



DESIGN AND PERFORMANCE OF WEFT-KNITTING MACHINERY

A thesis presented for the degree of

DOCTOR OF PHILOSOPHY

by

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CHAPTER I

INTRODUCTION

## CHAPTER I

### INTRODUCTION

#### Preamble

In recent years the knitting industry has expanded considerably due to the increased demand for knitted fabrics. Not only does the weft knitting industry produce more of the fabrics which are exclusive to the knitting trade, but it has also branched out into fields which were previously only open to the weaving trade.

This increasing demand is partly due to the development of new fibres, but mainly to the technological improvements of the knitting machines, which now provide many new structures and more scope for patterned fabrics. Another improvement in weft knitting machinery has been an increased rate of fabric production, but in this field the limitations of the machines have now almost been reached.

The work discussed in the following chapters is primarily concerned with an investigation into the fundamentals of the weft knitting machine to see if substantial increases in rates of fabric production can be achieved.

#### 1. A General Description of Knitting Machine Components

Weft knitting machinery may be divided into two groups - the circular type and the flat machine. Both types of machine can be used to produce a similar fabric construction but each has its own advantages. For example, the circular machine, because it is a continuous process, will produce fabric at a far greater rate than the flat machine, but the latter can be used to shape garments to the required dimensions.

FIG. 1

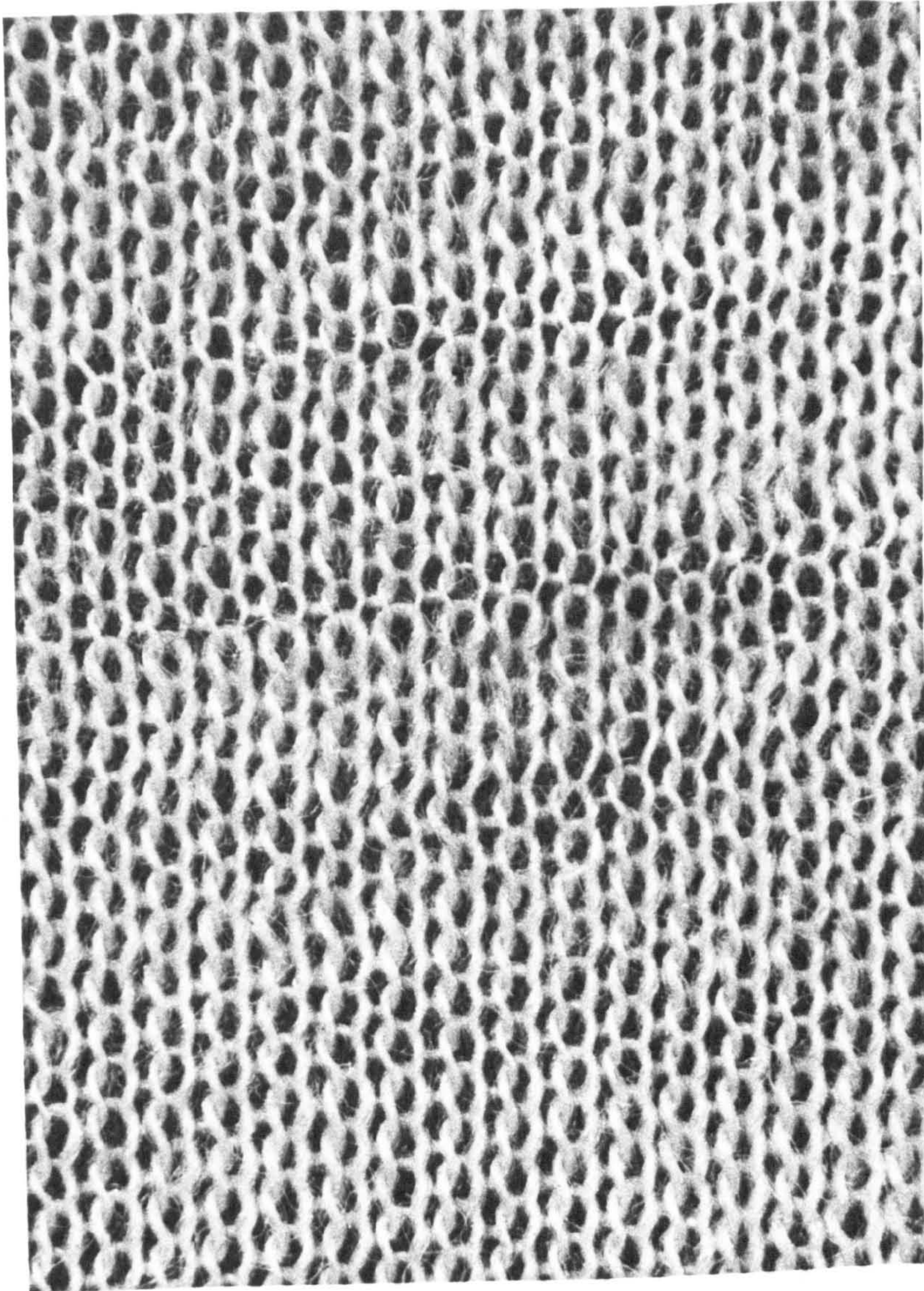
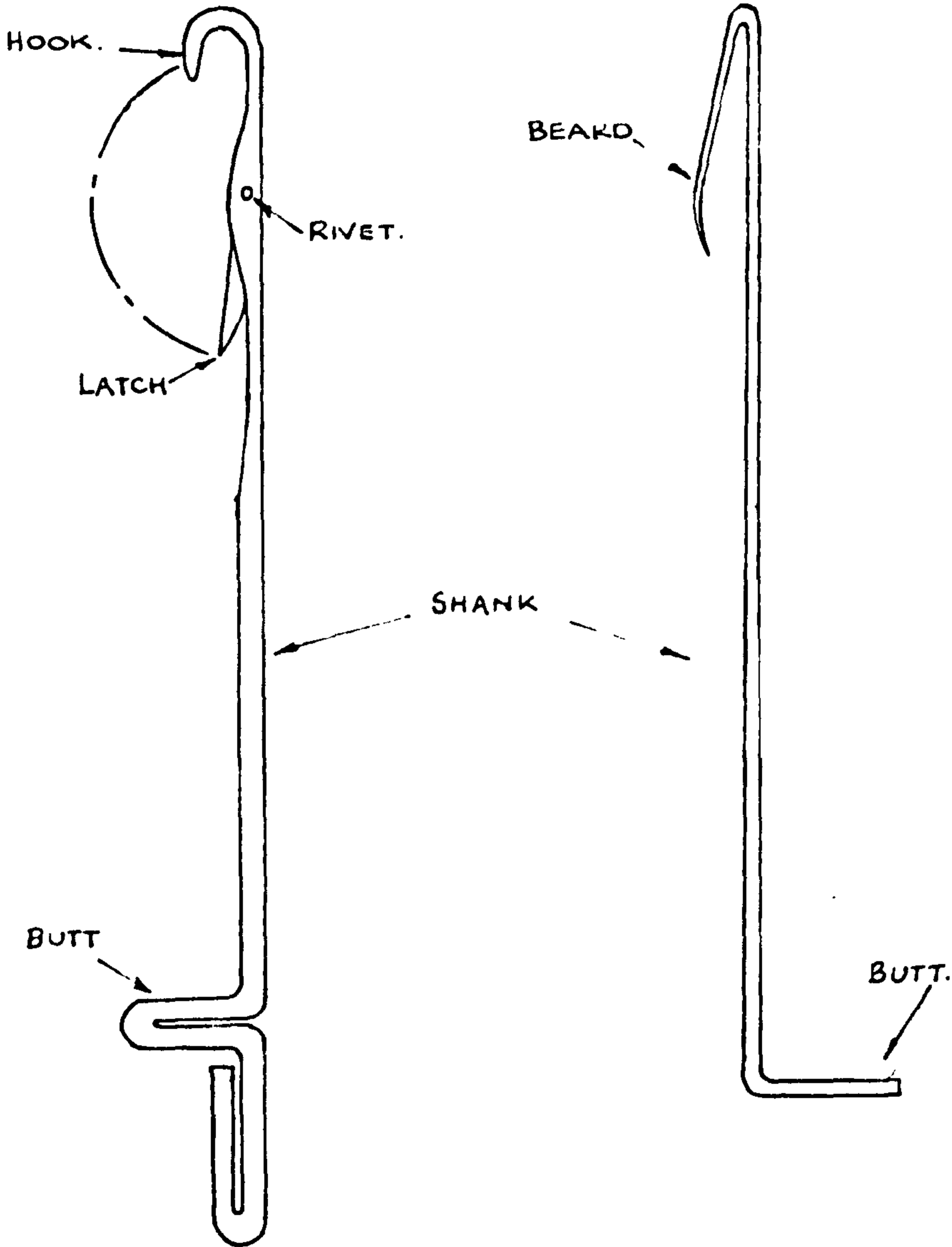


FIG 2

KNITTING MACHINE NEEDLES



LATCH NEEDLE

BEARDED NEEDLE

Weft knitted fabrics are comprised of a series of horizontal loops which are interlocked (Fig. 1). The fabric is extensible in both length and width which makes it suitable for close fitting garments. Complex knitting structures can be used to reduce the amount of stretch and, therefore, give a more rigid fabric which is comparable to the woven fabric,

(a) Machine Needles

The two types of needles used in the production of weft knitted fabrics are the latch needle and the bearded needle (Fig. 2).

The bearded needle is easier to manufacture as it is essentially a piece of thin wire bent to the correct shape; however, during loop formation external pressure is required to close the beard on to the shank.

The latch needle is more complicated and, therefore, more expensive to manufacture, since it is made up of more than one part. On the other hand, it does not need an external presser during loop formation as the latch is opened and closed by the yarn and this enables higher speeds to be achieved and requires less complicated mechanical motions to produce the loop forming action. For this reason the latch needle, used in conjunction with circular machines, was chosen for the first part of this work. In later work a new needle was designed for further increases in the rate of fabric production.

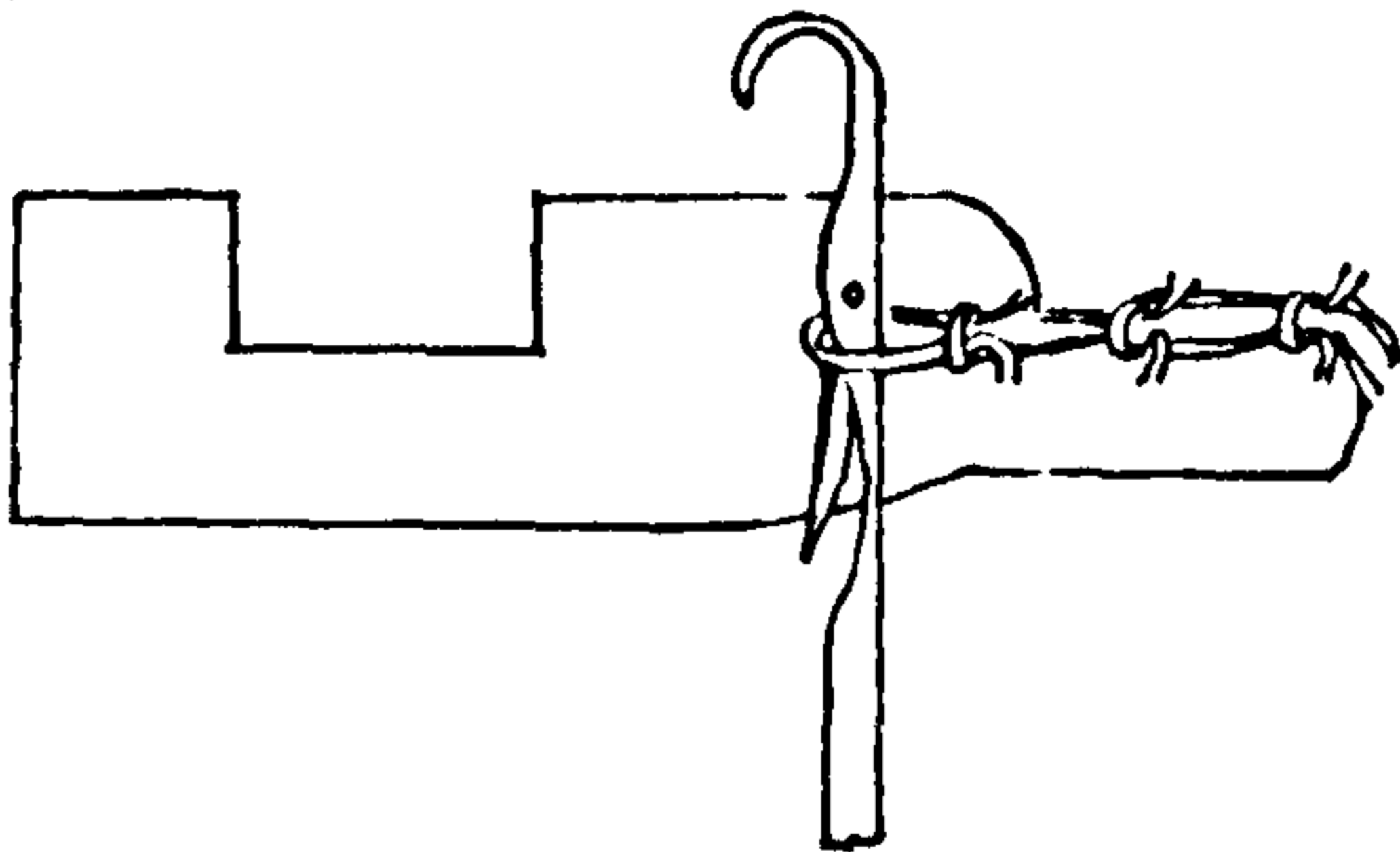
(b) Loop Formation with Latch Needles

The loop forming action with latch needles is shown in Fig. 3. The normal position of the latch needle, before the loop formation takes place, is one in which the old loop holds the latch open - this is termed the "running" position. (Fig. 3A).

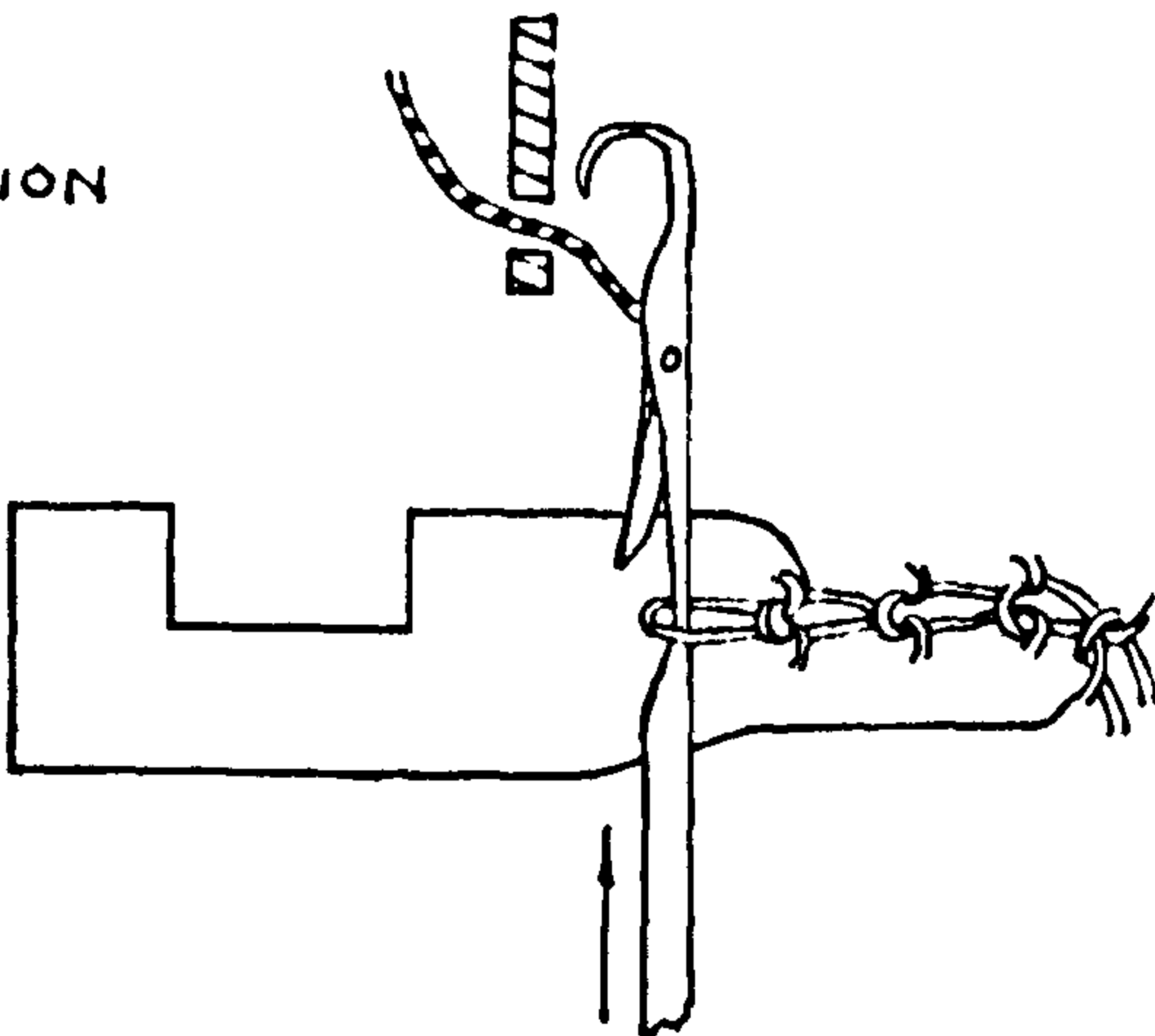
FIG 3

LOOP-FORMATION WITH LATCH NEEDLES

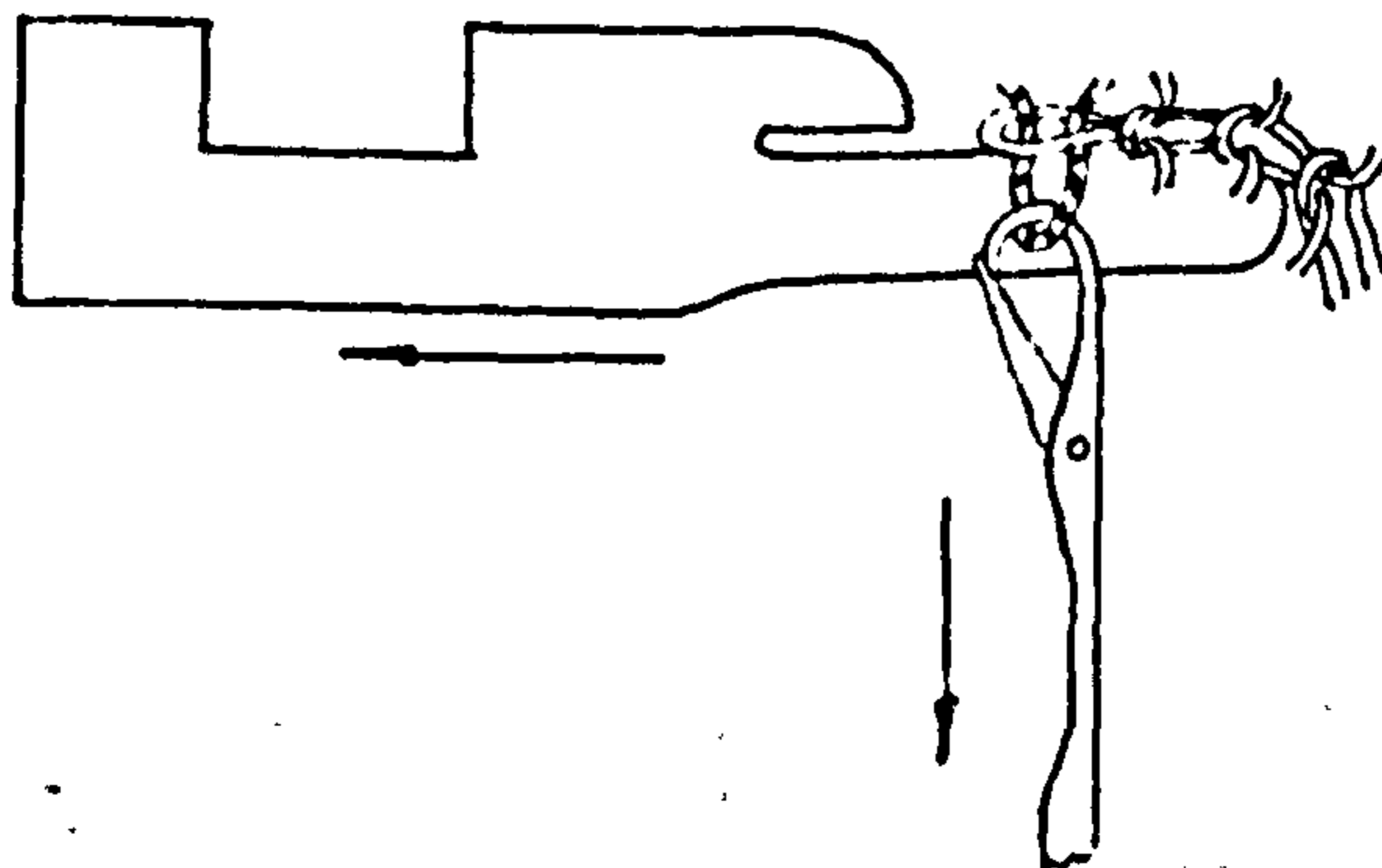
A. RUNNING POSITION



B. CLEARING POSITION



C. KNITTING POINT.



To produce a new loop the needle moves upwards so that the latch is released from the old loop, the fabric being held down to prevent the old loop riding up with the needle - this position is known as the "clearing" position. (Fig. 3B).

