POSITION OF THE CENTRE OF ROLLERS D1 AND D3 BY MEANS
OF CURSOR INPUT
PROMPT CURSOR LOCATE THE POSITION OF ROLLER 1
STORE/C XR1,YR1
PROMPT CURSOR LOCATE THE POSITION OF ROLLER 3
STORE XR3,YR3
REM DRAW ROLLERS 1 AND 3 ON THE SCREEN
LN4
X=XR1,Y=YR1
X=0,Y=D1/2
ACCEPT
X=XR3,Y=YR3
X=0,Y=D3/2
ACCEPT
REM THE FOLLOWING INPUT ROUTINE PROVIDES THE SYMBOL WITH MOST OF THE PARAMETER
VARIABLES NECESSARY FOR THE ANALYSIS AND DRAWING ROUTINES. REFER TO THE DRAWING
OF THE MECHANISM AT THE START OF SECTION A6.2, AND THE NOMENCLATURE FOR A
DEFINITION OF THE PARAMETERS USED.
PROMPT ENTER X,Y COORDINATES OF INPUT ROTATION
X=X3,Y=Y3
```
REM THIS IS THE END OF THE ROUTINE TO DRAW THE MECHANISM TO THE SCREEN
REM THE NEXT ROUTINE CALCULATES A, B, AND C IN TERMS OF GLOBAL
COORDINATES
LET C=((XR3-XR1)**2+(YR1-YR3)**2)**0.5
LET XSHIFT=XR3
LET YSHIFT=YR3
LET ROTN=ARCTAN((XR3-XR1)/(YR1-YR3))
LET AA=((XF-XSHIFT)**2+(YF-YSHIFT)**2)**0.5
LET AB=COS(ARCTAN((YF-YSHIFT)/(XF-XSHIFT))-ROTN)
LET B=AA*AB
LET AB=SIN(ARCTAN((YF-YSHIFT)/(XF-XSHIFT))-ROTN)
LET A=AA*AB
REM THE NEXT ROUTINE CALCULATES THE AMOUNT OF YARN HELD IN THE
COMPENSATOR
LET AA=ARCTAN(((C-A)/B)
LET AB=ARCSIN(((1+(D3/D1))*((D1/2))/(((C-A)**2)+(B**2))**0.5))
LET S1=(D1/2)*(90-AA+AB)*44/7360
LET AA=(((C-A)**2)+(B**2))/((1+(D3/D1)**2)-(D1/2)**2)**0.5
LET AB=(((C-A)**2)+(B**2))/((D1/D3)+1)**2)-(D3/2)**2)**0.5
LET HLEN=AA+AB
LET AA=ARCTAN(((C-A)/B)
LET AB=ARCSIN((1+(D3/D1))*(D1/2))/(((C-A)**2)+(B**2))**0.5))
LET AC=ARCTAN(A/B)
LET AD=ARCSIN((1+(D3/D2))*(D2/2)/(((A**2)+(B**2))**0.5))
LET S2=(D3/2)*(180-(AA-AB+AC-AD))*44/7360
LET AA=((A**2)+(B**2))/((1+(D3/D2)+1)**2)-(D2/2)**2)**0.5
LET AB=((A**2)+(B**2))/((1+(D2/D3)+1)**2)-(D3/2)**2)**0.5
LET NLEN=AA+AB
LET AA=ARCTAN(A/B)
LET AB=ARCSIN((1+(D3/D2)+(D2/2))/(((A**2)+(B**2))**0.5))
LET S3=(D2/2)*(90-AA+AB)*44/7360
REM THE NEXT ROUTINE HELPS TO DEBUG THE PROGRAM BY PRINTING THE
VALUES OF YARN LENGTH INTO THE DISPLAY SCRATCHPAD AREA
TP/S1=+/S1
TP/S2=+/S2
TP/S3=+/S3
TP/HLEN=+/HLEN
TP/NLEN=+/NLEN
REM FINALLY, CALCULATE THE TOTAL LENGTH OF YARN HELD IN THE
COMPENSATOR, AND PRINT IT TO THE SCRATCHPAD
LET LENGTH=S1+HLEN+S2+NLEN+S3
TP/LENGTH=/+LENGTH
TP/D1=/+D1
TP/D2=/+D2
TP/D3=/+D3
TP/A=/+A
TP/B=/+B
TP/C=/+C
REM WRITE THE YARN LENGTH AND CORRESPONDING INPUT CRANK ROTATION
(THETA) TO THE FILE CALLED "YARNDATA"
NWRITE LENGTH, THETA
REM THE NEXT ROUTINE INCREASES THETA, CALCULATES IF IT HAS REACHED 360°, AND IF IT
HAS NOT, RETURNS THE PROGRAM TO LABEL1
LET THETA=THETA+20
IF(THETA.GT.360)GOTO LABEL2
GOTO LABEL1
*LABEL2
REM CLOSE THE FILE CALLED "YARNDATA"
CLOSEIW
REM WRITE OUT A LIST OF THE IMPORTANT MECHANISM PARAMETERS BELOW
THE DRAWING ON THE DISPLAY
TE17
X=0,Y=-30
T=/X1=I+X1
X=0,Y=-40
T=/Y1=1+Y1
X=0,Y=-50
T=/X2=1+X2
X=0,Y=-50
T=/Y2=1+Y2
X=0,Y=-70
T=/X3=1+X3
X=0,Y=-80
T=/Y3=1+Y3
X=0,Y=-90
T=/R1=1+R1
X=0,Y=-100
T=/R2=1+R2
X=0,Y=-110
T=/L1=1+L1
X=0,Y=-120
T=/L2=1+L2
A7.8 Plotting the data.