The needle then moves downwards as new yarn is presented to the hook of the needle, the old loop closes the latch and the new yarn is trapped in the hook. As the needle descends further the old loop rides over the closed latch to join the fabric. The needle then moves down to its lowest position, (i.e. knitting point, Fig. 3C) and pulls through the new loop before rising to the running position.

The loop-forming action described above is for plain knitted fabrics and is the same for all knitting machines, although with more complicated fabric structures, slight variations in the loop formation are required.

### (c) Sinkers

These are used in conjunction with the needles to assist in the loop-forming action, those used in weft knitting machines being either loop-forming or web-holding types.

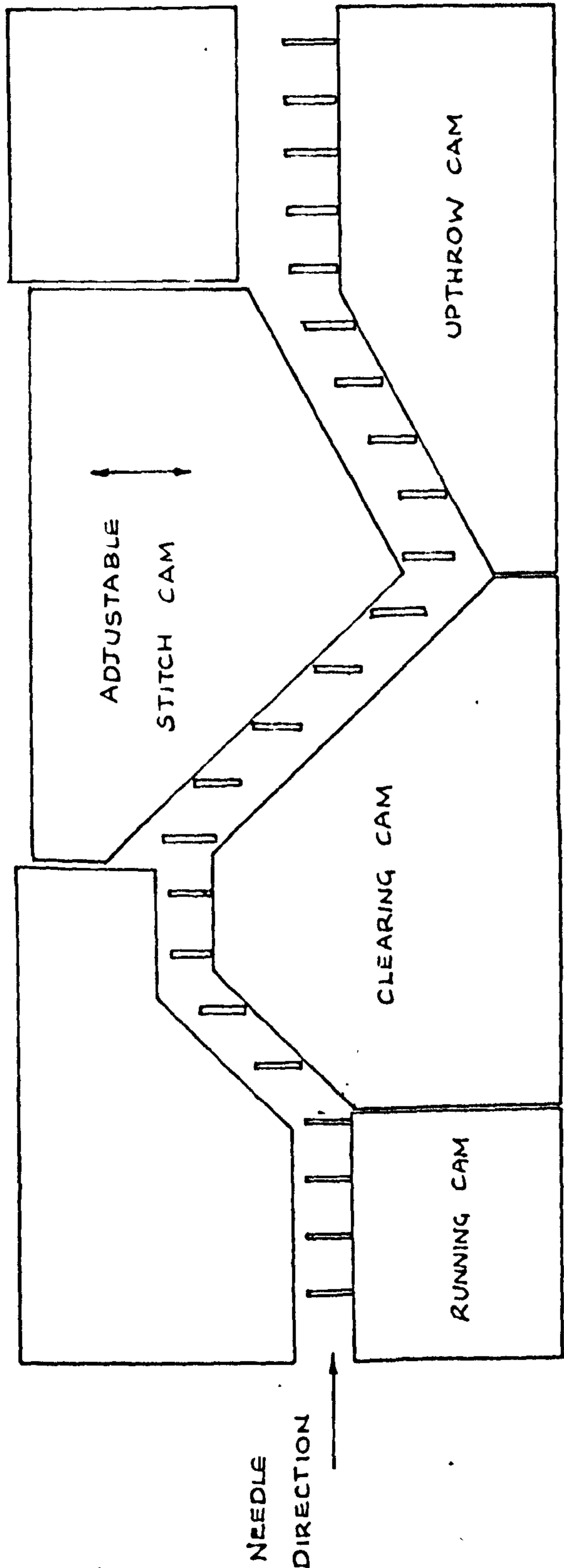
Loop-forming sinkers, which are normally used with bearded needles, form the loops by corrugating the yarn inbetween the needles; they are, therefore, absolutely essential in forming a loop.

The web-holding sinkers (shown in Fig. 3) are used with latch needles and hold down the loop as the needle rises up the clearing cam, thereby ensuring that the latch is freed from the old loop. This type of sinker is extremely useful if the fabric has pressed off the needles, as a new fabric may be started immediately. Without sinkers it is necessary to attach



Fig 4

TYPICAL COMMERCIAL CAM SYSTEM



one end of the fabric to the needles and the other to the fabric take-down unit, in order to provide sufficient tension to prevent the fabric rising up when the needles rise up at the clearing position.

#### (d) Cam Systems

The needles are held in tricks in the bed of the machine, with butts protruding outwards, and are free to slide up and down.

The cam system (see Fig. 4) fits closely to the machine bed and is made up of separate pieces of hardened steel comprising running, clearing, stitch and upthrow cams. These cams are designed to move the needle butts through the relative positions necessary for loop formation. The cams are made up of separate pieces to allow the stitch cam to be adjustable and, therefore, a range of fabric qualities to be produced. Lowering the stitch cam, for instance, means that the needle will move further down at the knitting point, draw more yarn and, therefore, produce a longer loop giving an overall slacker fabric.

#### (e) Rates of Fabric Production

To a large extent the cam system determines the rate of fabric production on a knitting machine. On a circular weft knitting machine the rate of fabric production is determined by :

- (i) the number of loop-forming sections (usually termed "feeders") on a machine,
- (ii) the circumferential speed of the knitting cylinder.

(i) A circular machine usually has many feeders positioned around its circumference. For example, in the case of a 30" diameter machine there may be as many as 36 feeders, each with its own yarn supply. This means that for every revolution of the machine 36 rows of loops are knitted.

The circumferential length of each feeder determines how many feeders can be fitted on to a machine, and this ultimately depends upon the maximum steepness of the cam angles, (usually  $45^{\circ}$ ), and the amplitude of vertical needle movement, i.e. the vertical distance between the needles at the clearing position and the knitting point.

(ii) The circumferential speed of the cylinder on a knitting machine is dependent upon its diameter, but the important factor, as regards the rate of fabric production, is the linear speed of the actual needle butts. For example, a 30" diameter machine runs at approximately 22 r.p.m. and an 8" diameter machine runs at approximately 85 r.p.m., but in both these machines the linear speed of the needle butts is approximately 170 ft/min.

In commercial practice, machines are not normally run with the linear speed of the needle butts above 200 ft/min. as the forces on the needle butts become too high and they tend to break. (1).

#### (f) Circular Weft Knitting Machines

The method of rotation on circular knitting machines is varied. In some machines the cam boxes rotate and the cylinder is fixed, whereas in others the cylinder rotates and the cam boxes are fixed. In each system the relative movement of needle butts to cam system is the same, although there are factors which give advantages to each type.

On the rotating cam box machine the yarn package supplies move with the cam boxes so that the chances of yarns becoming entangled, or even yarn packages coming off the revolving stand, are quite high, and this tends to limit the speed at which

the machine may run. However, since the cylinder is fixed the fabric is also stationary which allows an operator, who may be controlling a number of machines, to check for fabric faults as the machine is still knitting.

On the rotating cylinder type of machine the speeds may be higher but, since the fabric rotates, the examination of the fabric is usually made whilst the machine is stationary. Apart from the advantage of speed on this type of machine, another advantage has become more apparent over the past few years with the introduction of a reliable yarn speed measuring device. This device enables an operator to set all the stitch cams on the rotating cylinder machine to give the same yarn speed whilst the machine is running and thus obtain a level fabric.

## 2. Properties of a Knitted Fabric

A necessary property of a knitted fabric is that it will stretch in width and length so as to fit closely to the wearer. The fabric, however, when distorted like this, will not return to its original dimensions immediately. This feature of the fabric often leads to errors in the fabric manufacturing industries where it is desirable to check the dimensions of a fabric (i.e. courses and wales per inch) as soon as it has been knitted. In this case the fabric has been distorted in width by a stretcher board and in length by the take-down rollers, and it is therefore unlikely that any measurements will be accurate. To obtain the correct dimensions of c.p.i. and w.p.i. it is essential that the fabric is completely free from any previous strains. This can be achieved by allowing the fabric to relax naturally for a long period of time, or alternatively, for rapid relaxation, the fabric can be immersed in water.

These relaxation treatments are obviously inconvenient for the fabric manufacturers, and therefore it was important to find a more suitable method of stipulating the dimensions or quality of a fabric. A more precise method is to measure the stitch density (i.e. c.p.i. x w.p.i.) as to some extent any distortion that increases the length of the fabric is compensated by a decrease in width, although even this is only really accurate if the fabric is free from strain.

It was Doyle (2) who first showed that stitch length was closely related to stitch density and that it was unaffected by any strains in the fabric. He suggested that since a fixed amount of yarn was knitted into the fabric it must remain there whatever the extent of the fabric distortion. He also indicated that the stitch length could be measured by unroving a length of yarn from the fabric and dividing this length by the number of loops from which it was knitted.

Munden (3) showed conclusively that the relaxed dimensions of a fabric were determined by the stitch length. He did this by knitting fabrics to a wide range of stitch lengths, and providing relationships of fabric dimensions and stitch length. He showed that knitting variables such as yarn count, fabric take-down tension, machine gauge, etc. would affect the relaxed dimensions of a fabric, but that they did so by causing a change in the stitch length and, therefore, the relationships given below were true for all conditions of knitting.

These relationships of fabric dimensions and stitch length are :

$$\text{Stitch density } N = \frac{k_s}{l^2}$$

$$\text{Courses per inch (c.p.i.)} = \frac{k_c}{l}$$

$$\text{Wales per inch (w.p.i.)} = \frac{k_w}{l}$$

where  $l$  is the stitch length;  $k_s$ ,  $k_c$  and  $k_w$  are constants and their values after dry and wet relaxations were given as :

	<u>Dry relaxation</u>	<u>Wet relaxation</u>
$k_s$	19.0	21.6
$k_c$	5.0	5.3
$k_w$	3.8	4.1

### 3. Quality Control Systems

The technique of measuring stitch length by unroving yarn has certain practical disadvantages which prevent it from becoming a standard method of quality control in the factory. It is a slow and tedious procedure as a number of lengths of yarn have to be unroved and measured, and it also involves cutting the fabric. In addition, if it is necessary to knit to a specific stitch length, a certain amount of guesswork is required, the fabric must be knitted and the stitch length measured, and if this is not the value required, then the stitch cams must be adjusted, more fabric knitted and the stitch length measured again. This procedure must be repeated until the correct stitch length is obtained.

Instruments have recently been developed which eliminate these disadvantages and enable the stitch length to be controlled as the machine is running (4, 5, 6).

#### (a) Yarn Speed Meter

This instrument is designed for rotating cylinder machines, and it will measure the speed of the yarn accurately as it is fed to the needles. If the speed of the machine is known the relationship

$$\frac{\text{Yarn speed meter reading (ft/min)}}{\text{Machine speed (r.p.m.)}} = \text{course length } L \text{ (ft).}$$

will enable the stitch length to be obtained from

$$\frac{\text{Course length } (L) \text{ (ins)}}{\text{Number of needles } n} = \text{Stitch length } l \text{ (ins).}$$

If, therefore, a fabric is required with a specific stitch length, it is a simple matter to calculate the necessary yarn speed meter reading and then set each feeder to give this reading, by adjusting the stitch cam whilst the machine is running.

(b) Yarn Length Counter

On rotating cam box machines the yarn supplies rotate and, therefore, a yarn speed meter cannot be used. However, a yarn length counter has been designed which can be clamped to the machine and this will measure the quantity of yarn fed to each feeder. The stitch length can be calculated easily provided the number of machine revolutions are counted and the length of yarn fed to the needles is noted.

(c) Positive Feed

The positive feed system is designed to feed a fixed quantity of yarn to the needles irrespective of any of the knitting variables, such as yarn tension, stitch cam setting, yarn count, etc.

In one type of positive feed system the yarn is wrapped around a conical wheel, which is sufficiently rough to prevent the yarn slipping. The roller rotates at a constant speed in relation to the machine speed and supplies a constant quantity of yarn to the needles.

In another type, the yarn is passed between a roughened belt and a free wheel; the belt, which is driven from a control wheel, drives the yarn to the needles at a constant speed.

A range of stitch lengths may be obtained by feeding in different amounts of yarn to the needles. With the first type this is achieved by moving the yarn to a different position on the conical wheel, and to obtain an even larger range of stitch length the machine gear can be altered to give different speeds to the roller. In the second type stitch length changes can be obtained by altering the diameter of the control wheel which in turn alters the belt speed and, therefore, the yarn speed.

One advantage of the second method, often referred to as "trip-tape", is that the stitch length of all the feeders can be changed at the main control wheel, whereas in the conical roller method the yarn of the feeders has to be adjusted to a different position on each individual conical roller.

Both these systems, which are very similar, guarantee an excellent quality fabric and have been so successful over the last ten years that almost all new knitting machines are fitted with one or other of these positive feed devices.

The stitch cam, which in previous quality control systems has been the normal method of altering the stitch length, is now used to control the tension of the yarn between the positive feed roller and the needles. For instance, if insufficient yarn is supplied to the needles, then the tension becomes too high and the yarn may break - by raising the stitch cam the tension will be reduced; alternatively, if too much yarn is fed to the needles, the tension becomes so low that dropped stitches will occur - a downward adjustment of the stitch cam will increase the tension and prevent this.



#### 4. Survey of Previous Work on the Mechanisms of Loop-Forming

##### (a) General Survey

It may be seen from an examination of the  $45^{\circ}$  commercial cam system (shown in Fig. 4), that the separate cam pieces are essentially linear cams and, consequently, the needle butts will change direction abruptly when moving from one linear cam to another.

From a theoretical point of view the needle butt would have infinite acceleration at the ends of the linear cams and the butts would shear off. In practice, however, this does not normally happen, partly because the needle butt movement is controlled between two cam tracks, and partly because the needle butts have a natural resilience which enables them to absorb the shock. However, a theoretical investigation of the forces of the needle butts in the linear cam system, made by Munden (7), showed that if the  $45^{\circ}$  cam angle were increased to  $55^{\circ}$ , then the forces would increase three times. With cam angles above  $55^{\circ}$  severe butt breakage might occur, especially if the friction between the needle butts and cam track were high due to inadequate lubrication.

It has been stated that the rates of fabric production can be increased by :

- (1) increasing the number of feeders on a machine,
- (2) increasing the rotational speed of the machine.

In recent years most of the increases in rates of fabric production have been achieved by fitting as many feeders to the circumference of the machine as is physically possible. The point has now been reached, however, where any further increases in this direction can only be achieved by :

- (i) reductions in circumferential length, i.e. by increasing the angles of the cam system, which normally range between  $45^{\circ}$  and  $50^{\circ}$ , or
- (ii) reducing the vertical amplitude - this is usually dependent upon the dimensions of the knitting needle.

Speeds of machines have only increased by a marginal amount over the past twenty years. One reason for this is that speed increases would inevitably affect many working parts of the machine, apart from the cam system and, consequently, more sophisticated and expensive machine bearing systems, gearing techniques, yarn stop motions, etc. would be necessary. The other reason is that the needle butt reactions on the linear cam track would also increase with increases in speed and could lead to severe butt breakage.

(b) Knapton's researches (8)

This work was concerned with an analysis of the loop-forming action and included investigations into increasing the rates of fabric production. For a brief summary, Knapton's work may be divided into three parts :

- (i) An investigation into the effect of cam setting and yarn tension on stitch length.
- (ii) A calculation of the maximum yarn tension in the knitting cycle and the importance of the positions of these maximum tensions.
- (iii) Measurements of the forces and investigations into increasing the rates of fabric production.

(1) It was shown that stitch cam settings had no real control over the final stitch length of the fabric, even when the depth was positioned accurately. Although the stitch cam initially determined the length of yarn pulled into a loop, the final stitch length was determined mainly by the tension of the yarn as it was fed to the needles. Using a fixed cam setting, it was shown that changes in yarn tension could alter the stitch length by as much as 20%, and an increase in tension always produced a smaller stitch length.

Previous to Knapton's work a number of reasons had been suggested to explain this variation in stitch length (9). It was thought possible, for instance, that the needle was extending the yarn as it descended to the knitting point, especially with high yarn input tensions. Knapton showed, however, that the yarn would break before it could extend to give this 20% variation in stitch length. Another suggestion was that the inertia of the needle, after the knitting point, could fluctuate sufficiently to give a variation in stitch length. In this case, often referred to as "needle fling", the needle would tend to move further below the knitting point with low yarn tensions than with high yarn tensions; this would produce a variation in stitch length with the values of stitch length being greater than that predicted by the cam setting.

Knapton showed that under all knitting conditions the actual stitch length was always less than the theoretical stitch length, predicted by the geometry of the needles on the stitch cam, and was 20% less than the theoretical length at low yarn input tensions, and as high as 40% less under high input tensions.

To explain these observations it must be assumed that as the needle moves down the stitch cam to form a loop, some of the yarn in the loop is robbed from the previously formed loop. The reasons for the variation in stitch length with change in knitting tension can only be explained by "robbing-back".

- (ii) A method was shown for calculating the maximum yarn tensions, developed in the knitting cycle, from the geometry of the knitting elements and the known values of yarn friction. These calculated values corresponded closely to the knitting performance of the yarn. For example, the calculated maximum tensions of a waxed yarn were much lower than those of an unwaxed yarn. When knitting the yarns the waxed yarn knitted satisfactorily but the unwaxed yarn produced holes in the fabric. The breaking load of the unwaxed yarn was measured and it proved to be lower than the calculated values of maximum tension.

The positions of the maximum tensions in the knitting cycle had a larger bearing on the stitch length than their actual values. It was shown that the calculated stitch length, obtained from the geometry of the knitting elements at the positions of maximum tensions, corresponded closely to the actual measured stitch lengths in the fabric.

The high maximum values of tension increased the effort required to rotate the machine and the wear on the knitting element and it was therefore considered important to see if reductions could be made in these values.

Calculations made using a  $60^\circ$  cam system, as opposed to the  $45^\circ$  cam system showed greatly reduced maximum tensions

due to a reduced number of yarn/metal contacts in the knitting cycle. This was proved practically, as the reduced maximum tensions allowed a high friction yarn to knit more satisfactorily with a  $60^{\circ}$  cam system than with a  $45^{\circ}$  cam. In addition, it was shown that when using paraffin wax emulsion, a reduction in yarn friction value was obtained which resulted in lower values of maximum tension.

(iii) A torque measuring device was designed which could measure the effort required to rotate the machine when using different knitting conditions. It was mounted on the main driving shaft of the machine and any twist that developed in the shaft was detected and measured by a change in capacitance between two condenser plates.

Several linear cams were made with angles ranging from  $45^{\circ}$  to  $65^{\circ}$  and it was shown that the torque increased with an increase in the cam angle when the machine was rotating without yarn. The increase in torque was assumed to be due to the larger reactions of needle butt forces on the steeper cams. However, when a yarn was knitted it was observed that the torque was reasonably constant with an increase in cam angle. It was explained that the reduced maximum tension, due to less yarn/metal contacts, had balanced out the increased reaction of needle butts.

It was suggested from these tests that an increase in cam angle from  $45^{\circ}$  to  $65^{\circ}$  might prove acceptable in practice and that this increase could lead to twice the number of feeders that are fitted as standard on present-day machines.

It was pointed out, however, that certain problems arose from the use of steep angle cams; the running forces of individual needles on the cam track increased rapidly with increase in cam angle and on starting the machine these forces were particularly high.

It was suggested that a non-linear cam could be used to reduce these forces, although it was appreciated that the curved shape would limit the number of feeders that could be fitted to the machine. However, it was suggested that a combination of the steep angled, curved cam and an increase in machine speed would provide the same increases in rates of fabric production as those achieved with the steep angled linear cams.

A parabolic curved cam was designed and made, but the performance of this cam was not investigated through lack of available time.

(c) Wignall's research (10)

This work included investigations into the effect of knitting variables on stitch length and machine forces.

He devised a mechanical torque meter that worked by means of a spring-controlled peg attached to the drive of the machine. This moved a cam pointer when a change in torque occurred and appeared to give reliable results.

His work on a commercial  $45^{\circ}$  cam system showed that an increase in speed reduced the stitch length but increased the torque required to rotate the machine; with lower stitch cam settings, which produce longer stitches, a decrease in torque occurred.

On working with yarns having different characteristics such as yarn thickness, friction, extensibility, etc., it was seen that these properties had very little effect on the stitch length, although the torque increased with yarn thickness and with high friction yarns.

An analysis was also made on the forces at various positions in the knitting cycle, and it was seen that the proportion of forces at each position changed significantly with changes in knitting variables. The measurement of these resultant horizontal forces, produced by the needles, was achieved by maintaining separate floating cams in equilibrium, using a spring balance.

#### 5. Purpose of Present Work

This work is primarily concerned with investigations into increasing the rate of fabric production on circular weft knitting machinery.

In Phase I, (Chapters 2 - 5), non-linear cam systems with controlled acceleration characteristics are investigated, as it is considered that a combination of steeper angled, non-linear cams and higher circumferential machine speeds will not only provide satisfactory needle butt performance but also give increases in rates of fabric production.

The work in Phase I compares the performance of linear and non-linear cams, and includes :

- (1) Calculations of the yarn tensions developed within the loop-forming portions of the various cams, and a comparison of the measured and calculated stitch length values.

- (ii) A comparison of the measured and calculated horizontal forces acting on the needles with and without yarn supplied to the needle hooks. Calculations of the acceleration forces acting on the needles and an assessment of the maximum machine speed, when using a non-linear cam.
- (iii) A comparison of stitch length, fabric appearance and needle butt performance, for the various cams, at speeds much greater than commercial speeds.

Phase II of this work considers the possibility of increasing the rate of fabric production by reducing the amplitude of vertical needle movement (i.e. the distance between the clearing height and the knitting point); a reduction of this amplitude would provide a shorter circumferential length and therefore allow more feeders to be fitted to the machine. This work (discussed in Chapter 6) includes :

- (i) Designs of new types of latch needle to give reductions in latch motion.
- (ii) An application of a compound needle to a circular weft knitting machine.
- (iii) An application of a lifting sinker motion to a circular weft knitting machine.



CHAPTER 2

CAM DESIGN AND MANUFACTURE

CHAPTER 2

CAM DESIGN AND MANUFACTURE

Preamble

This chapter is concerned with the design and manufacture of the various cam systems used in Phase I of this work.

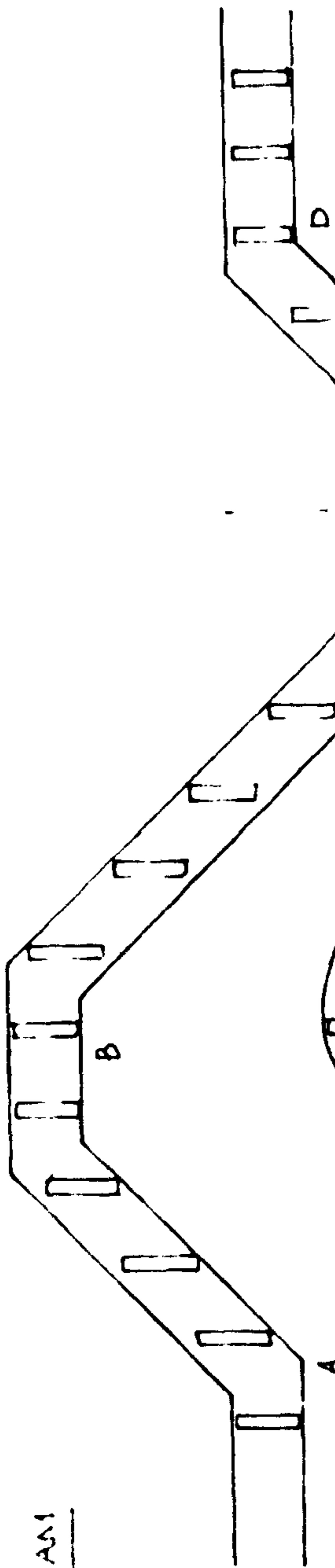
A description of the design and construction of three non-linear cam systems and a specially shaped "straight and curved" cam system will be given; these cams have controlled acceleration characteristics and are designed for high speed knitting.

In addition, two commercial-type linear cam systems have been constructed in order to compare their performance with that of the high speed, curved cam systems.

FIG. 5

LINEAR CAM

CLEARING HEIGHT



RUNNING POSITION

FIG. 6

NON-LINEAR CAM

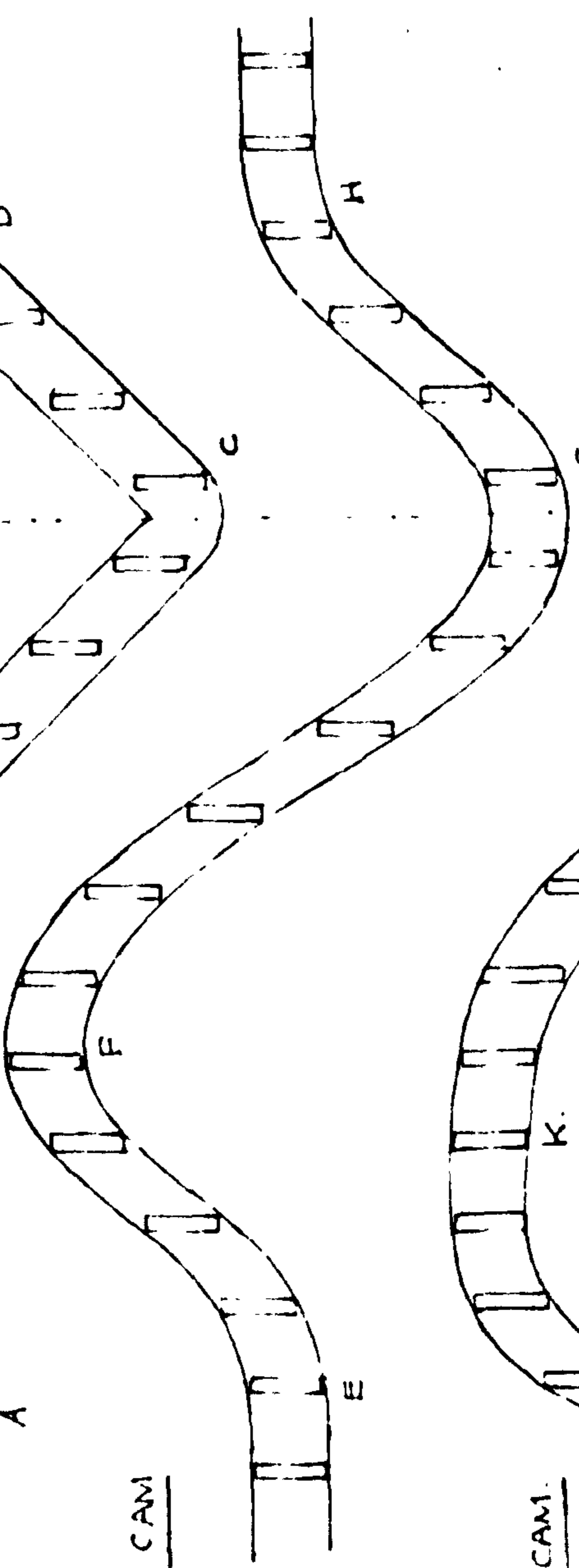
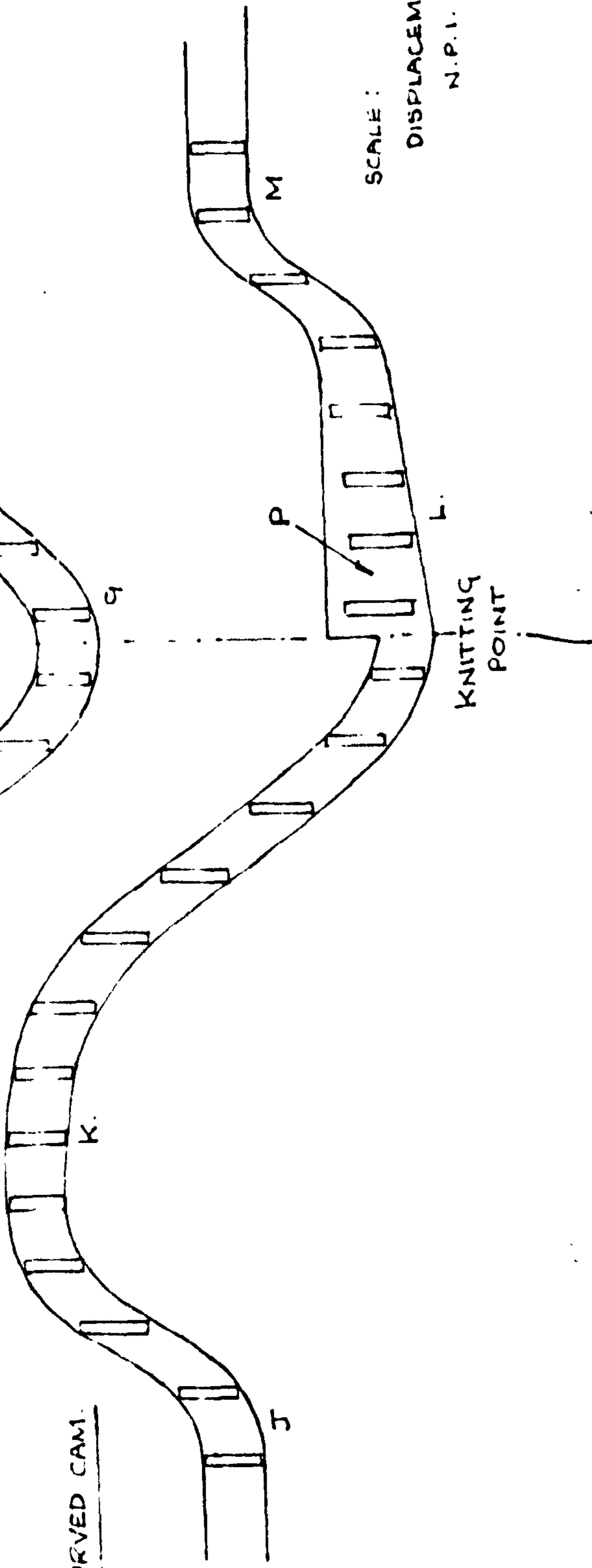


FIG. 7

ST. CURVED CAM



SCALE:

DISPLACEMENT 3

N.P.I. 1:6

KNITTING POINT

A. General Design and Special Requirements for High Speed Knitting Cam Systems

1. Non-linear cam system

It has been indicated in Chapter 1 that the linear cam system used on most commercial machines would be unsuitable for knitting at speeds greater than commercial speeds, since the positions where the needles change direction give instantaneous changes in acceleration (see Fig. 5 A, B, C, D) and this would eventually lead to needle butt breakage.

For high speed knitting, therefore, it was considered that a non-linear cam was required which would provide complete control over the needle butt acceleration; the proposed shape of a non-linear cam suitable for the knitting action is shown in Fig. 6.

Unfortunately when designing a non-linear cam it is insufficient to join straight lines with smooth curves, or even use circular arc curves, as these will not appreciably improve the acceleration characteristics of the cam system. If there is a sudden change in curvature  $1/\rho$  then there will be a sudden change in acceleration, since acceleration towards the centre of curvature is equal to  $v^2/\rho$ . Great emphasis is placed upon the importance of controlled acceleration, as this characteristic is closely connected with the performance of a cam (11). The acceleration is proportional to the force that acts on the cam system and, therefore, a poor acceleration curve will cause dynamic disturbances which will be detected by the needle butts, whereas a good acceleration curve will give smooth needle butt action.

It will be shown later in this chapter how the leading edges of a non-linear cam were designed mathematically to give the necessary controlled acceleration.

To take full advantage of a mathematical design, the non-linear cam must be made with an upper and lower profile designed to ensure that the needle butt is positively controlled throughout the knitting operation. With such a cam, however, it would not be possible to fit the separate adjustable stitch cam usually found on commercial machines, since this would immediately introduce the points of sudden acceleration and therefore restrict the machine speeds.

It was important, therefore, to investigate alternative methods of obtaining a range of stitch lengths, if the non-linear cams were to fulfil the requirements of a commercial machine. This aspect is discussed below.

(i) Positive feed as a stitch length controller

Some preliminary experiments were made with a  $45^{\circ}$  linear cam system on a Kirkland 8" diameter weft knitting machine, using positive feed as the means of altering the stitch length.

By feeding different known quantities of a 1/28 worsted yarn to the needles, at a fixed cam setting, it was possible to assess the range of stitch lengths that could be obtained under these conditions. The results are given in Table I and they show that a 26% range of stitch length can be obtained before the tension becomes high enough to break the yarn (i.e. above 25 gms), or low enough to cause dropped stitches (i.e. below 3 gms).

TABLE I

Supplied Stitch Length (ins)	Resulting Tension (gms)
0.189	3
0.166	7
0.147	13
0.139	25

It is to be expected that, when using a stronger man-made yarn, an even greater range of stitch length could be achieved, although a 26% range of stitch length represents a wide range in fabric construction and provides almost as wide a range as would normally be used on any one commercial knitting machine.

From these observations it seems possible that positive feed could be used to obtain a range of stitch length when using non-linear cams.

(ii) Dial and cylinder adjustments as a stitch length controller

There are possibilities that a high yarn tension, such as the 25 gms observed in Table I, when using positive feed as the stitch length controller, might reduce the life of the needles and cause wear on the cams. For instance, the high tensions would produce strains on the needle hooks as the needles moved down the stitch cam to form a loop and, at the same time, high forces would be acting on the butts which would tend to wear the cams.

If this proved to be a serious problem on the plain knitting machine (i.e. cylinder only), modifications to the design of the machine could be made to provide a vertical adjustment of the cylinder with respect to the knitting cams. This vertical adjustment would enable a range of stitch length to be obtained.

If the knitting machine were of the dial and cylinder type then no modifications to the machine would be necessary, as a range of stitch length could be achieved by an adjustment of the height of the dial above the cylinder - a facility which is already provided on these machines.

## 2. "Straight and curved" cam system

An important aspect of non-linear cams is that they must be constructed with carefully chosen techniques to give a high degree of accuracy, if the conditions of controlled acceleration are to be achieved. It was realised that the most critical points were those where the needle butts meet the stitch cam and upthrow cam (i.e. F and G see Fig. 6), since a discontinuity in the acceleration curves at these points would result in high impact forces of the needle butts on the cam track.

In view of the possibility of the commercial machine builders not having the time or facilities to achieve the necessary high accuracy, it was considered to be worthwhile investigating an additional type of curved cam, whose performance would not be completely dependent upon the accuracy with which it was made.

If a cam system were fitted with low-angled ramps at the points F and G, then the impact forces on the needle butts would be low and consequently, the performance of the cam would not be seriously affected; the remaining portions of the cam system would be designed with mathematically curved shapes. The proposed shape of this cam is shown in Fig. 7 with low angles at K and L, and will be described throughout the thesis as a "straight and curved" cam.

Experiments with a linear cam, discussed in more detail in Chapter 5, showed that the needle butts were held against the upper face of the stitch cam by the tension in the yarn during loop formation, and this tension also caused the butts to move upwards after the knitting point by approximately  $1/32"$ .

It was considered that if the portion after the knitting point were cut away, to allow the needle butts to take up their natural position, the forces acting on the butts would be reduced. A relieved portion was therefore incorporated in the design of the "straight and curved" cam (Fig. 7).

It may be observed that since this cam system has low-angled, straight line ramps at the top and bottom of the stitch cam portion, it would be possible to incorporate a separate adjustable stitch cam. The mathematical design and construction of the "straight and curved" cam is shown later in this chapter.

### 3. Some Aspects concerning both Non-Linear and "Straight and Curved" Cams

#### (1) Circumferential length

The obvious disadvantage of non-linear or "straight and curved" cams is that for a given maximum cam angle, a curve requires a longer circumferential length to



produce a particular vertical displacement of the needle than does a linear cam. However, since the acceleration would be controlled in the non-linear cams and in the curved portions of the straight and curved cam, much higher maximum knitting speeds were anticipated, and these increases in speed might be sufficient to provide significant increases in rates of fabric production.

The straight and curved cam system has a disadvantage when compared with the non-linear cam, in that the low-angled ramps would necessitate an even longer circumferential length. However, it was hoped that the increased speeds achieved through having low sudden acceleration forces would still allow substantial increases in rates of production to be made over those of the linear cams, even though these increases would not be as high as those of the non-linear cams. On the other hand, it was considered possible that poor accuracy, when manufacturing the non-linear cams, would lead to the straight and curved cam becoming a better high speed cam.

(ii) The Life of the Cams and the Needles

Other important aspects of performance, which depend to some extent on the shape of the cam system, are those of wear on the cams and the life of the needles.

It is well known that at the points of the commercial cam which have high sudden acceleration forces, the needles tend to bounce causing wear (12) and even grooves in the cam. Also the continuous bouncing of the needle

eventually weakens the butt and it breaks off. It was hoped that controlled acceleration cam systems would reduce this wear on the cams and also extend the life of the needle butts.

The other parts of the needle which are continually being overworked are the latch, the latch pivot and the hook. The commercial cam system, with its points of high sudden acceleration, offers no real control over the motion of the latch; consequently the continual flinging open of the latch and rapid closing of the latch on to the hook can eventually lead to bent latches and hooks and latch pivot fatigue.

It was hoped that the controlled acceleration cam systems would give smoother latch movement and subsequently give a longer needle life.

(iii) Special Facilities for stitch length adjustments

For a complete set of experimental results, it was considered important to be able to set the non-linear cam systems at different vertical heights and hence, obtain a range of stitch lengths in this way as well as by using positive feed as a stitch length controller. All the cams were designed, therefore, to fit to the same cam section which was slotted to allow for this vertical adjustment.