The data representing the amount of yarn held in the compensator at particular angles of rotation of the mechanism input crank were held in the file called "YARNDATA". However, the effectiveness of a particular form of the mechanism is not measured by the absolute stored yarn length, but by the variation over one cycle. The data therefore needed to be normalised before it could be compared to the ideal, i.e. the lowest point on the curve needed to correspond to zero yarn accumulation. This normalisation, and the plotting of the "yarndata" file to the screen was achieved by a simple DOGS parametric called "GRAPH1". The following flow chart outlines the the structure of the Graph1 parametric.
SET INITIAL VALUES FOR MINIMUM AND MAXIMUM YARN LENGTH IN THE COMPENSATOR

CHECK THE DATA IN "YARN DATA" AND FIND THE MINIMUM YARN LENGTH STORED DURING ONE CYCLE OF THE MECHANISM, AND RECORD THE CORRESPONDING ANGLE OF MECHANISM INPUT CRANK ROTATION.

FIND THE MAXIMUM YARN LENGTH STORED DURING A CYCLE AND THE ASSOCIATED VALUE OF ROTATION.

CALCULATE THE AMPLITUDE OF THE YARN LENGTH / ROTATION CURVE.

PROGRAM USER DEFINES THE POSITION OF THE GRAPH ORIGIN BY SCREEN CURSOR.

DRAW THE GRAPH FRAME

"NORMALISE" THE DATA SO THAT THE LOWEST POINT ON THE CURVE SITS ON THE X-AXIS

SCALE THE DATA SO THAT IT FITS WITHIN THE HORIZONTAL FRAME SIZE

PLOT A GRAPH OF THE DATA WITHIN THE FRAME.

The parametric coding (called "GRAPH1") based upon the above flow diagram is listed below:

START 3.2
REM CALC MAX AND MIN LENGTHS AND CORRESPONDING THETA
LET MAXLEN=0
LET MINLEN=1000
OPEN/R/YARNDATA/
*LABEL2
NREAD LENGTH, THETA
IF(LENGTH.GE.MINLEN)GOTO LABEL1
LET MINLEN=LENGTH
LET MINTHETA=THETA
*LABEL1
LET N=N+1
IF(THETA.LT.359.5)GOTO LABEL2
CLOSE/R
OPEN/R/YARNDATA/
*LABEL3
NREAD LENGTH, THETA
IF(LENGTH.LE.MAXLEN)GOTO LABEL4
LET MAXLEN=LENGTH
LET MAXTHETA=THETA
*LABEL4
IF (THETA.LT.360)GOTO LABEL3
CLOSE/R
REM CALCULATE HEIGHT OF CURVE
LET HEIGHT=MAXLEN - MINLEN
REM REQUEST THAT POSITION OF GRAPH ORIGIN IS DEFINED
PROMPT CURSOR ENTER THE GRAPH ORIGIN
STORE/C XORIGIN, YORIGIN
REM MOVE USER ORIGIN TO GRAPH ORIGIN
FA2
X=XORIGIN, Y=JORIGIN
REM DRAW GRAPH FRAME
LT15
LN1
X=0, Y=0
X=200, Y=0
X=0, Y=100
X=-200, Y=0
X=0, Y=-100
REM SET SCALING FACTORS FOR DATA
LET XSCALE=200/360
REM DRAW GRAPH
OPEN/R/YARNDATA/
LET N=1
*LABEL6
NREAD LENGTH, THETA
IF(N.EQ.1)GOTO LABEL5
LET Y2=LENGTH - MINLEN
LET X2=THETA*XSCALE
LN2
X=X1, Y=Y1
X=X2 - X1, Y=Y2 - Y1
LET X1=X2
LET Y1=Y2
IF(THETA.LT.359)GOTO LABEL6
GOTO LABEL7
*LABEL5
LET Y1=LENGTH - MINLEN
LET X1=THETA'XSCALE
LET N=N+1
GOTO LABEL6
*LABEL7
CLOSE/R
END

A7.9 Plotting the ideal compensator curve.

The parametrics "MECHANISM" and "GRAPH1" draw the mechanism and plot the variation of accumulated yarn with respect to time. To enable a rapid assessment to be made of the suitability of the mechanism based upon it's ability to accumulate yarn in a manner which reflects the actual need of the machine, the following parametric was written to provide a plot of the ideal variation of accumulated yarn with respect to time.

The ideal data was provided by the program "YARVEL" described in detail in appendix 4, and was written to a data file (called "IDEAL") on the Apollo system. Once the IDEAL data file was established, the parametric outlined in the following flow diagram was used to provide a screen plot.