The straight and curved cam was intended as a cam system that had individual stitch cam adjustment. However, for the purpose of this work it was more convenient to make the cam with complete upper and lower tracks, (i.e. no separate pieces) and obtain a range of

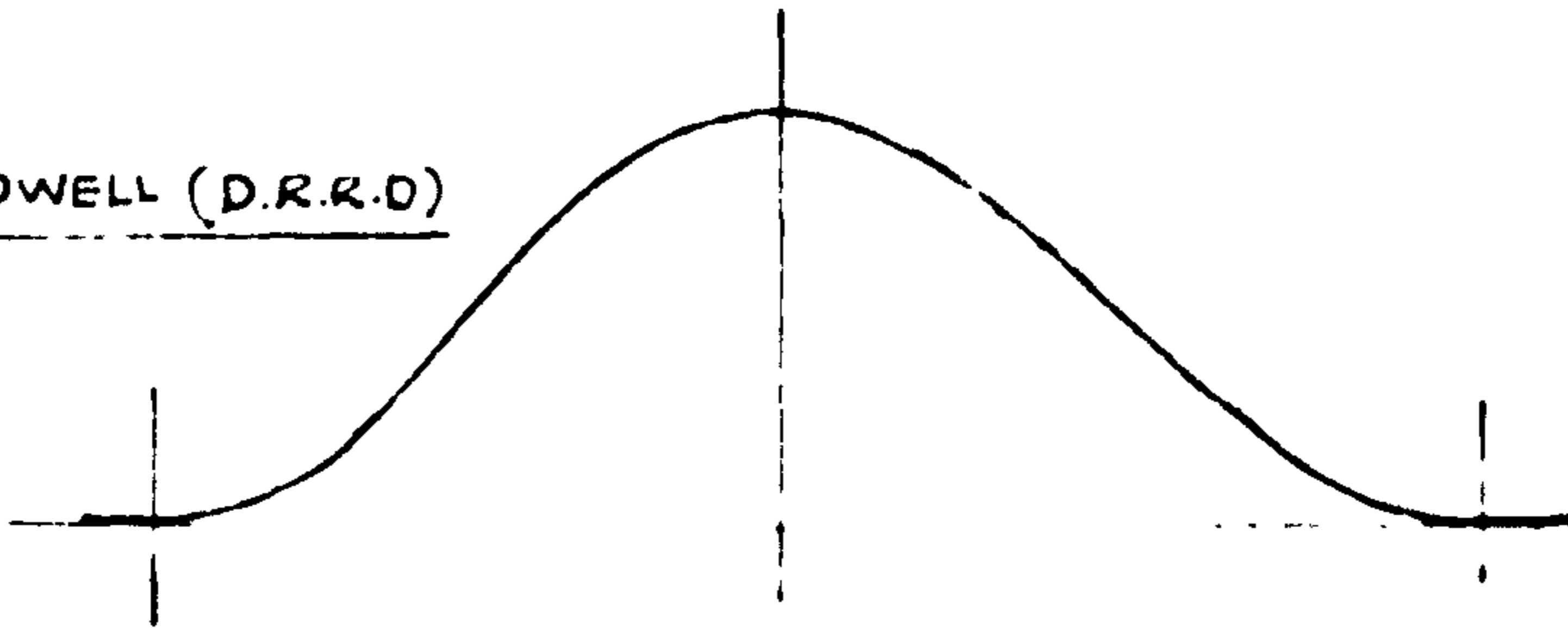
Fig 8

VARIOUS DISPLACEMENT MOTIONS

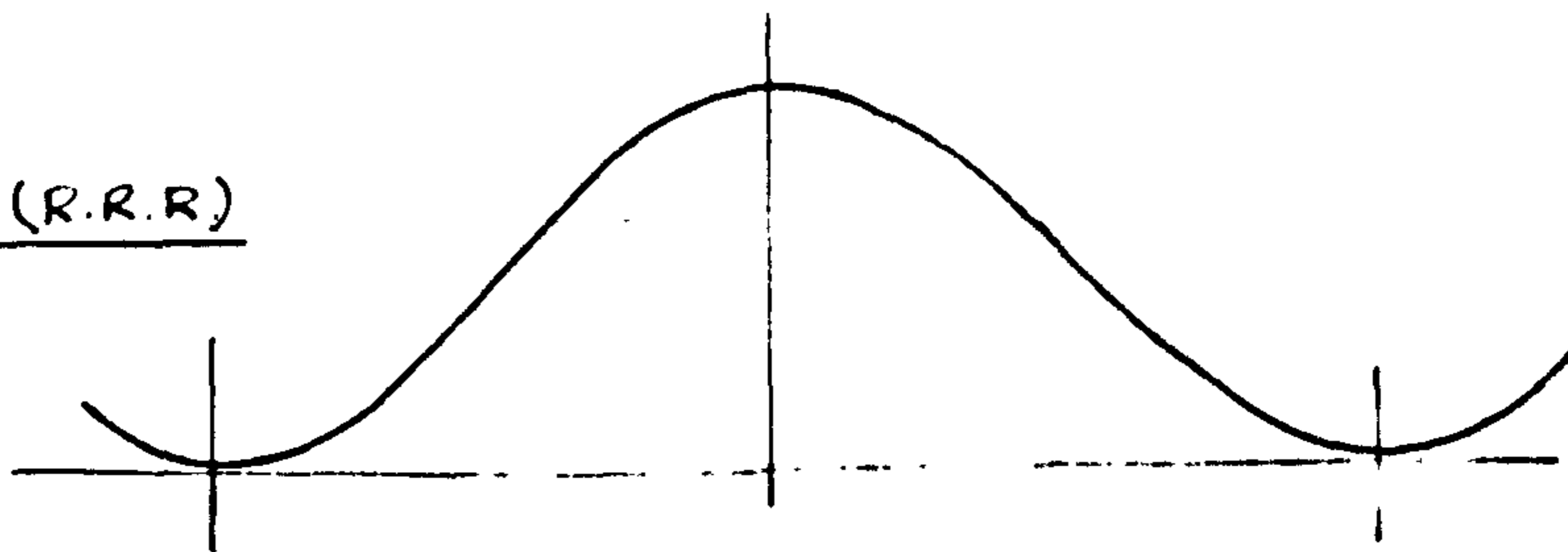
DWELL - RISE - DWELL (D.R.D)



DWELL - RISE - RETURN - DWELL (D.R.R.D)



RISE - RETURN - RISE (R.R.R)



stitch lengths in the same way as with the non-linear cams. If the straight and curved cam proved to be successful as a high speed cam, only small modifications to the design would be necessary to convert this cam to give a separate adjustable stitch cam, if this were required. These modifications would be unlikely to affect the performance of the cam.

## B. General Cam Design

A complete study of general cam design has been made by Rothbart (11) and this proved invaluable in finding the most suitable mathematical curve for the non-linear and straight and curved knitting machine cam systems.

An examination of some well known basic and advanced curves is shown below, and it will be illustrated that the suitability of a particular curve depends to a large extent on the shape of the acceleration curve, and also on the actual displacement motion required.

The various displacement motions which are examined in this section are summarized below and shown in Fig. 8.

Dwell-rise-dwell (D.R.D.).

Dwell-rise-return-dwell (D.R.R.D.).

Rise-return-rise (R.R.R.).

### 1. Characteristics of Some Basic Curves

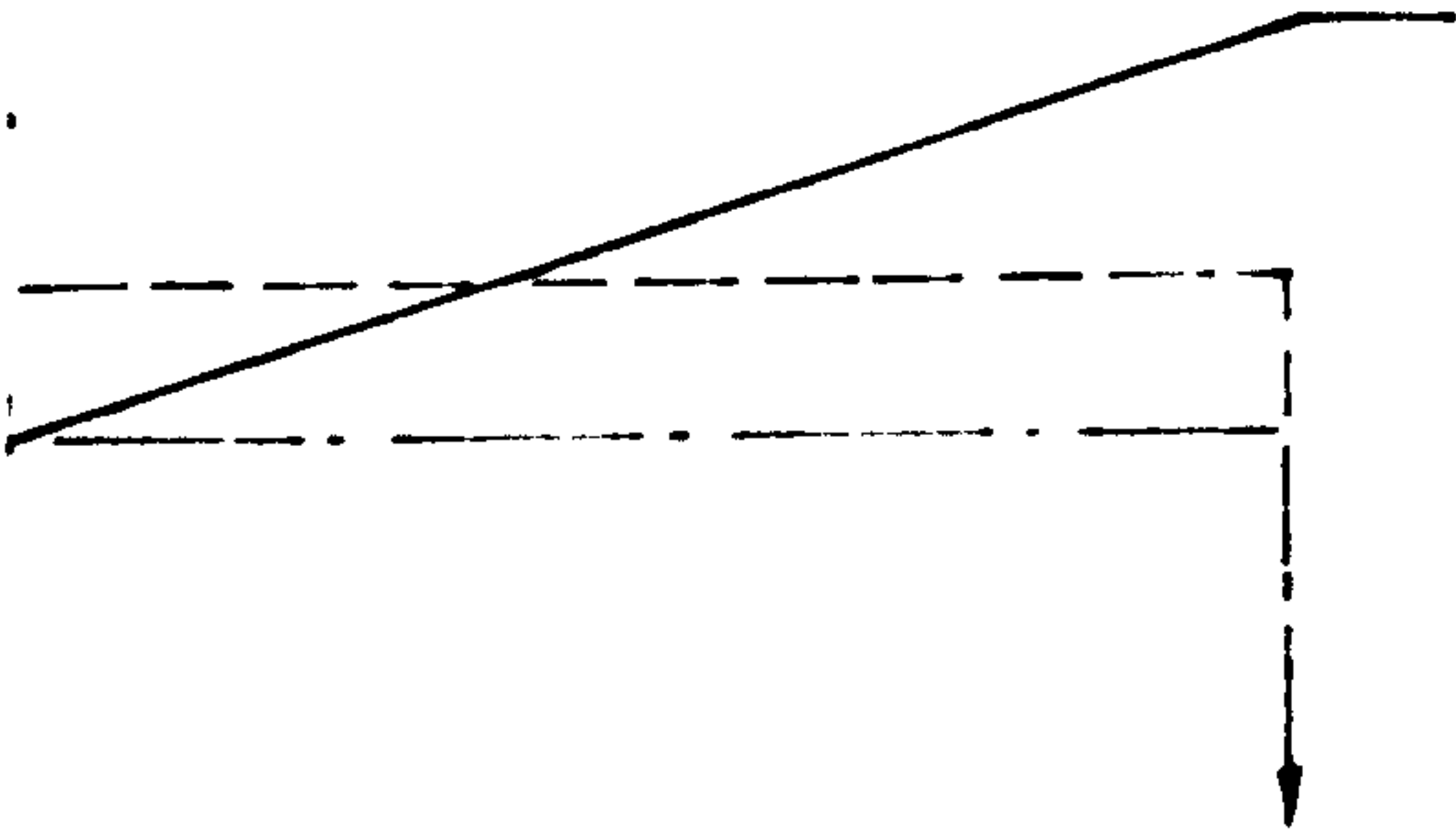
#### (a) Dwell-Rise-Dwell Curves

##### (i) Linear or Constant Velocity Curve

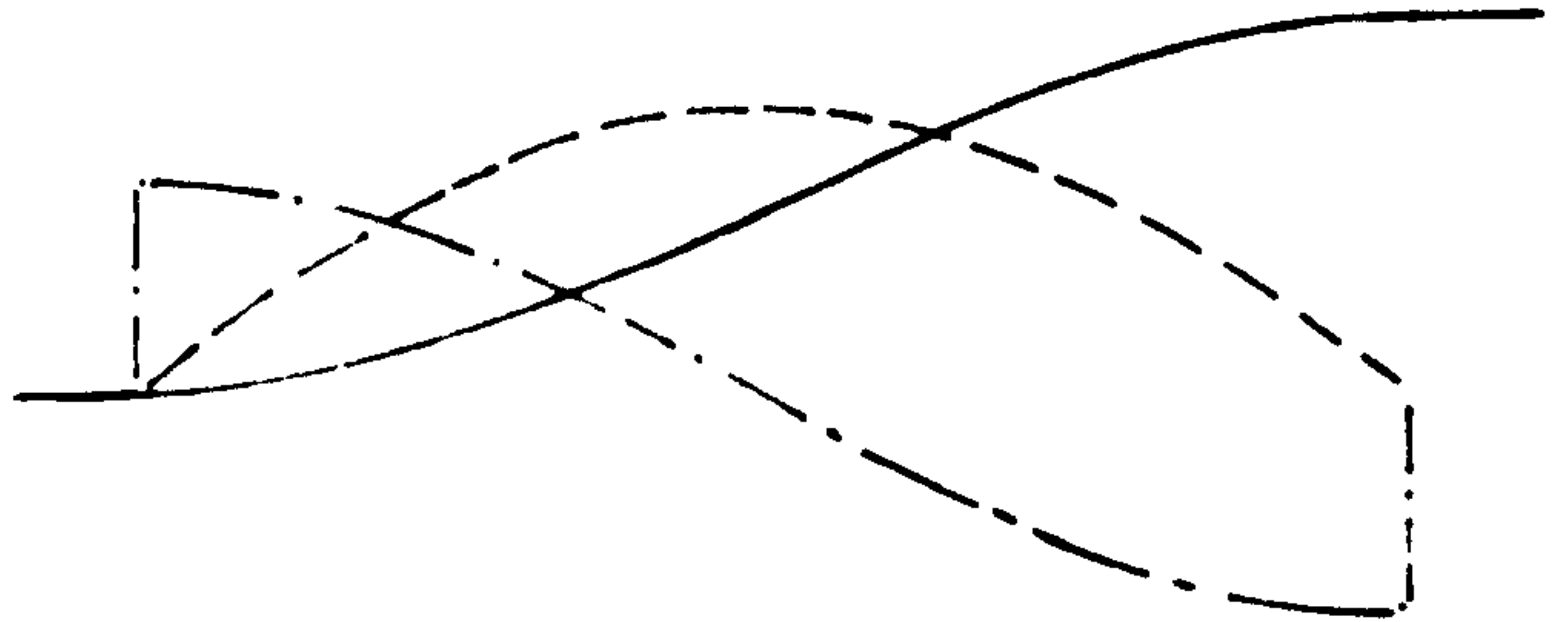
It has already been emphasised that the linear cam is normally used for the cam design of the loop-forming action in knitting machinery. This is a simple

CHARACTERISTICS OF VARIOUS DWELL-RISE-DWELL CURVES

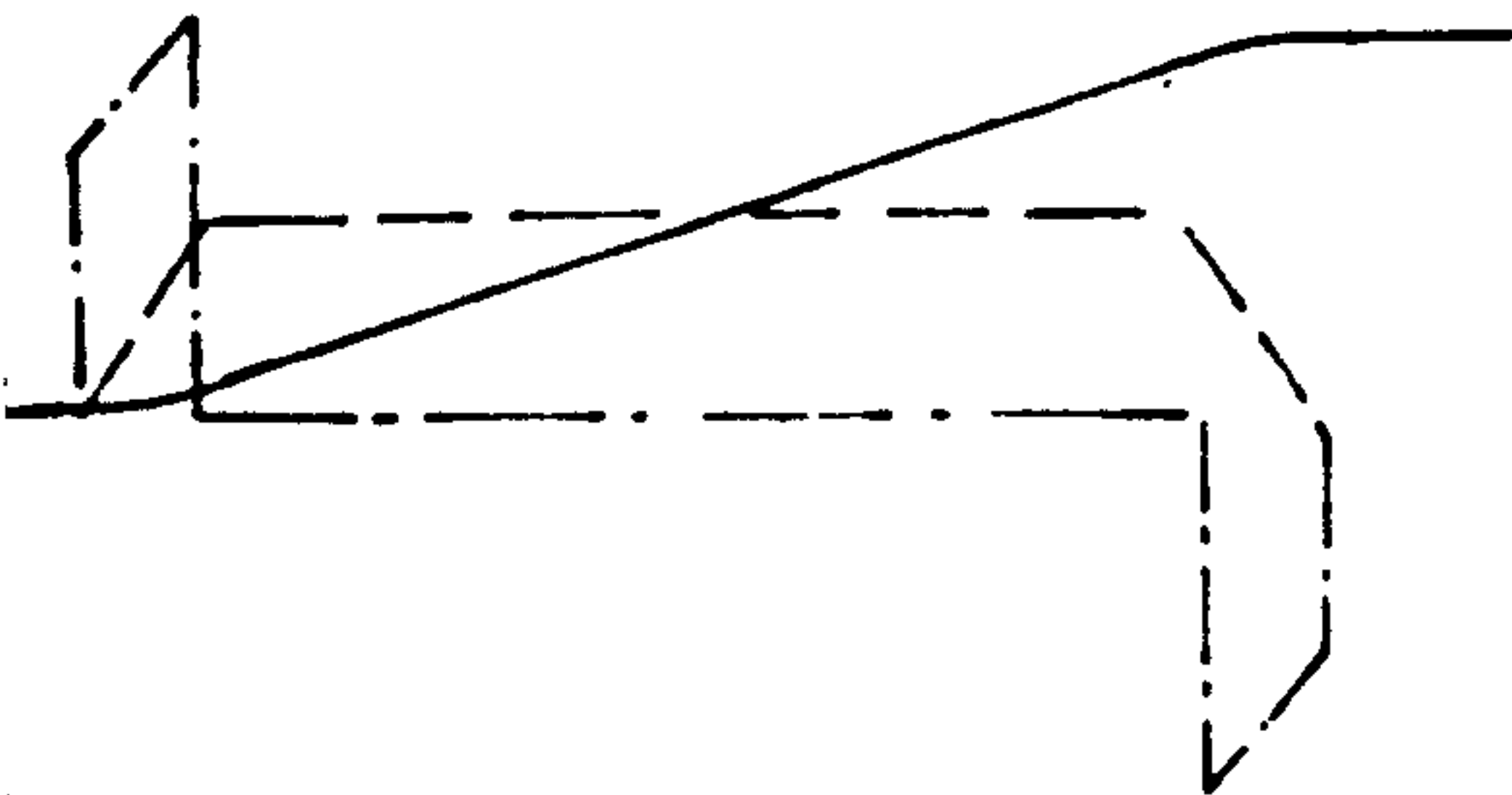
a) LINEAR



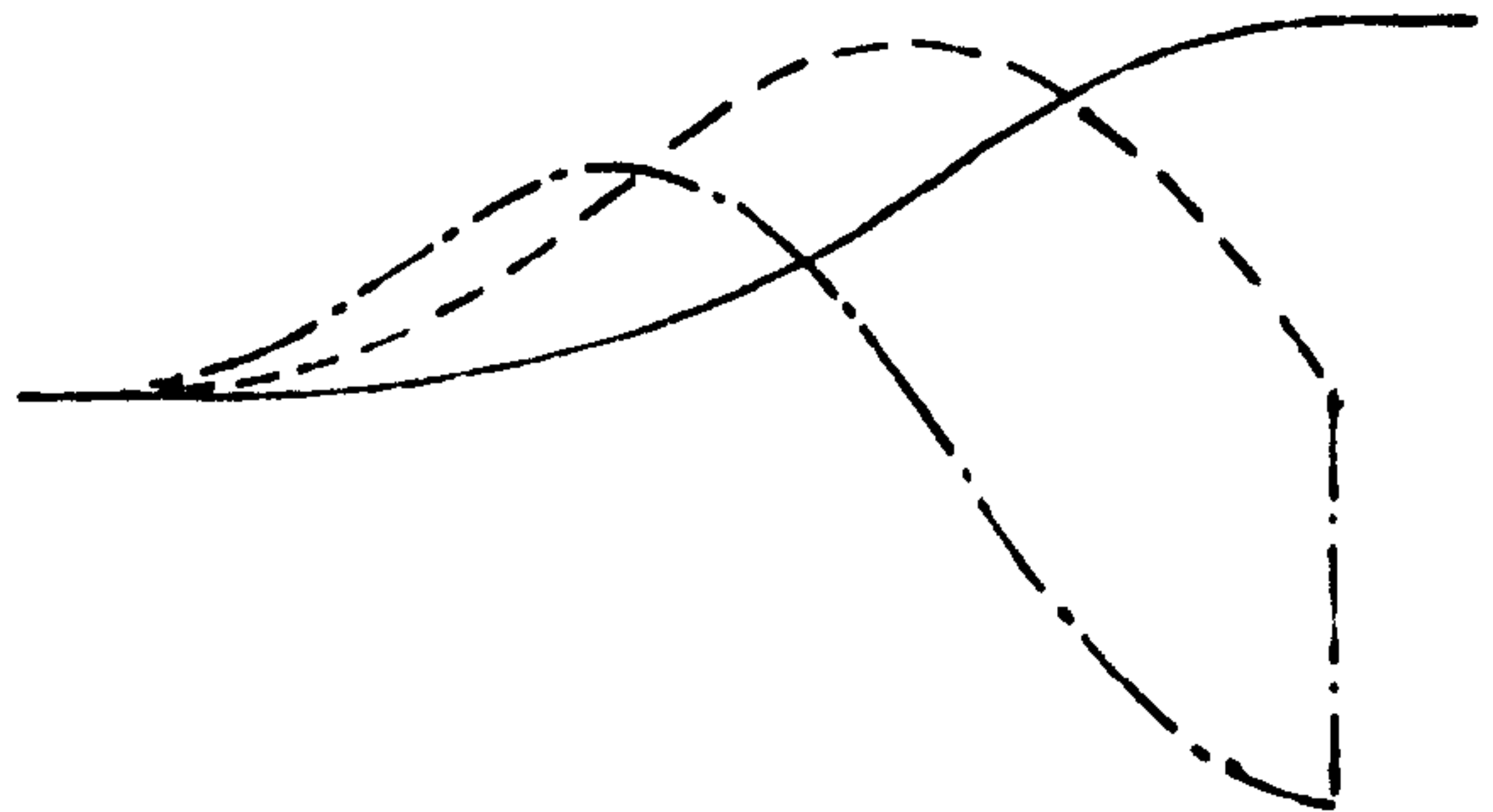
(d) SIMPLE HARMONIC



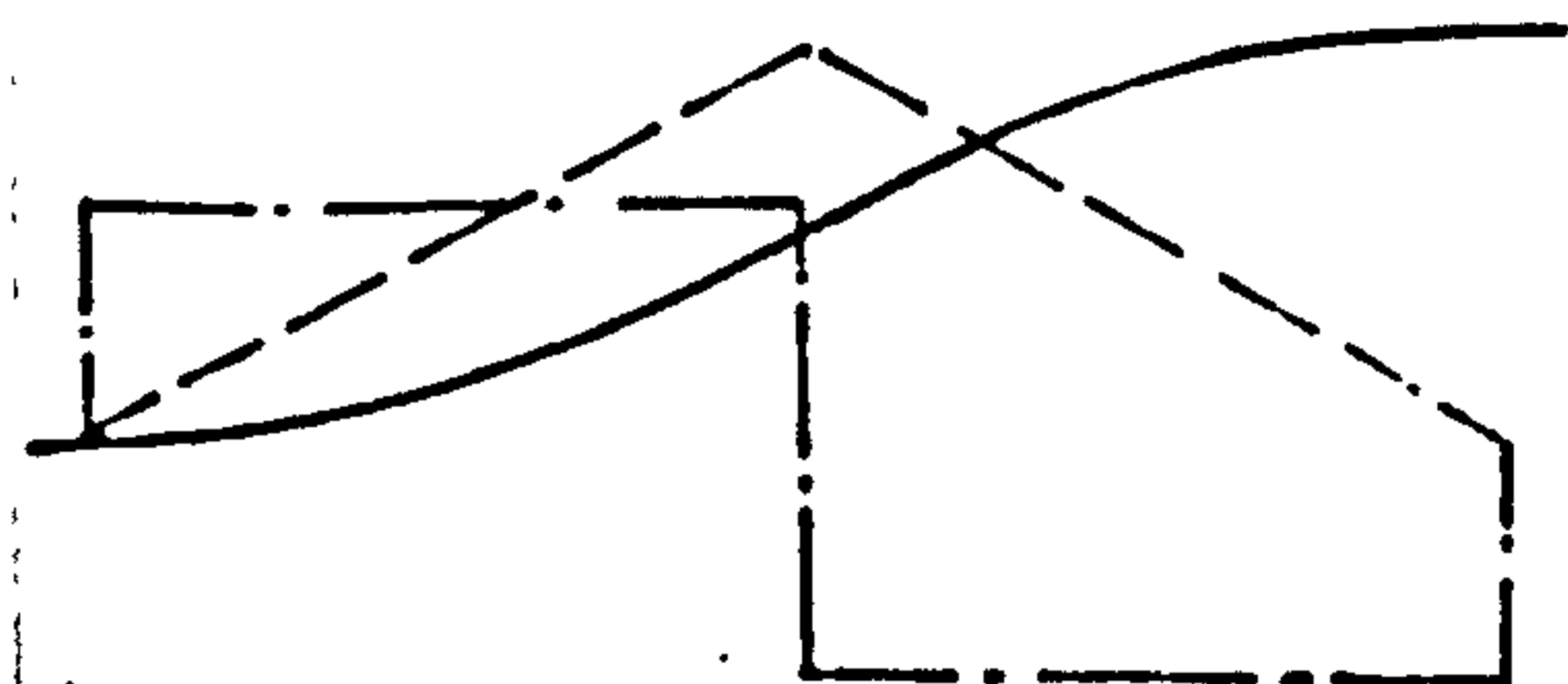
b) STRAIGHT-LINE CIRCULAR ARC



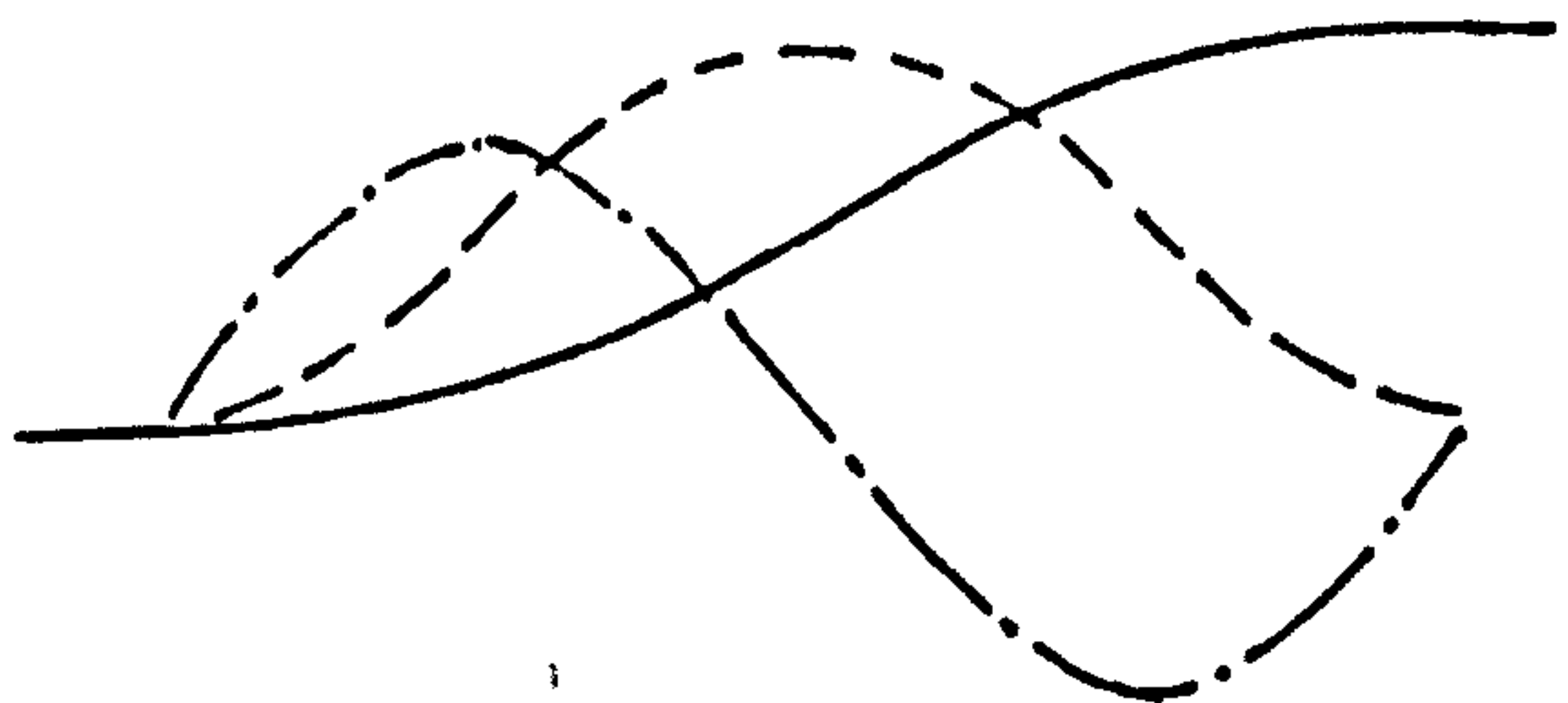
(e) DOUBLE HARMONIC



c) PARABOLIC



(f) CYCLOIDAL



\_\_\_\_\_ DISPLACEMENT  
 - - - - - VELOCITY  
 - . - . - ACCELERATION

polynomial curve giving the smallest length for a given rise. . . During the rise the rate of change of displacement is uniform, the velocity constant, and the acceleration is zero. At the ends of the curve there is an instantaneous change in velocity giving a theoretically infinite acceleration which transmits a large shock to the needle butts. This curve can only be used at low speeds with small maximum angles (a, Fig. 9).

(ii) Straight-Line Circular Arc Curve

This curve is also used for the cam design of the loop-forming action in commercial knitting machinery. It has radii between the dwell and rise giving improvement over the straight line curve. However, there are still large acceleration values at the ends of the curve, which prevent it from being used at much higher speeds than the linear cam. (b, Fig. 9).

(iii) Parabolic, Constant Acceleration Curve

This polynomial curve has constant positive and negative acceleration with a lower maximum acceleration value than any other curve. This often misleads designers into thinking that it is the best curve, whereas due to the sudden changes in acceleration at the ends and at the transition point - or point of maximum angle - which cause vibration and wear, it is probably one of the worst curves for high speeds. It is only suggested for moderate or low speed cams. (c, Fig. 9).

(iv) Simple Harmonic Motion Curve

This trigonometric curve is often used because it is simple to understand and construct. The basis of the harmonic curve is the projection (on a diameter), of the constant angular velocity movement of a point on the circumference of a circle. The movement is similar to that of a swinging pendulum. This curve is an improvement over the above curves, but since the change in acceleration is abrupt at the ends this design of cam is unsuitable for high speeds. (d, Fig. 9).

(v) Double Harmonic Motion Curve

This curve is composed of the difference between two harmonic curves, one being one-quarter of the amplitude and twice the frequency of the other.

The rate of acceleration change at the beginning of the displacement curve is smaller than that of the simple harmonic motion curve, but it requires a longer cam to give the same vertical displacement. Also there is a larger sudden change in acceleration at the end of this curve than in the simple harmonic motion curve, which prevents it from being used for high speeds. However, it is frequently used in dwell-rise-return-dwell cams as there are then no abrupt changes in acceleration and it can therefore be used at high speeds in this instance. (e, Fig. 9).

(vi) Cycloidal Curve

This trigonometric curve is produced from the locus of a point on a circle which is rolled on a straight line. There are no sudden changes in acceleration at

Fig. 10

DWELL - RISE - RETURN - DWELL CURVE

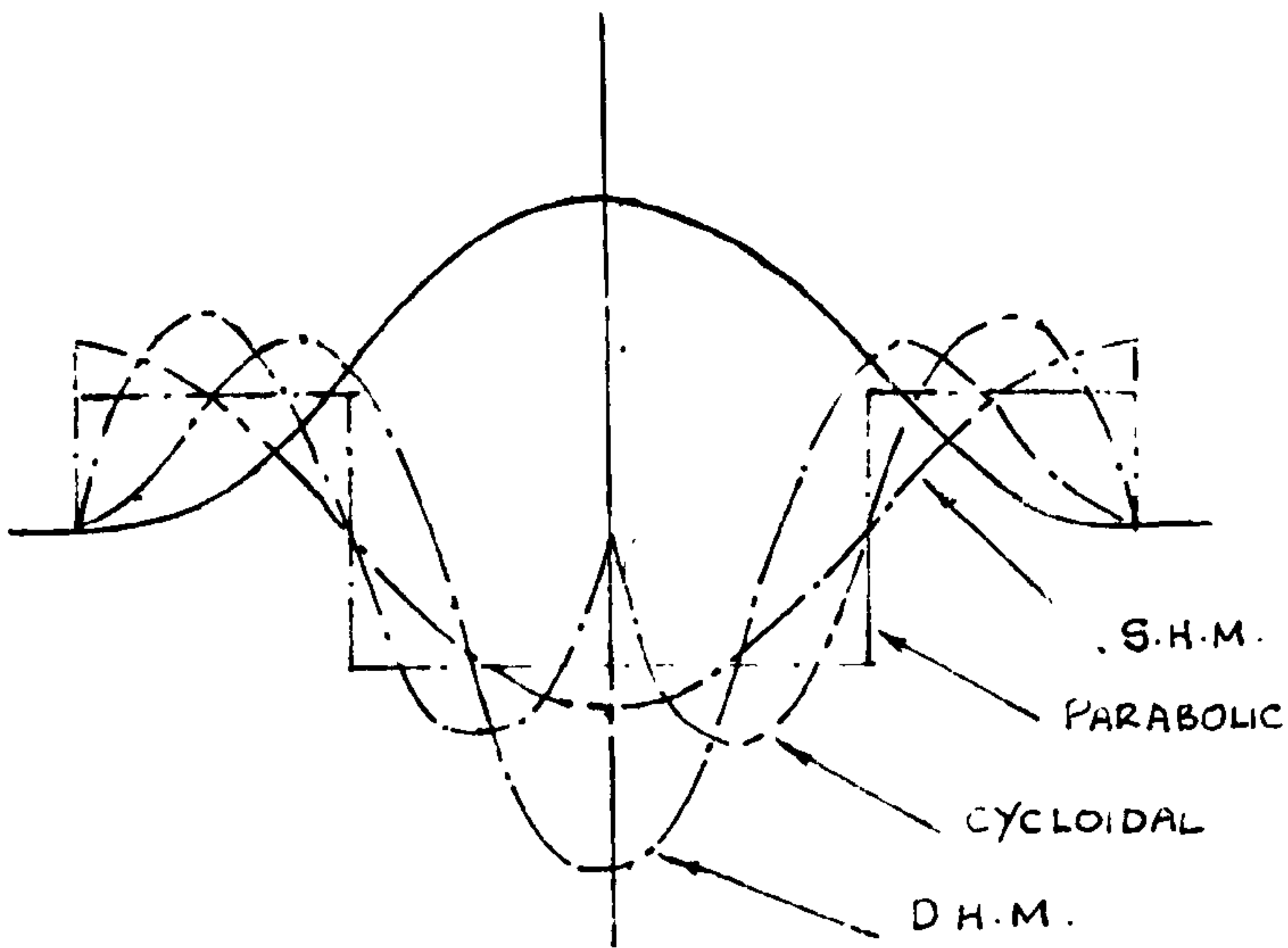


Fig. 11

RISE - RETURN - RISE CURVE

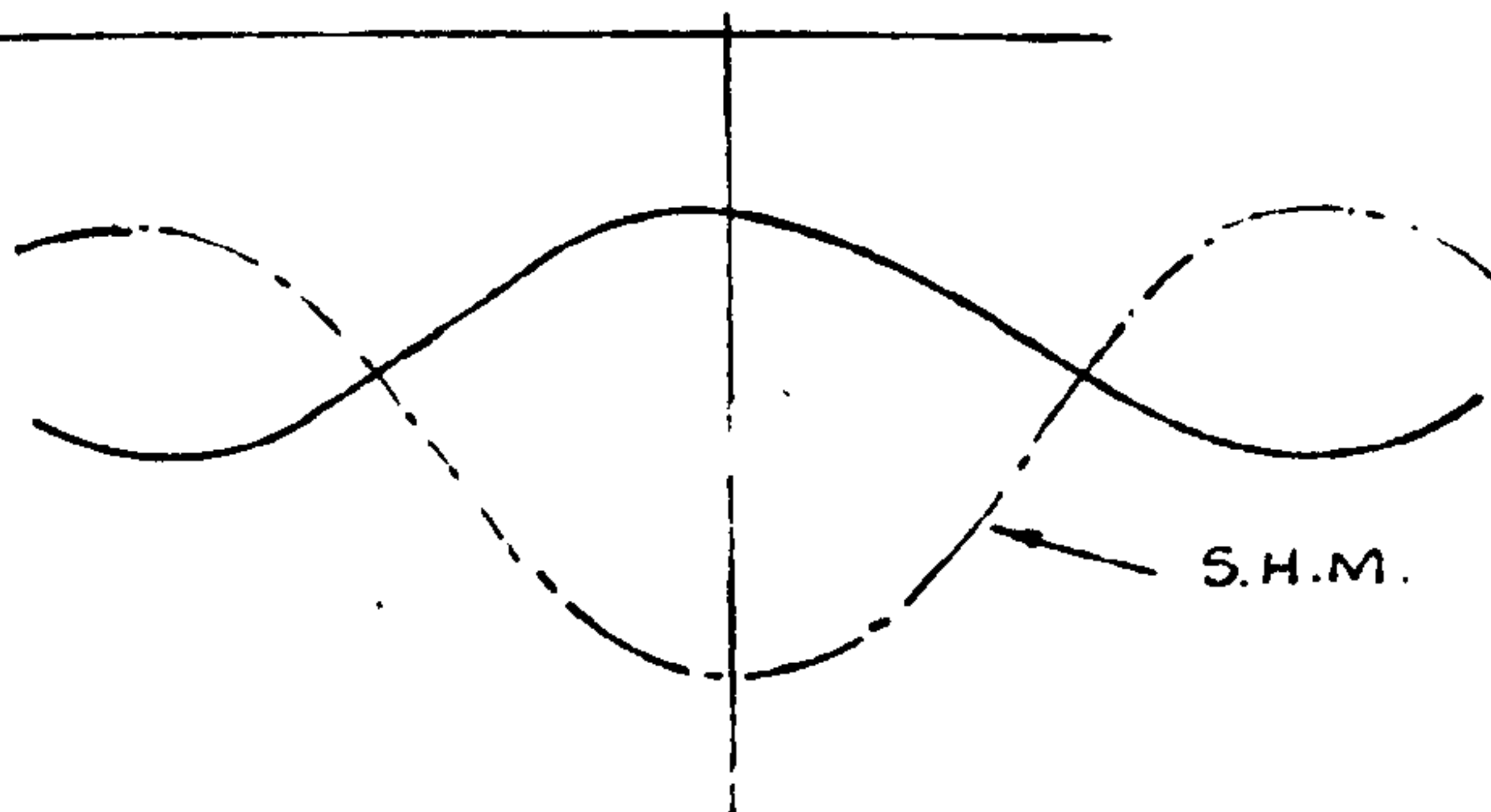
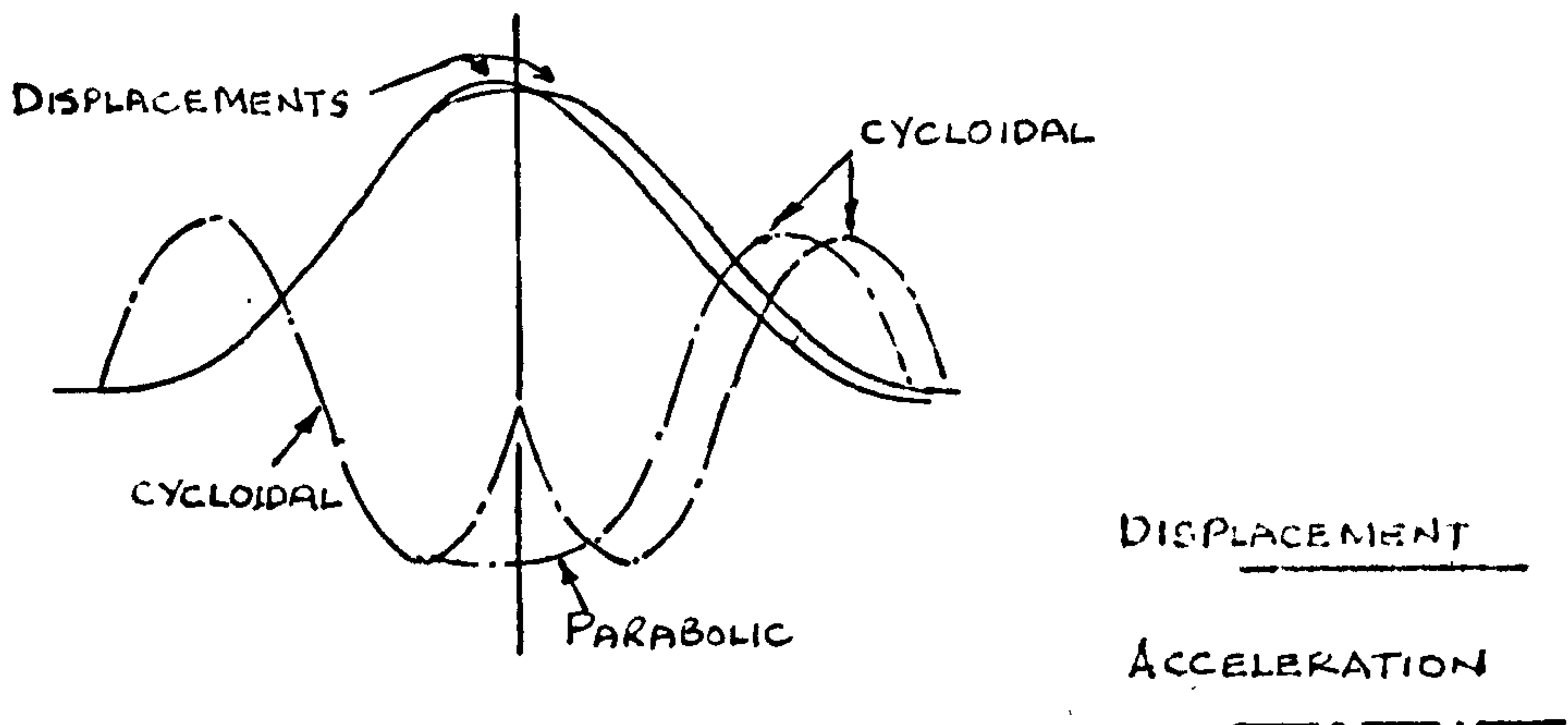


Fig. 12

COMBINATION OF BASIC CURVES





the ends of this curve, and if these ends can be manufactured accurately it is the best cam for high speeds. (f, Fig. 9).