DRAW THE X-AXIS OF THE GRAPH WITH THE ORIGIN DEFINED BY CURSOR INPUT

FIND THE TIME PERIOD COVERED IN IDEAL

CALCULATE SCALING FACTORS

FIND THE MAXIMUM LENGTH STORED DURING A CYCLE OF THE MECHANISM

READ DATA AND PLOT THE GRAPH

The parametric based upon the above flow diagram is called "GRAPH2" and the coding is listed below:

START/3.2
OPEN/R/IDEAL/
PROMPT CURSOR POSITION GRAPH ORIGIN
STORE/C XORIGIN, YORIGIN
REM MOVE USER ORIGIN TO GRAPH ORIGIN
FA2
X=XORIGIN, Y=YORIGIN
REM DRAW GRAPH FRAME
LT14
LN2
X=0, Y=0
X=400, Y=0
REM FIND WHAT THE MAXIMUM TIME VALUE IS IN IDEAL SO THAT IT CAN BE USED TO
REM CALCULATE THE HORIZONTAL SCALE
*LABEL1
NREAD LENGTH, TIME
IF(LENGTH.GT.999)GOTO LABEL2
LET MAXTIME=TIME
GOTO LABEL1
*LABEL2
TP/MAXTIME=/+MAXTIME
LET XSCALE=400/MAXTIME
TP/XSCALE=/+XSCALE
CLOSE/R
LET N=0
REM READ ALL THE DATA AND FIND THE MAXIMUM LENGTH
OPEN/R/IDEAL/

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LET MINLEN=1000
*LABEL3
NREAD LENGTH, TIME
LET N=N+1
IF(LENGTH.GT.MINLEN)GOTO LABEL4
LET MINLEN=LENGTH
*LABEL4
IF(LENGTH.LT.999)GOTO LABEL3
CLOSE/R
N=N+1
REM READ THE DATA AND PLOT IT ON THE GRAPH
LET C=0
OPEN/R/IDEAL/ LN1
LET X2=0
LET Y2=0
TP/XSCALE=+/XSCALE
*LABEL5
NREAD LENGTH, TIME
IF(TIME.GT.999)GOTO LABEL6
TP/LENGTH=+/LENGTH
TP/TIME=+/TIME
LET XPOSN=(TIME*XSCALE) - X2
LET YPOSN=(LENGTH - MINLEN) - Y2
LET X=XPOSN
LET Y=YPOSN
LET C=C+1
LET XPRINT=TIME*XSCALE
TP/X=+/XPRINT
LET YPRINT=LENGTH - MINLEN
TP/Y=+/YPRINT
GOTO LABEL5
CLOSE/R
TP/XSCALE=+/XSCALE
TP/MINLEN=+/MINLEN
*LABEL6
END
A7.10 Determining the difference between the actual and ideal curves.

Following the graphical representation of the mechanism using the "MECHANISM" parametric, "GRAPH1" plots the curve of the actual amount of yarn accumulated against time, and "GRAPH2" plots the curve of the ideal amount of yarn accumulated against time. To enable direct comparisons to be made between these two curves, a fourth parametric called "ERROR" was written.

The purpose of "ERROR" was to enable the author to superimpose the actual and ideal graphs of accumulated yarn lengths, and to plot the difference between the two curves by manually selecting the points on the ideal curve which were of interest. A detailed description of the way in which these parametrics were used is provided in section A7.11. A flow diagram for the "ERROR" parametric is given on the following page.
Before starting the parametric, move the user origin to the origin of the actual/ideal graph.

OPEN FILE "ERRODAT" TO RECEIVE ERROR DATA IN PREPARATION FOR PLOTTING AN ERROR GRAPH

DRAW THE "ERROR" GRAPH FRAME BELOW THE EXISTING ACTUAL/IDEAL GRAPH

WINDOW IN SO THAT THE GRAPHS FILL THE SCREEN

ASK THE USER TO WINDOW IN TO AN AREA OF INTEREST ON THE ACTUAL/IDEAL GRAPH, AND DRAW THAT AREA TO FILL THE SCREEN

USER RECORDS A POINT OF INTEREST ON THE "IDEAL" CURVE USING A CURSOR HIT

SYSTEM DRAWS A VERTICAL LINE THROUGH THE DEFINED POINT, AND THE USER PICKS THE ASSOCIATED POINT ON THE "ACTUAL" GRAPH

SYSTEM CALCULATES ERROR BETWEEN ACTUAL AND IDEAL POINTS, STORES INFORMATION IN ERRODAT AND SUBSEQUENTLY DRAWS THE ERROR GRAPH
The coding for the ERROR parametric is listed below:

```
START / 3.2
OPEN/ERRORDAT
LET N=0
LN2
X=0, Y=-150
X=400, Y=0
LN2
X=0, Y=-50
X=0, Y=-250
LN1
X=0, Y=-250
X=400, Y=0
X=0, Y=200
X=-400, Y=0
REM WINDOW INTO ACTUAL / IDEAL GRAPH
*LABEL2
VS9
X=-20, Y=-270
X=420, Y=120
REM WINDOW THE APPROPRIATE AREA
LET N=N+1
STORE/C XWIN, YWIN
STORE/C XWIN2, YWIN2
VS9
X=XWIN, Y=YWIN
X=XWIN2, Y=YWIN2
PROMPT IDENTIFY A POINT ON THE IDEAL LINE
STORE/C XIDEAL, YIDEAL
LT12
LN2
X=XIDEAL, Y=-250
X=0, Y=200
IF(XIDEAL.LT.0)GOTO LABEL1
PROMPT IDENTIFY CORRESPONDING POINT ON ACTUAL LINE
STORE/C XACTUAL, YACTUAL
LET YERROR=YIDEAL - YACTUAL
NWRITE XIDEAL, YERROR
GOTO LABEL2
*LABEL1
```
A7.11 Using the parametrics to design a mechanical compensator.