(vii) Summary of the Characteristics of the Basic Dwell-Rise-Dwell Curves

From the brief description of the basic D.R.D. curves it is observed that all the curves, apart from the cycloidal, produce sudden acceleration at the ends of the cams causing high shock, vibration and wear, which makes them unsuitable for high speeds. The cycloidal cam, however, has a smooth, continuous acceleration curve and therefore is the most suitable for high speeds. It will be shown later how this curve was used in the design of the "straight and curved" cam.

(b) Basic Curves Applied to Dwell-Rise-Return-Dwell and Rise-Return-Rise Motion

(i) D.R.R.D. Curves

An interesting feature when basic curves are applied to the D.R.R.D. motion is that the cycloidal curve, which has been shown to be the best dwell-rise-dwell cam, now exhibits a sudden change in the acceleration curve at the maximum rise point. (Fig. 10). This gives a sudden change in pulse and makes it unsuitable for high speeds in dwell-rise-return-dwell cams, since vibration and hence wear would occur at the maximum rise point (11).

The simple harmonic motion and the parabolic curves have continuous acceleration at the maximum rise point, but they still exhibit the sudden change in acceleration at the ends.

The most suitable curve for symmetrical D.R.R.D. cams is the double harmonic curve which now has a continuous acceleration curve throughout the motion, and later it will be shown how this curve was also used in the design of the "straight and curved" cam.

(ii) Symmetrical Rise-Return-Rise Cams

This motion (Fig. 11) has no dwells and a simple harmonic curve can be used as this will give continuous displacement, velocity and acceleration characteristics. This shape could be used to produce a multi-feed, high speed cam system.

2. Characteristics of Advanced Curves

Basic curves are easy both to construct and analyse and are, therefore, frequently used in cam design. However, with stepped cams or unsymmetrical cams, such as the knitting cam, basic curves are often inadequate, as the characteristic curves of displacement, velocity and acceleration may require modification. In these cases more advanced curves are necessary.

When designing moderate to high speed cams, irrespective of whether the cam is a symmetrical D.R.D. cam or a complicated unsymmetrical stepped cam, the characteristics of the cam are usually selected so as to provide :

- (i) continuous displacement, velocity and acceleration;
- (ii) lowest maximum acceleration value;
- (iii) rate of change of acceleration (i.e. Pulse) which is not too large.

A finite pulse is a critical factor in designing high speed cam systems. An infinite pulse (i.e. a discontinuous acceleration curve) could lead to serious vibration effects of the needle butt motion in the cam system. The aim for good performance is to have a finite pulse with a value which is not too large.

Advanced curves are considered here in two ways :

- (a) Combination of Basic Curves.
- (b) Polynomial equations.
- (a) Combination of Basic Curves

Any combination of basic curves may be used to give the good acceleration characteristics necessary for high speed cams. When combining basic curves, the curved portions must be joined with the same slope (i.e. tangentially) to give continuous displacement and velocity curves - with high speed cams the acceleration curve should also be continuous.

A further examination of the cycloidal D.R.R.D. cam will show the effect of using combinations of basic curves.

It was stated that the sudden change in acceleration curve at the maximum rise point would prevent this curve from being used at high speeds. This sudden change in the acceleration curve can be eliminated, however, by combining the parabolic curve with the cycloidal curve at the maximum rise point (see Fig. 12). To do this the desired boundary conditions of vertical acceleration are fitted into the formulae for parabolic and cycloidal curves, and the new curve portion is then calculated. In the case of unsymmetrical cams or stepped cams, the mathematics become very involved, when using the method of combinations of basic curves, as many simultaneous equations have to be solved.

APPLICATION OF POLYNOMIAL EQUATIONS TO CAM DESIGN

FIG. 13

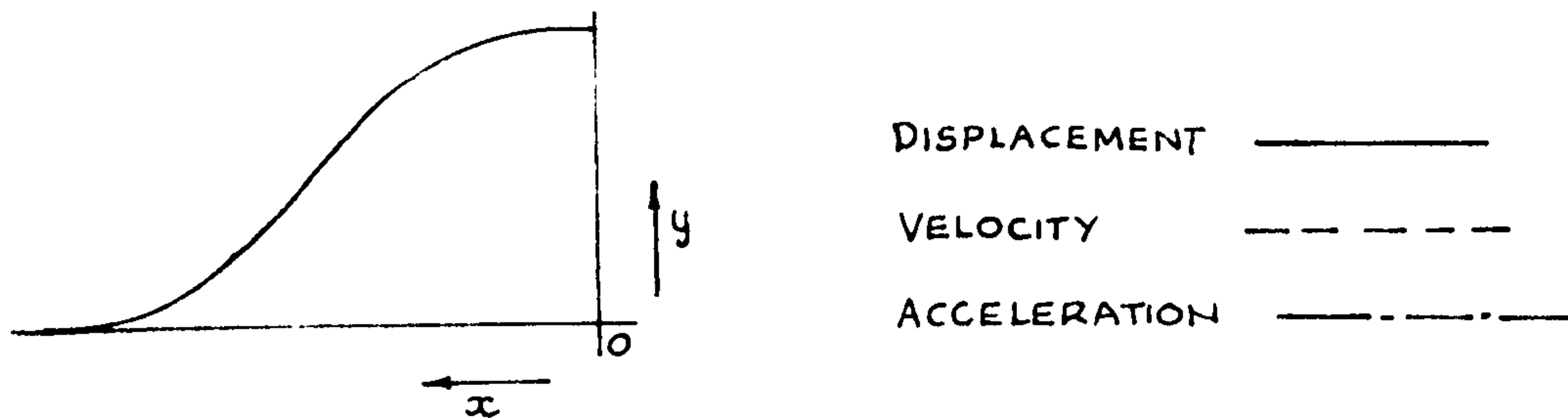
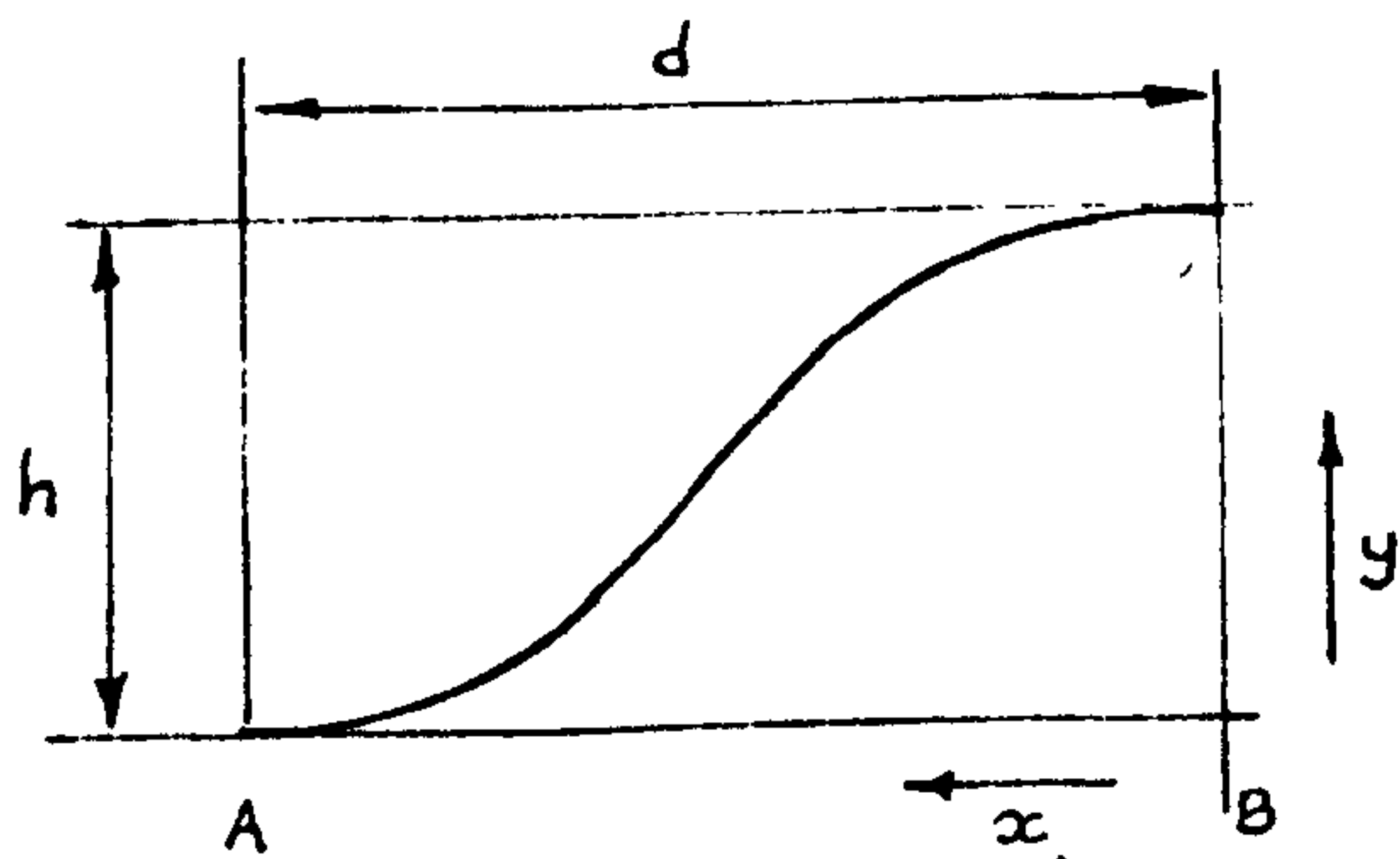


FIG 14. (i)



(ii)

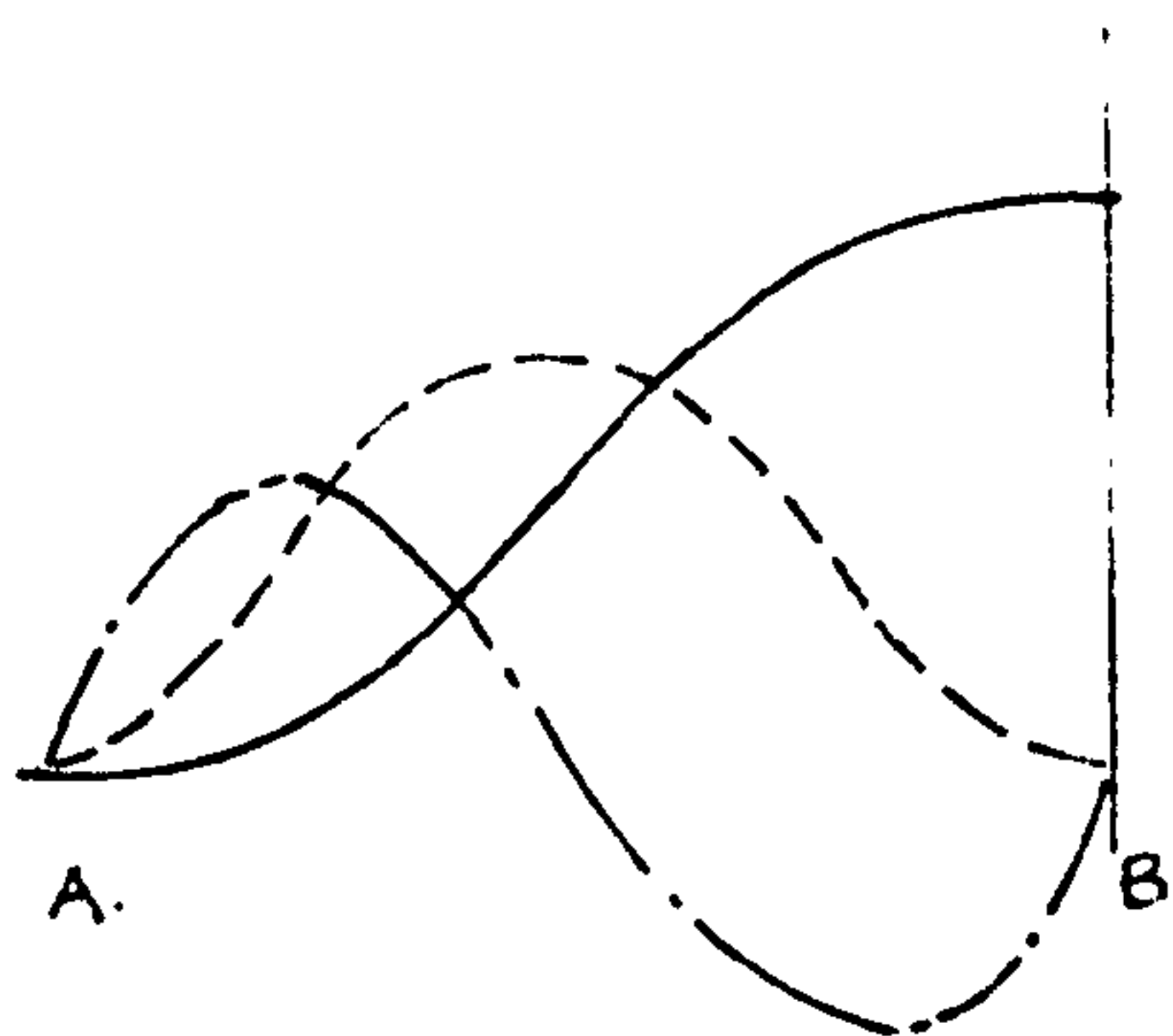
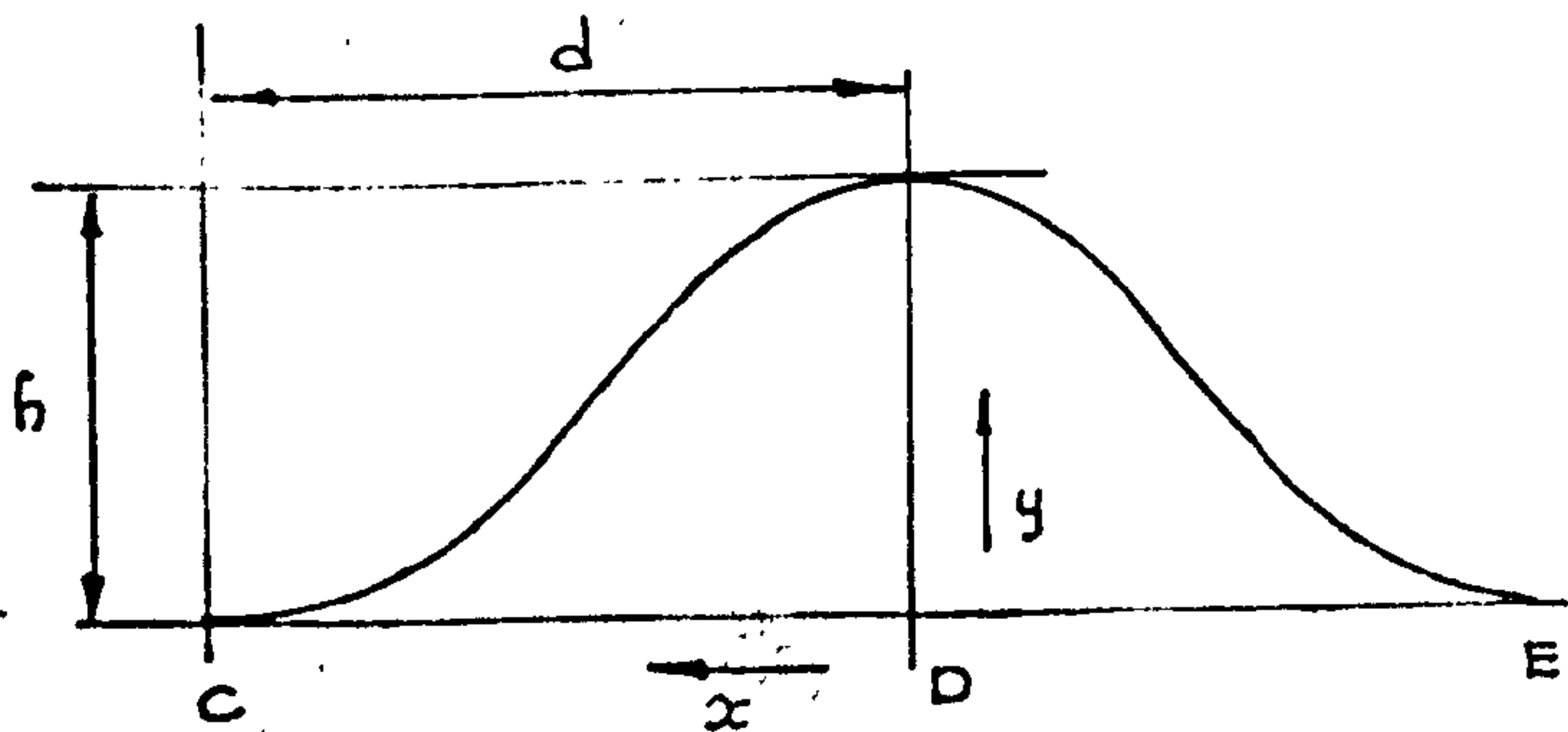
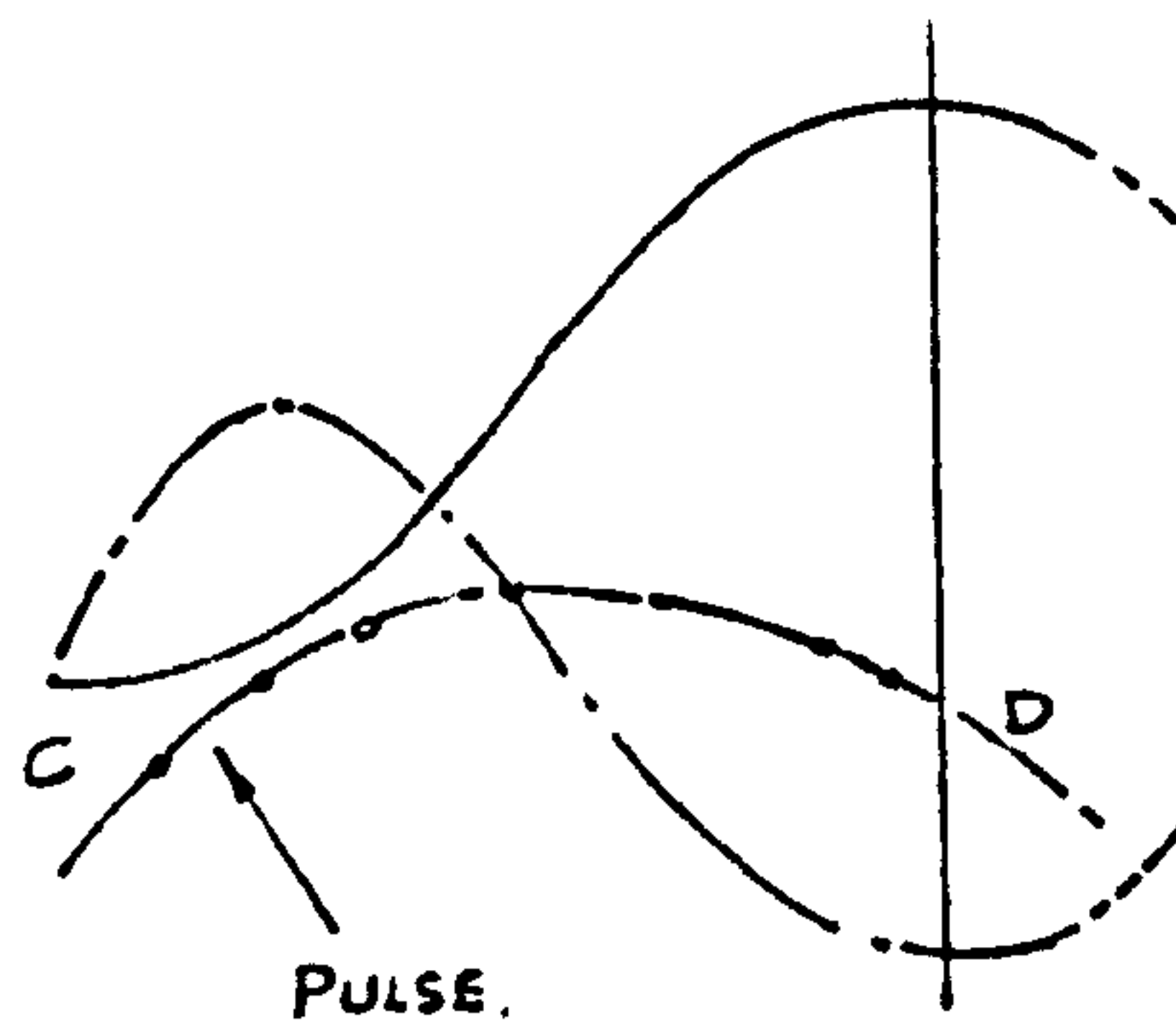