The use of the parametrics described in this appendix are illustrated in this section by reference to a series of photographs taken from the screen of an Apollo DN3000 which was used to run the DOGS program. Their application is described in chapter 5.

The first photograph (A7-32(7)), shows the screen output from "MECHANISM" overlaid onto a sectional representation of existing components in the Masterspinner. Although it was appreciated that the mechanism would be incorporated into a completely re-designed machine, it was considered useful to use the space constraints of the existing machine as a reasonable bound to the envelope of motion of the proposed positive compensator motion. The colour coding of the various components are as listed below:

- white (solid line) - existing Masterspinner components
- red (chain dot) - existing Masterspinner steel panels which are easily removed or pierced
and eccentric. However, as previously stated, these panels are easily removed and such a violation does not prevent the fitting of such a mechanism to the machine.

A7-33(8) Typical envelope from the "MECHANISM" parametric

Finally, the next photograph (A7-34(9)) shows (for the particular mechanism shown in the top left hand corner) the actual variation in accumulated yarn drawn from parametric "GRAPH1" in yellow, and the ideal variation drawn from parametric "GRAPH2" in magenta (pink). These two graphs are then shown combined, and the lower graph (dark blue) drawn from parametric "ERROR" shows the difference between the two previous curves. A good mechanism will obviously show only small perturbations on the "ERROR" graph throughout the package build.
A7-34(9) Screen output from the GRAPH and ERROR parametrics

A7.12 Conclusions on appendix 7

The parametric programs developed in this appendix provide a powerful iterative tool for the design of the 7-bar compensator mechanism. A description of the use made of the programs, and the conclusions which were drawn as a result, are presented in chapter 6.
Appendix 8

ANALYSIS OF DISTRIBUTION BAR SHAPE
Appendix 8
Analysis of distribution bar shape.

A8.1 Outline
As shown in appendix 2, the distribution bar contributes significantly to the tension variations which occur in the yarn as it is wound onto a package. In the case of machines which are constructing cylindrical packages (or cheeses), manufacturers often make the distribution bar curved so that the yarn path length remains approximately constant. When winding conical packages, some machine manufacturers, including Platt Saco Lowell fit a straight distribution bar. This is because machines spinning onto conical packages are invariably fitted with either mechanical or passive compensators, and these devices are usually considered adequate to deal with yarn demand variations from both package peripheral speed variations and yarn path length variations. However, appendix 2 shows that the effect of yarn path length variations can be considerable, and this can impose a substantially increased demand on the work required of the compensator.

This appendix analyses the shape of a curved distribution bar which will remove yarn path length variations, and thereby reduce the work required by the compensator. The appendix also analyses the errors in yarn path length (and hence tension variation) as a result of using a distribution bar of semi circular profile rather than the exact profile.

A8.2 Analysis of bar profile
This analysis will refer to the diagram shown on the next page(A8-2(1)), which represents the yarn travelling from point pq where it leaves the compensator, to point rq where it first makes contact with the package. The curved distribution bar is in the plane ABCD.
A8-2(1) Machine parameters affecting distribution bar geometry.

In the above diagram:

- **oq-oq** is the line along which the drive roller contacts the package (known as the "nip line"). The yarn also winds onto the package along this line.
- **rq** This point is called the "winding point", and corresponds to the position of the reciprocating yarn guide.
- **xq1** is the distance of the winding point from the mid point of yarn guide travel.
- **W** is the distance from the plane of the distribution bar to the nip line, measured normal to the plane.
- **pq** is a fixed point through which the yarn passes prior to passing over the distribution bar.
- **Q** is the distance of point pq from the plane of the distribution bar, measured normal to the plane.
- **ABCD** is the plane in which the distribution bar lies.
mq is the point on the distribution bar over which the yarn passes on its way from pq to rq when the guide is at point rq.

xq and yq are the coordinates of point mq.

L_q is the length of the yarn path between points pq and rq via mq.

The other parameters are described in the following analysis.

For a given guide displacement xq1, and a given yarn path length L_q, the possible positions of mq on the plane ABCD describe an "ellipse" if the yarn is kept taught. The major axis of this ellipse will lie along the line lq - nq.

A8-3(2) Locus of possible points of penetration of yarn through the plane of the distribution bar.

Assumption: the friction coefficient between the yarn and the distribution bar is zero. This simplifies the analysis by removing the necessity to calculate the "lag" in the yarn - bar contact point due to friction. This simplification is justified on the basis that the friction coefficient is low, and lag is small.
From diagram on page A8-(2)1:

\[ J^2 = x_q^2 + y_q^2 \]

and,

\[ l_q^2 = J^2 + Q^2 \]

therefore:

\[ l_q^2 = x_q^2 + y_q^2 + Q^2 \] ................................................ (1.A8)

Also from diagram A8-2(1):

\[ k = \left( H^2 + x_q^2 \right)^{0.5} - \left( x_q^2 + y_q^2 \right)^{0.5} \] .......................... (2.A8)

and,

\[ l_q^2 = W^2 + k^2 \]

therefore:

\[ l_q^2 = W^2 + \left[ \left( H^2 + x_q^2 \right)^{0.5} - \left( x_q^2 + y_q^2 \right)^{0.5} \right]^2 \] .... (3.A8)

If \( L_q \) is the yarn path length between \( pq \) and \( rq \), then:

\[ L_q = l_q1 + l_q2 \] ................................................................. (4.A8)

and, from 1.A8 and 3.A8:

\[ L_q = \left( x_q^2 + y_q^2 + Q^2 \right)^{0.5} + \left\{ W^2 + \left[ \left( H^2 + x_q^1 \right)^{0.5} - \left( x_q^2 + y_q^2 \right)^{0.5} \right]^2 \right\}^{0.5} \] .......................... (5.A8)

From similar triangles, and using diagram A8-3(2):

\[ \tan(\beta_q) = \frac{y_q}{x_q} = \frac{H}{x_q^1} \]

therefore:

\[ y_q = \frac{H x_q}{x_q^1} \] ................................................................. (6.A8)
A8.3 A computer model to calculate the distribution bar profile.