FIG. 15 (i)



(ii)



(b) Polynomial Equations

The method of solution by polynomial equations is more appropriate for unsymmetrical cam design, since this method is easier, more direct and more accurate than the method of combining basic curves (13).

An examination of Fig. 13 shows a diagram of the displacement curve for a dwell-rise-dwell cam, where the horizontal displacement is  $x$  and the vertical displacement is  $y$ . The displacement curve is defined by the polynomial equation of the form :

$$y = C_0 + C_1x + C_2x^2 + \dots + C_nx^n \quad \dots (1)$$

The vertical velocity ( $dy/dt$ ), acceleration ( $d^2y/dt^2$ ), and pulse ( $d^3y/dt^3$ ) of the needle are therefore given by

$$\text{Vertical velocity} = \frac{dy}{dt} = \frac{dy}{dx} \times \frac{dx}{dt} = v \frac{dy}{dx}$$

where  $v$  is the constant horizontal velocity of the needle.

$$\text{Vertical acceleration} = \frac{d^2y}{dt^2} = v^2 \frac{d^2y}{dx^2}$$

$$\text{and Vertical pulse} = \frac{d^3y}{dt^3} = v^3 \frac{d^3y}{dx^3}$$

Because  $v$  is a constant, the boundary conditions may be written in terms of derivatives with respect to  $x$ , which are easily calculated from a polynomial equation of the type (1). For convenience, derivatives with respect to  $x$  will be denoted by primes, e.g.  $y' = \frac{dy}{dx}$ .

It will be shown later that the characteristics of the polynomial curve depend upon which boundary conditions are chosen for control. The number of boundary conditions used

determine the number of terms in the polynomial equation and, as the number of terms becomes higher the initial and final displacement of the curve becomes slower ; greater accuracy is therefore necessary at the end points of the curve.

(i) Dwell-Rise-Dwell Curve using Polynomials

Any curve can be duplicated closely, mathematically, using polynomial equations; an example of this is given below in which polynomial equations have been used to give curve characteristics that are very similar to the cycloidal curve.

An examination of the cycloidal D.R.D. curve (f, Fig.9) has shown that the velocity and acceleration values are zero at the ends of the curve, this being extremely important for a high speed cam system. To provide similar curve characteristics when using polynomial equations, boundary conditions of vertical velocity and vertical acceleration equal to zero must be used in the calculation.

From an examination of Fig. 14(i), which illustrates a D.R.D. displacement motion, the necessary boundary conditions at positions A and B may be written down.

$$\text{At B, } x = 0, y = h, y' = 0, y'' = 0.$$

$$\text{At A, } x = d, y = 0, y' = 0, y'' = 0.$$

$$\text{Assume } y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5.$$

$$\text{Then, } y' = C_1 + 2C_2x + 3C_3x^2 + 4C_4x^3 + 5C_5x^4;$$

$$y'' = 2C_2 + 6C_3x + 12C_4x^2 + 20C_5x^3.$$

Substituting the boundary conditions and solving the resulting equations gives

$$y = h - \frac{10h}{d^3} x^3 + \frac{15h}{d^4} x^4 - \frac{6h}{d^5} x^5$$

Therefore

$$\frac{y}{h} = 1 - 10 \left(\frac{x}{d}\right)^3 + 15 \left(\frac{x}{d}\right)^4 - 6 \left(\frac{x}{d}\right)^5$$

The curves of velocity, acceleration and pulse may be obtained from the above equation. Plotting these curves (see Fig. 14(ii)) shows that the characteristics are very similar to the cycloidal and, therefore, this cam could be used for high speeds.

(ii) Dwell-Rise-Return-Dwell Curve using Polynomial Equations

When two cycloidal dwell-rise-dwell curves are joined together to give a dwell-rise-return-dwell motion, there is a sudden change in the acceleration curve at the maximum rise point (see Fig. 10). This change in the acceleration curve may be avoided, when using polynomial equations, by fitting a boundary condition of  $y''' = 0$  at the maximum rise point.

Examination of Fig. 15 shows a dwell-rise-return-dwell motion, and the boundary conditions for the dwell-rise portion CD are

$$\text{At D, } x = 0, y = h, y' = 0, y''' = 0;$$

$$\text{At C, } x = d, y = 0, y' = 0, y'' = 0.$$

The basic polynomial equation required is

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5$$

Substituting boundary conditions and solving the resulting simultaneous equation gives

$$\frac{y}{h} = 1 - \frac{10}{3} \left(\frac{x}{d}\right)^2 + 5 \left(\frac{x}{d}\right)^4 - \frac{8}{3} \left(\frac{x}{d}\right)^5 \quad \dots (2)$$

$$\frac{y'}{h} = -\frac{20}{3d} \left(\frac{x}{d}\right) + \frac{20}{d} \left(\frac{x}{d}\right)^3 - \frac{40}{3d} \left(\frac{x}{d}\right)^4$$

$$\frac{y''}{h} = -\frac{20}{3d^2} + \frac{60}{d^2} \left(\frac{x}{d}\right)^2 - \frac{160}{3d^2} \left(\frac{x}{d}\right)^3$$

$$\frac{y'''}{h} = \frac{120}{d^3} \left(\frac{x}{d}\right) - \frac{480}{3d^3} \left(\frac{x}{d}\right)^2.$$

To calculate the return-dwell curve (D.E.) the actual value of acceleration at the maximum rise point D, obtained from formula (2), is used as a boundary condition for curve DE.

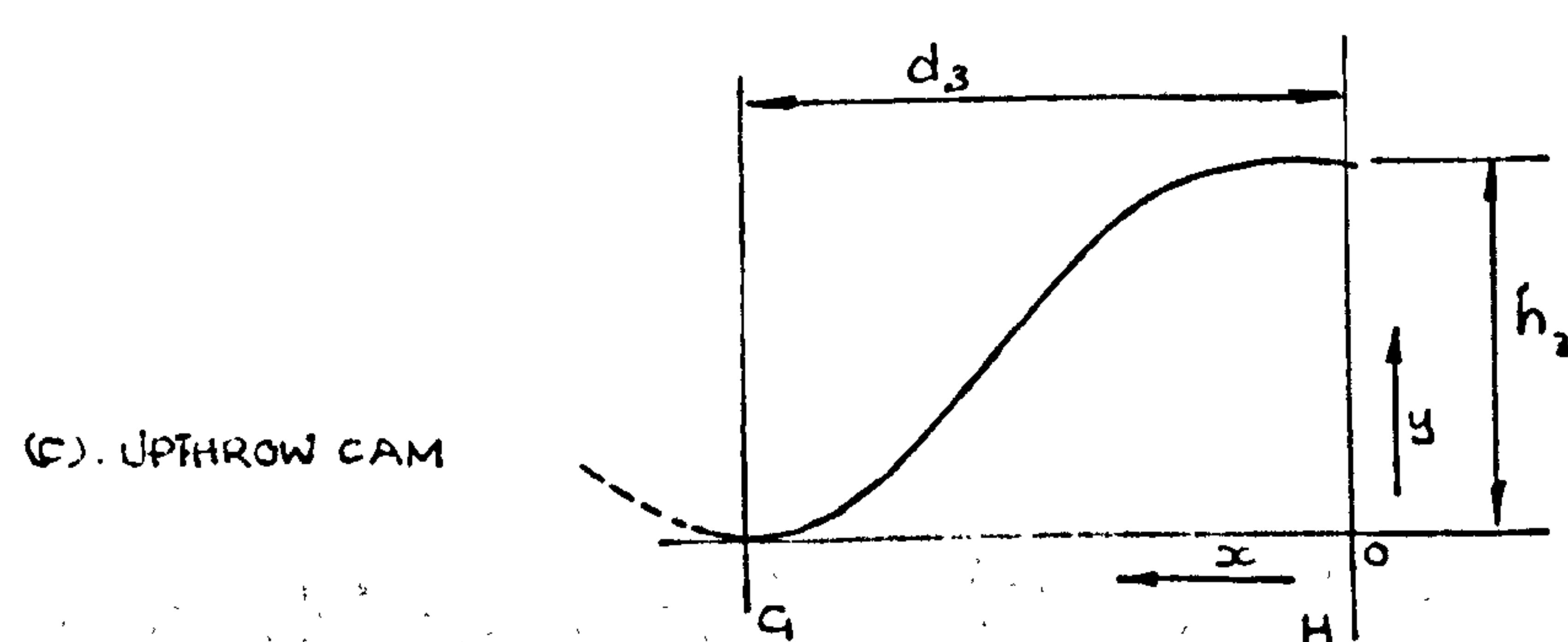
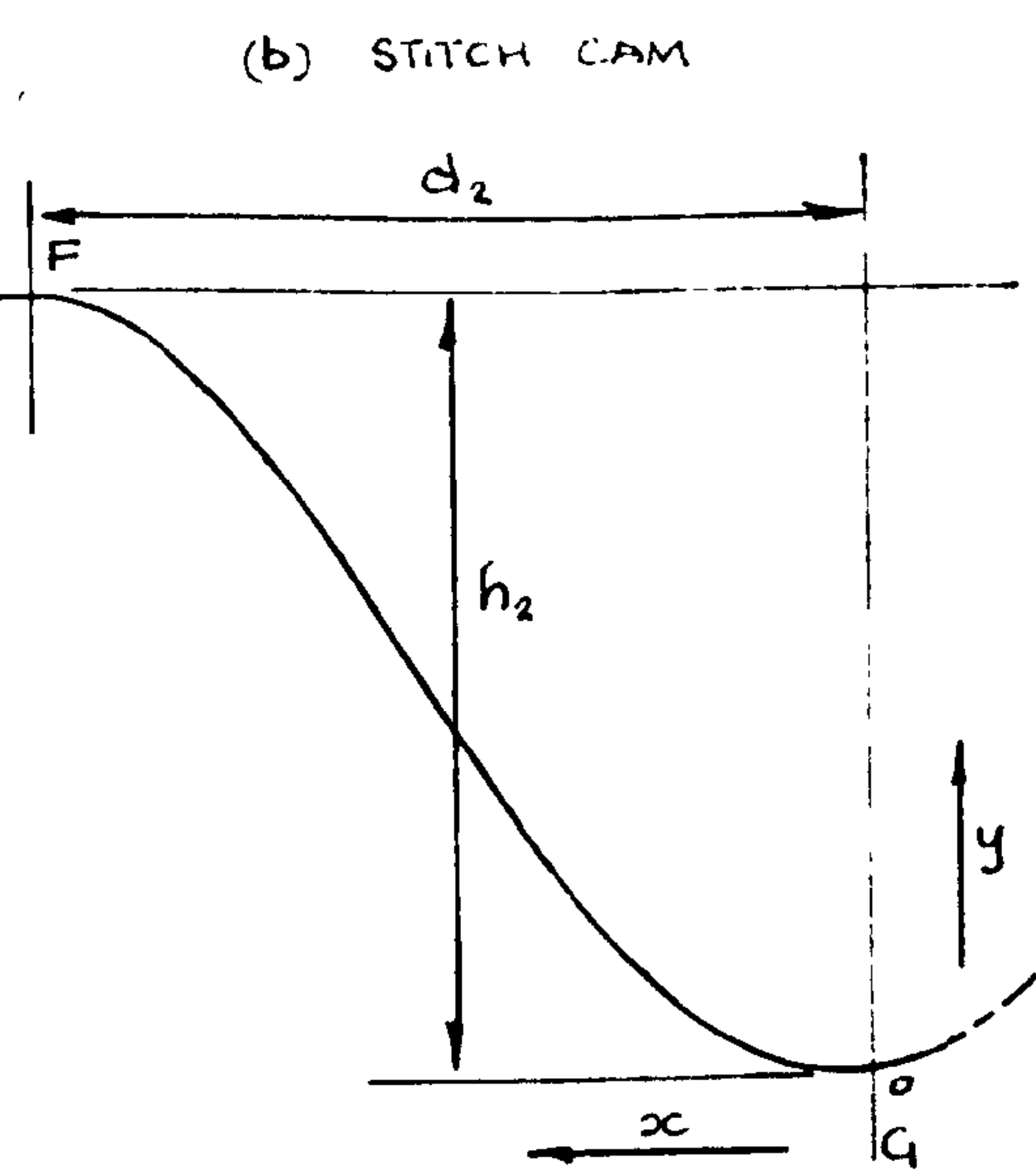
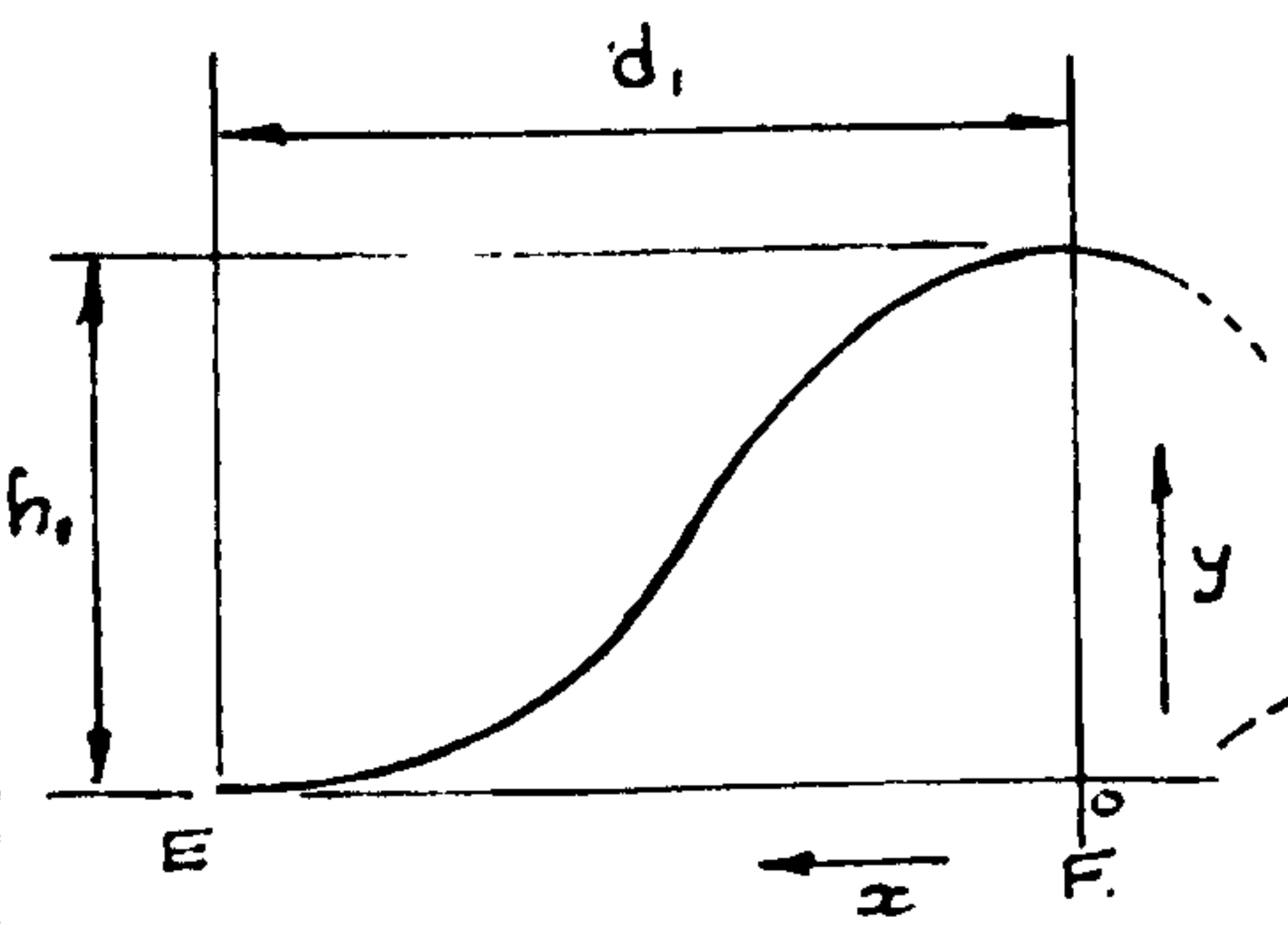
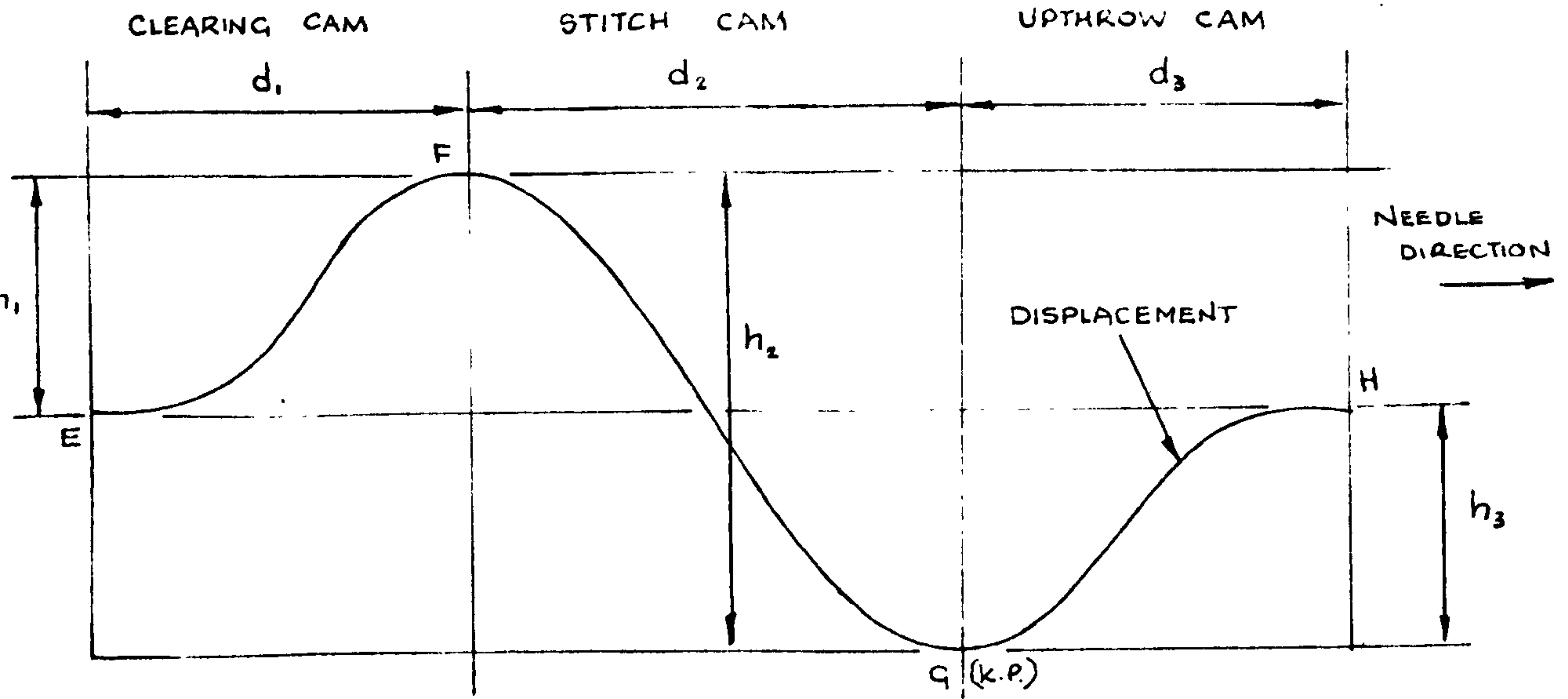
In this way the two curves CD and DE will be joined at position D with a continuous acceleration curve.