Based upon the equations of section A8.2, the flowchart shown below describes a computer program written in Turbo BASIC for an IBM computer, to calculate coordinate points for a curved distribution bar which will provide a constant yarn path length.

Enter input variables: 
H, W, and P

Set the start value of xq1 as zero, ie the guide starts in the centre of the package.

Set the yarn path length (L0) to a large number

Set the value of xq to zero

Calculate path length (L) from equations 1.A8, 2.A8, 3.A8, and 4.A8

xq = xq + 0.01

Set L0 = L

Reset xq = xq + 0.01

Yes

L>L0

No

xq1 = xq1 + 5

Output xq, xq1, and L

A8.4 Computer coding
The following is the computer coding based upon the above flow chart. Programming and execution were carried out on a Sinclair QL.

100 H=100
110 W=160
120 P=17.32
130 XQ=0
140 L0=10000
150 FOR XQ1=0 TO 80 STEP 5

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160 A=(XQ**2+H**2+P**2)**0.5
170 B=(H**2+XQ1**2)**0.5 - (XQ**2+H**2)**0.5
180 C=(W**2+B**2)**0.5
190 L=A+C
200 IF L>L0 THEN 240 ELSE 210
210 L0=L
220 XQ=XQ+0.01
230 GOTO 160
240 PRINT "XQ1=","XQ1;" XQ=";XQ;" L=";L
250 NEXT XQ1

A8.5 Results from the computer program of section A8.4
The following table lists the results of running the program in the previous section. For each value of xq1 (ie the distance of the yarn guide from the centre of the package), the corresponding value of xq is given. Also, in the third column, is given the corresponding value of yq which has been calculated by hand from equation 6.A8.

<table>
<thead>
<tr>
<th>xq1</th>
<th>xq</th>
<th>yq</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>4.994</td>
<td>99.88</td>
</tr>
<tr>
<td>10</td>
<td>9.951</td>
<td>99.51</td>
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<tr>
<td>15</td>
<td>14.834</td>
<td>98.893</td>
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<td>20</td>
<td>19.61</td>
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<td>37.066</td>
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<td>56.317</td>
<td>80.452</td>
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<tr>
<td>75</td>
<td>58.586</td>
<td>78.114</td>
</tr>
<tr>
<td>80</td>
<td>60.571</td>
<td>75.713</td>
</tr>
</tbody>
</table>

These results are based upon values of H, P, and W which are obtained when a curved bar is bolted directly onto the front of the steel channel containing the guide cams and shaft. However, the value of H chosen for the analysis would not allow the fitting of the current torsion spring passive compensator. In fact, the value of H was set at 100 mm so that the curved distribution bar could be used with the pneumatic compensator.
A8.6 Errors incurred if a distribution bar of circular profile is used

With $H=100$, $W=160$, and $Q=17.32$, the calculated yarn path length from equation 5.A8 is 261.488 mm, and this will be constant for all values of $x_1$ if a curved distribution bar is fitted with the $x_q,y_q$ profile defined in section A8.5. Several manufacturers of open-end spinning machinery fit a distribution bar which follows the arc of a circle. The advantage of this is that the former over which the bar is bent, is easily machined in comparison to one which carries the optimum profile. However, it is interesting to note the errors which occur if a bar of circular profile is fitted, and the following section explores this effect.

The diagram shown below represents the circular arc of which the distribution bar is a part. The yarn is represented by the line 'r' which, in this view, is a radius of the circle. To keep similarity with the profiled bar calculated in the previous sections, the arc would have a radius of 100 mm. The parameters $x$, and $y$ would be the coordinates of the point where the yarn passed over the bar.

\[
y = (r^2 - x^2)^{0.5}
\]

Therefore:

\[
y = (r^2 - x^2)^{0.5}
\]

Calculating $L_q$ for a bar of circular profile:

\[
L_q = \left( x^2 + y^2 + p^2 \right)^{0.5} + \left\{ W^2 + \left[ \left( H^2 + xq_1^2 \right)^{0.5} - \left( x^2 + y^2 \right)^{0.5} \right]^2 \right\}^{0.5}
\]

Parameters affecting yarn length with a distribution bar of circular arc profile.
The bottom line in the table shows the errors in yarn path length (in mm) which occur when a distribution bar of circular profile is used instead of a properly profiled bar. The yarn path length with a properly profiled bar (with H, P, and W set as previously) is 261.488 mm. As can be seen, when the yarn guide is at a position 75mm to either side of centre, the circular profile gives a yarn path error of 1.95 mm. For a yarn with a stiffness of 8 gram/mm, the resulting tension change is 15.6 gram. Graph A8-8(5), shows the increase in tension which occurs at different values of $xq1$ when a circular profile distribution bar is fitted rather than one with a properly calculated profile.

The author acknowledges the valuable contribution of his research assistant Mr. J. Okello in the analysis and subsequent testing of the curved distribution bar.
Appendix 9

YARN HELD BY PASSIVE BOLLARD COMPENSATOR
Appendix 9.
Yarn held by the passive bollard compensator.

A9.1 Outline

The motion of the bollard compensator is of basic importance to the work described in this thesis. In particular, an understanding of the nature of the motion is important for the following three reasons:

(1) on the current Masterspinner, the bollard compensator is not satisfactory at high speed, and an analysis of the motion would help to understand why this is the case

(2) appendix 8 develops an analysis which facilitates the design of a curved distribution bar which removes the influence of "free yarn length" changes on yarn tension. The effect of this modified bar on bollard behavior is of obvious interest

(3) removing the spring from the bollard compensator, and converting it (the compensator) into a positive device by fitting a motor, is one option for the solution of the compensator problem. Defining the relationship between the rotation of the compensator front plate and the length of yarn stored, is a basic requirement for determining the motion of the motor.

Because items (2) and (3) assume that the yarn leaves the top of the compensator via a fixed point (point 8 in diagram A9-2(1)), the following analysis makes the assumption that this is, in fact, the case.

A9.2 Analysis

Diagram A9-2(1) represents the passive bollard compensator currently fitted to Platt Saco Lowell machines.
The yarn passes through fixed point 7, and then around the two pins (each of radius $R_b$) of the compensator before continuing through fixed point 8. The coordinates of the upper pin centre relative to the centre of rotation at point 4 is given by:

$$\sin(\pi - \theta_b) = \frac{d}{\left(\frac{c}{2}\right)}$$

hence;

$$d = \frac{c}{2} \sin(\pi - \theta_b) \quad \text{(1.A9)}$$

similarly;

$$e = \frac{c}{2} \cos(\pi - \theta_b) \quad \text{(2.A9)}$$
Length of yarn between points 3 and 6

If the length of yarn between 3 and 6 is \( l_b \), then:

\[
\frac{l_b}{2} = \left[ \left( \frac{c}{2} \right)^2 - \left( R_b \right)^2 \right]^{0.5}
\]

and:

\[
l_b = 2 \left[ \left( \frac{c}{2} \right)^2 - \left( R_b \right)^2 \right]^{0.5}
\] ................................. (3.A9)

Angle of lap \( \xi_b \)

Distance 1 to 3 = \( R_b \) (= the radius of the compensator pin).

A9-3(2) Geometry of yarn path

\[
\sin \alpha_b = \frac{R_b}{\left( \frac{c}{2} \right)} = \frac{2R_b}{c}
\]

\[
\alpha_b = \sin^{-1} \left( \frac{2R_b}{c} \right)
\] ................................. (4.A9)
\[ \varepsilon_b = \gamma_b - (\pi - \theta_b) \]  
\[ \zeta_b = \frac{\pi}{2} - \varepsilon_b \]

Combining equations (5.A9), and (6.A9) gives:

\[ \zeta_b = \frac{\pi}{2} - \left[ \gamma_b - (\pi - \theta_b) \right] \]

Therefore from 4.A9;

\[ \zeta_b = \pi \cdot \theta_b + \sin^{-1}\left(\frac{2R_b}{c}\right) \]

Angle of lap \( \eta_b \)

---

A9-4(3) Geometry of yarn path

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\[
\tan \phi_b = \frac{e}{a + d}
\]

and hence;

\[
\phi_b = \tan^{-1} \left( \frac{e}{a + d} \right) \tag{8.A9}
\]

\[
f = \left[ e^2 + (a + d)^2 \right]^{0.5} \tag{9.A9}
\]

\[
\sin \Omega_b = \frac{R_b}{f}
\]

Substituting for \( f \) out of equation (9.A9);

\[
\sin \Omega_b = \frac{R_b}{\left[ e^2 + (a + d)^2 \right]^{0.5}} \tag{10.A9}
\]

\[
\Omega_b = \sin^{-1} \left[ \frac{R_b}{\left\{ e^2 + (a + d)^2 \right\}^{0.5}} \right] \tag{11.A9}
\]

Now;

\[
\kappa_b = \frac{\pi}{2} - \left( \Omega_b + \phi_b \right)
\]

and;

\[
\eta_b = \frac{\pi}{2} - \kappa_b = \Omega_b + \phi_b \tag{12.A9}
\]
Combining equations (12.A9), (11.A9) and (8.A9) gives;

\[ \eta_b = \sin^{-1} \left[ \frac{R_b}{\left\{ \frac{e^2}{2} + (a + d)^2 \right\}^{0.5}} \right] + \tan^{-1} \left( \frac{e}{a + d} \right) \cdots (13.A9) \]

**Total angle of lap.**

\[ \frac{\pi}{2} + \eta_b + \zeta_b \]

\[ \frac{\pi}{2} + \sin^{-1} \left[ \frac{R_b}{\left\{ \frac{e^2}{2} + (a + d)^2 \right\}^{0.5}} \right] + \tan^{-1} \left( \frac{e}{a + d} \right) + \pi - \theta_b + \sin^{-1} \left( \frac{2R_b}{c} \right) \]
Length of lapped yarn.

The length \( L_b \) contacting the pin = \( R_b \left( \frac{\pi}{2} + \eta_b + \zeta_b \right) \)

where \( \pi, \eta_b, \) and \( \zeta_b \) are in radians.

Therefore, from (14.A9):

\[
L_b = R_b \left\{ \frac{3\pi}{2} + \sin^{-1}\left[ \frac{R_b}{\left(e^2 + (a+d)^2\right)^{0.5}} \right] + \tan^{-1}\left( \frac{e}{a+d} \right) - \theta_b + \sin^{-1}\left( \frac{2R_b}{c} \right) \right\}
\]

.................................................. (15.A9)

Length (g) of yarn from inlet point 7 to tangency point 5

\[
g = \left[ (f)^2 - (R_b)^2 \right]^{0.5}
\]

and from (9.A9):

\[
g = \left\{ \left[ e^2 + (a+d)^2 \right] - (R_b)^2 \right\}^{0.5}
\]

.................................................. (16.A9)
Total length \( z_b \) of yarn in the compensator:

\[
z_b = \frac{2}{\pi} \left[ \left( \frac{c}{2} \right)^2 + \left( R_b \right)^2 \right]^{0.5} + \left\{ \left[ \theta^2 + (a+d)^2 \right] - \left( R_b \right)^2 \right\}^{0.5} + \left\{ \left[ \theta^2 + (b+d)^2 \right] - \left( R_b \right)^2 \right\}^{0.5}
\]

\[
+ 2R_b \left[ \frac{3\pi}{2} + \sin^{-1} \left( \left\{ \theta^2 + (a+d)^2 \right\}^{0.5} \right) \right] + \tan^{-1} \left( \frac{\theta}{a+d} \right) - \theta + \sin^{-1} \left( \frac{2R_b}{c} \right)
\]

\[
\text{................................................................. (17.A9)}
\]

and to express \( \theta_b \) in terms of the other variables:

\[
\theta_b = \frac{3\pi}{2} + \sin^{-1} \left( \left\{ \frac{R_b}{\theta^2 + (a+d)^2} \right\}^{0.5} \right) + \tan^{-1} \left( \frac{\theta}{a+d} \right) + \sin^{-1} \left( \frac{2R_b}{c} \right)
\]

\[
- \left[ z_b^{-2} \left( \left( \frac{c}{2} \right)^2 + \left( R_b \right)^2 \right) + \left\{ \left[ \theta^2 + (a+d)^2 \right] - \left( R_b \right)^2 \right\}^{0.5} + \left\{ \left[ \theta^2 + (b+d)^2 \right] - \left( R_b \right)^2 \right\}^{0.5} \right] \frac{2R_b}{z_b}
\]

\[
\text{................................................................. (18.A9)}
\]

It should be noted that in (17.A9) and (18.A9) parameters \( e \) and \( d \) are dependent upon \( \theta_b \), ie:

\[
e = \frac{c}{2} \cos \left( \pi - \theta_b \right) \text{ .................................................. (19.A9)}
\]

and

\[
d = \frac{c}{2} \sin \left( \pi - \theta_b \right) \text{ .................................................. (20.A9)}
\]
Appendix 10

MANUFACTURING DRAWINGS FOR THE 7-BAR MECHANISM
QUANTITY: ONE
MATERIAL: MILD STEEL
COMPUTER FILE: OUT CRANK

HOLE: HB

ALL DETAILS OF THIS HOUSING ARE THE SAME AS FOR SECTION AA ON THE INPUT MEMBER (DRAWING MC.11)

Loughborough University of Technology
Department of Mechanical Engineering

This drawing conforms to BS 308
Dimensions in millimetres

TOLERANCES UNLESS OTHERWISE STATED:
LINEAR
ANGULAR

SURFACE TEXTURE \( V = \mu m \) MAX

PROJECT TITLE: MECHANICAL COMPENSATOR

DATE: FEB 1989
APPROVED

COURSE: YEAR: GROUP

SCALE: 1:1

ANGLE PROJN
QUANTITY: ONE
MATERIAL: MILD STEEL
COMPUTER FILE: OPSPACER
CHAMFER BOTH ENDS 1 ° 45 DEGREES
UNDERCUT BEARING SEATS 1.5 WIDE X 0.5 DEEP

QUANTITY: ONE
MATERIAL: MILD STEEL
COMPUTER FILE: OBSHAFT

LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY
DEPARTMENT OF MECHANICAL ENGINEERING
THIS DRAWING CONFORMS TO BS.308
DIMENSIONS IN MILLIMETRES
TOLERANCES UNLESS OTHERWISE STATED:
LINEAR...........
ANGULAR........
SURFACE TEXTURE V = μm MAX

PROJECT TITLE
MECHANICAL COMPENSATOR

DRAWN I.C. WRIGHT
DATE FEB 1989
APPROVED

OUTPUT BEARING SHAFT

COURSE YEAR GROUP DATE SCALE ANGLE PROJ.

MC.12
DETAILS OF BEARING BORE AND CIRCLIP GROOVE ARE AS SHOWN ON SECTION AA FOR INPUT MEMBER (DRG. MC.1)

QUANTITY: ONE
MATERIAL: MILD STEEL
COMPUTER FILE: CONCRANK
LOUGHBOROUGH UNIVERSITY
OF TECHNOLOGY
DEPARTMENT OF MECHANICAL ENGINEERING

TOLERANCES UNLESS OTHERWISE STATED:
LINEAR...........
ANGULAR...........
SURFACE TEXTURE $V = \mu m$ MAX

THIS DRAWING CONFORMS TO BS.308
DIMENSIONS IN MILLIMETRES

PROJECT TITLE
MECHANICAL COMPENSATOR

DRAWN I.C. WRIGHT
APPROVED

DATE FEB. 1989

CONTRIBUTED CRANK LOCKING SCREW

COURSE YEAR GROUP

SCALE 1:1

DRAWING NUMBER MC. 10

QUANTITY: ONE
MATERIAL: MILD STEEL
COMPUTER FILE: CCLSCREW
KEYWAY DETAILS

HOLE, 10 DIA.

QUANTITY: ONE
MATERIAL: MILD STEEL
COMPUTER FILE: CCSLEEVE

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OF TECHNOLOGY
DEPARTMENT OF MECHANICAL ENGINEERING
THIS DRAWING CONFORMS TO BS.308
DIMENSIONS IN MILLIMETRES

TOLERANCES UNLESS OTHERWISE STATED:
LINEAR...........
ANGULAR...........
SURFACE TEXTURE $V = \mu m$ MAX

PROJECT TITLE
MECHANICAL COMPENSATOR

DRAWN I.C.WRIGHT
TITLE
TITLE
CONTROL CRANK SLEEVE

DATE FEB 1989
APPROVED

COURSE
YEAR
GROUP

DATE

SCALE 1:1

DRAWING NUMBER MC.9

ANGLE PROJ.
QUANTITY: FOUR
MATERIAL: MILD STEEL
COMPUTER FILE: BSPAC2
MATERIAL: MILD STEEL
QUANTITY: FOUR
COMPUTER FILE: BSPAC1

TOLERANCES UNLESS OTHERWISE STATED:
LINEAR...........
ANGULAR...........
SURFACE TEXTURE √ = µm MAX

MATERIAL: MILD STEEL
QUANTITY: FOUR
COMPUTER FILE: BSPAC1

PROJECT TITLE
MECHANICAL COMPENSATOR

DRAWN: I.C. WRIGHT
DATE: FEB 1989
APPROVED

COURSE: YEAR: GROUP

SCALE: 1:1

DRAWING NUMBER: MC.7

THIS DRAWING CONFORMS TO BS.308
DIMENSIONS IN MILLIMETRES

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FEB 1989

C.V. WRIGHT

NON-CIRCLIP END

Bearing Spacer

APPROVED

MC.7
INTERNAL BORE
10.2 / 10.3 DIAMETER
CONCENTRIC WITH 0.D.

QUANTITY: ONE
MATERIAL: MILD STEEL

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THIS DRAWING CONFORMS TO BS.308
DIMENSIONS IN MILLIMETRES

TOLERANCES UNLESS OTHERWISE STATED:
LINEAR........
ANGULAR........
SURFACE TEXTURE $\sqrt{=} = 1\mu m$ MAX

PROJECT TITLE
MECHANICAL COMPENSATOR

DRAWN I.C.WRIGHT
DATE FEB 1989

TITLE
OUTPUT BEARING SPACER
APPROVED

COURSE YEAR GROUP

DATE

SCALE 1:1

DRAWING NUMBER
MC.6

ANGLE PROJ
CENTRES PARALLEL WITHIN 0.05

QUANTITY: ONE
MATERIAL: MILD STEEL

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THIS DRAWING CONFORMS TO BS.308
DIMENSIONS IN MILLIMETRES

PROJECT TITLE
MECHANICAL COMPENSATOR

DATE FEB '89
APPROVED

SCALE 1:1
ANGLE PROJ.

MC.4
MC.3

0.5 R (MAX)

5

10

25.0

34.9

34.9

34.9

34.9

60.5

SECTION AA

CHF 1 0.45 DEGREES

QUANTITY: ONE

MATERIAL: MILD STEEL

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TOLERANCES UNLESS OTHERWISE STATED:
LINEAR........
ANGULAR........
SURFACE TEXTURE \( \nu = \) \( \mu \) \text{MAX}

PROJECT TITLE
MECHANICAL COMPENSATOR

DRAWN I.C. WRIGHT

TITLE
OUTPUT BEARING CAP

DATE FEB '89
APPROVED

COURSE YEAR GROUP

DATE

SCALE

DRAWING NUMBER

MC.3

THIS DRAWING CONFORMS TO BS.308
DIMENSIONS IN MILLIMETRES

ANGLE PROJN
Loughborough University of Technology
Department of Mechanical Engineering

Tolerances unless otherwise stated:
Linear........
Angular........
Surface texture $V = \mu m$ Max

Dimensions in millimetres

Quantity: One
Material: Mild Steel

Project Title: Mechanical Compensator

Drawn: L.C. Wright
Date: Feb '89
Approved

Course: Year: Group

Scale: 1:1

3D0 Angle proj

Drawing Number

This drawing conforms to BS. 308

MC. 2