The procedure given above will be used to provide a continuous acceleration curve for the non-linear knitting cam system. This is discussed in the following section.



FIG 16

POLYNOMIAL EQUATIONS APPLIED TO A NON-LINEAR CAM SYSTEM



C. Specific Cam Design for the Knitting Cam Systems used in this Work

1. Calculations for the Non-Linear Cams using Polynomial Equations

(a) Introduction

It was realised from the examination of basic and advanced curves given in Section B, that the best method of obtaining a displacement curve, with controlled acceleration characteristics, for the knitting action, was to use polynomial equations.

An examination of Fig. 16 shows the general shape of the non-linear displacement curve for the knitting action, where EF is the clearing cam, FG the stitch cam, and GH the upthrow cam.

The principal dimensions (h and d) of each part of the cam are shown on this diagram.

When calculating the displacement curve by the method of polynomial equations, it was necessary to consider the curves EF, FG and GH separately. It was essential, however, that the boundary conditions of displacement, velocity, acceleration and pulse agreed at the positions F and G, thus ensuring that the above curve characteristics were continuous throughout the knitting action. A summary of the mathematical procedure for calculating the separate parts of the knitting action is given below.

(b) Mathematical Procedure

(i) Clearing Cam (EF)

This curve is shown separately in (a), Fig. 16, and the boundary conditions at E and F may be written down as follows.

$$\text{At E, } x = d_1, y = 0, y' = 0, y'' = 0.$$

$$\text{At F, } x = 0, y = h_1, y' = 0, y''' = 0.$$

The polynomial equation is of the form

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5.$$

Substituting the boundary conditions and solving the resulting simultaneous equations gives the following displacement equation for the clearing cam,

$$\frac{y}{h_1} = 1 - \frac{10}{3} \left(\frac{x}{d_1}\right)^2 + 5 \left(\frac{x}{d_1}\right)^4 - \frac{8}{3} \left(\frac{x}{d_1}\right)^5. \dots (3)$$

(ii) Stitch Cam (FG)

To join the clearing cam and stitch cam together the velocity, acceleration and pulse values at the clearing height F must be the same. The boundary conditions at F, (b), Fig. 16, may, therefore, be written down as follows :

$$\text{At F, } x = d_2, y = h_2, y' = 0, y'' = Q, y''' = 0,$$

where Q is the vertical acceleration value at the clearing height and is obtained from the second derivative of formula (3), when  $x = 0$ .

At position G the boundary conditions are,

$$\text{At G, } x = 0, y = 0, y' = 0, y''' = 0.$$

The polynomial equation is of the form,

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5 + C_6x^6.$$

Substituting the boundary conditions and solving the resulting simultaneous equations gives the following displacement equation for the stitch cam :

$$\begin{aligned} \frac{y}{h_2} = & \left(5 + \frac{Q}{2} \frac{d_2^2}{h_2}\right) \left(\frac{x}{d_2}\right)^2 - \frac{5}{2} \left(6 + Q \frac{d_2^2}{h_2}\right) \left(\frac{x}{d_2}\right)^4 + \\ & + \left(16 + 3 Q \frac{d_2^2}{h_2}\right) \left(\frac{x}{d_2}\right)^5 - \left(5 + Q \frac{d_2^2}{h_2}\right) \left(\frac{x}{d_2}\right)^6 \dots (4) \end{aligned}$$

(iii) Upthrow Cam (GH)

To join the stitch cam and upthrow cam together the velocity, acceleration and pulse values at the knitting point G must be the same. The boundary conditions at position G ((c), Fig. 16) may therefore be written down as follows :

$$\text{At G, } x = d_3, y = 0, y' = 0, y'' = R, y''' = 0,$$

where R is the vertical acceleration value at the knitting point and is obtained from the second derivative of formula (4), when  $x = 0$ .

At position H the boundary conditions are :

$$\text{At H, } x = 0, y = h_3, y' = 0, y'' = 0.$$

The polynomial equation is of the form,

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5 + C_6x^6.$$

Substituting the boundary conditions and solving the resulting simultaneous equations gives the following displacement equation for the upthrow cam.

$$\frac{y}{h_3} = 1 - \left(20 - 2R \frac{d_3^2}{h_3}\right) \left(\frac{x}{d_3}\right)^3 + \left(45 - \frac{11}{2} R \frac{d_3^2}{h_3}\right) \left(\frac{x}{d_3}\right)^4 -$$

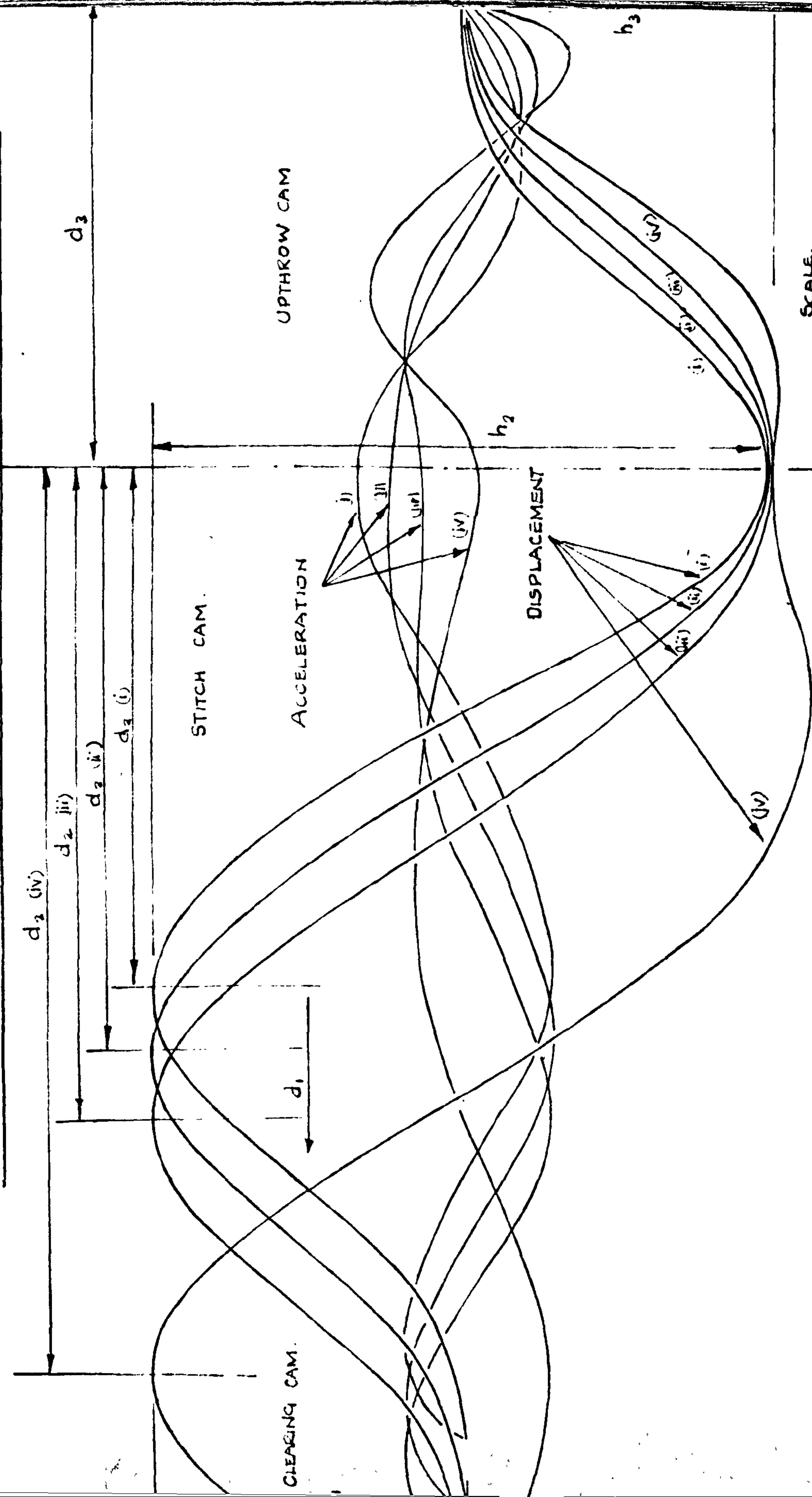
$$- \left(36 - 5R \frac{d_3^2}{h_3}\right) \left(\frac{x}{d_3}\right)^5 + \left(10 - \frac{3R}{2} \frac{d_3^2}{h_3}\right) \left(\frac{x}{d_3}\right)^6 \dots (5)$$

(c) To Establish Actual Values of h and d for the Knitting Action

The distance from the running position to the maximum clearing height (i.e.  $h_1$ , see Fig. 16), is the distance necessary for the needle to rise and clear the loop off the latch. The distance from the clearing position to the knitting point (i.e.  $h_2$ ) is the necessary downward movement of the needle for the old loop to ride over the latch and pull through an average sized new loop below the top of the knitting cylinder. The distance from the knitting point back to the running position ( $h_3$ ) is the upward needle movement necessary to make the new loop hold open the latch. The value of  $h_2$  was obtained from the dimensions of the latch needle, and to provide a symmetrical cam system the running positions E and H (see Fig. 16) were positioned midway between the clearing height and the knitting point, so that  $h_1$  equals  $h_3$ . The value of  $h_2$  was measured and found to be  $2/3''$ . Thus,  $h_1 = h_3 = 1/3''$ .

It has been shown that the formulae for the non-linear displacement curves are designed to give continuous displacement, velocity and acceleration characteristics throughout the knitting action. However, in order to obtain the best high

COMPARISONS OF VARIOUS DISPLACEMENT RATIOS FOR A NON-LINEAR CAM.



SCALE.

DISPLACEMENT. 5:7 :  
ACCELERATION 1in = 15

speed characteristics, such as small acceleration values with relatively small pulse values, it was necessary to substitute different values of  $d_1$ ,  $d_2$  and  $d_3$  into the respective formulae, for the clearing, stitch and upthrow cams, and then to plot out the values of the curve characteristics.

Since the calculations for just one set of  $d$  values (e.g.  $d_1 = 0.5''$ ,  $d_2 = 0.8''$ ,  $d_3 = 0.5''$ ) proved to be such a lengthy procedure, it was considered worthwhile to make a computer programme of the formulae.

As an example of some of the computer results, a comparison of four ratios of  $d$  values is given in Fig. 17, where the values of  $d_1$  and  $d_3$  are fixed, but  $d_2$  is varied, viz:

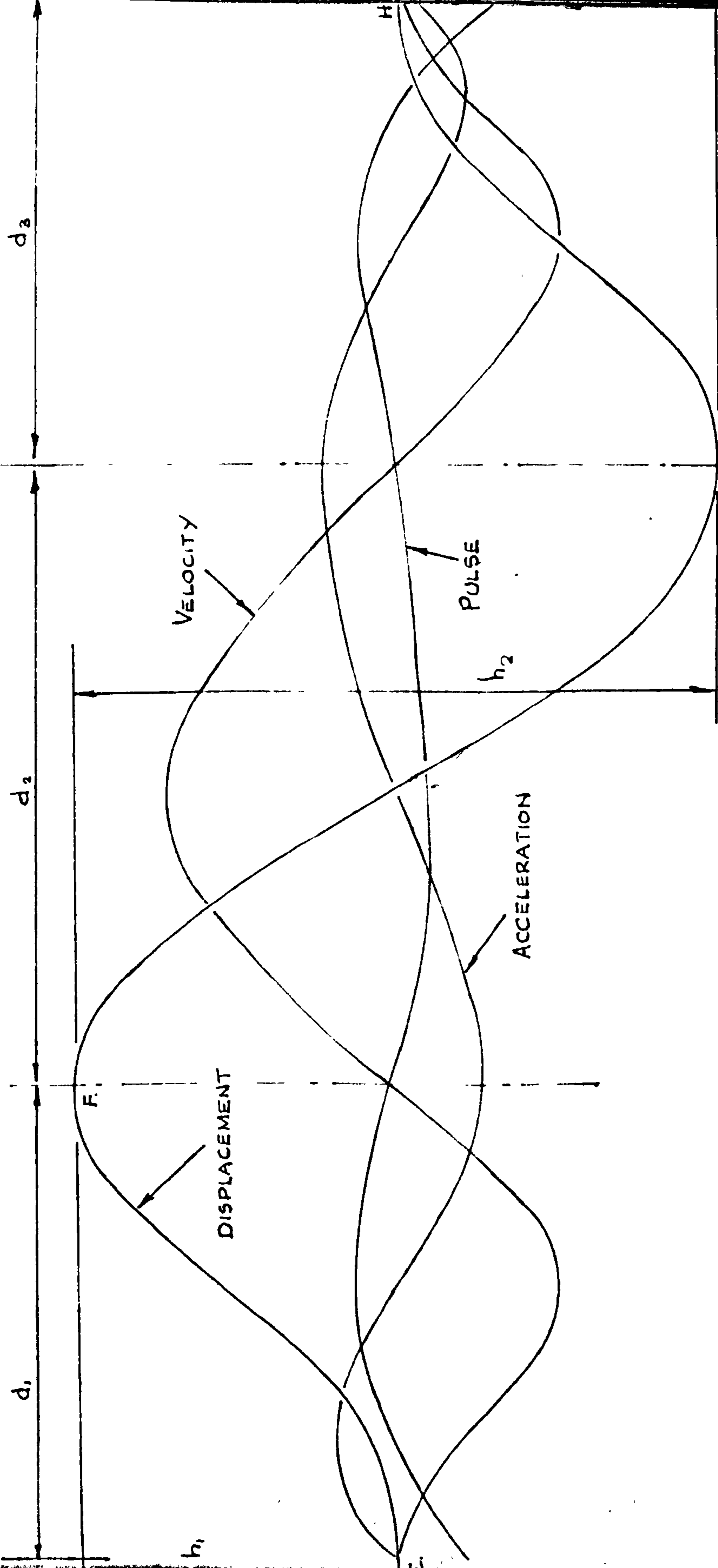
$d_1 = 0.52''$	$d_2 = 0.60''$	$d_3 = 0.52''$	... (i)
$d_1 = 0.52''$	$d_2 = 0.66''$	$d_3 = 0.52''$	... (ii)
$d_1 = 0.52''$	$d_2 = 0.75''$	$d_3 = 0.52''$	... (iii)
$d_1 = 0.52''$	$d_2 = 1.05''$	$d_3 = 0.52''$	... (iv)

The displacement ratio (iv) which has the greatest value of  $d_2$  is unsuitable for the knitting action since it has a large dip on both sides of the knitting point, which would introduce high knitting tensions during loop formation. In addition, the acceleration curve has a corresponding dip which would be undesirable for high speeds.

The other ratios (i) and (iii) provide acceptable displacement and acceleration characteristics, although the maximum acceleration value of ratio (i) is higher than that of ratio (ii) at the knitting point, and the acceleration curve of the ratio (iii) is slightly higher than that of (ii) at the end of the upthrow cam.

# CURVE CHARACTERISTICS OF A NON-LINEAR KNITTING CAM SYSTEM

CLEARING CAM  $d_1$  STITCH CAM.  $d_2$  UPTHROW CAM  $d_3$



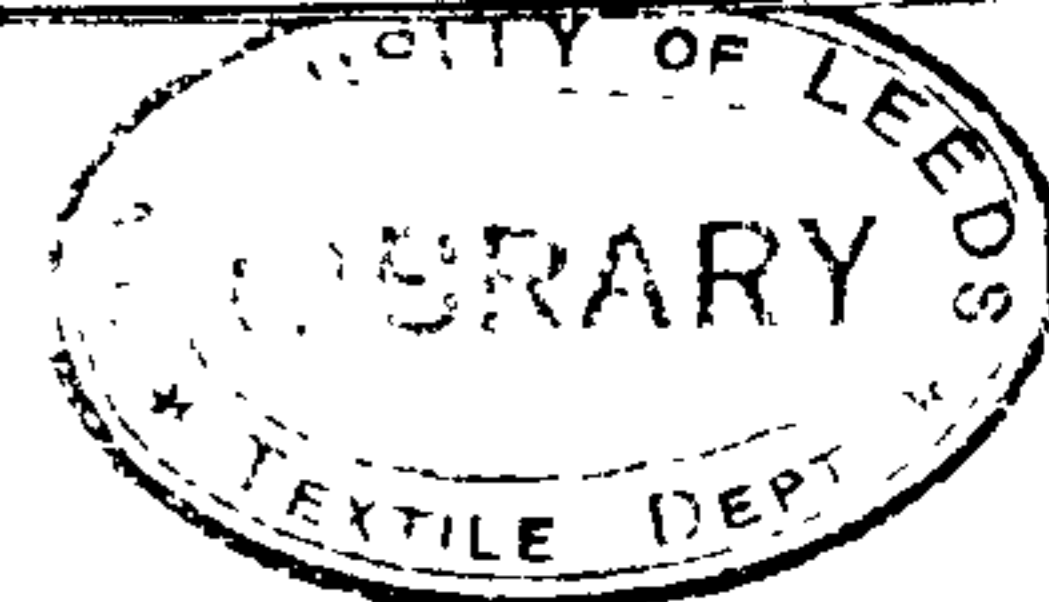
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DISPLACEMENT. 5.7 :

ACCELERATION 110 = 15

44





It was considered, therefore, that the ratio with the best high speed characteristics was ratio (ii), i.e.

$d_1 = 0.52''$ ,  $d_2 = 0.66''$ ,  $d_3 = 0.52''$ , and this was chosen for the non-linear cam system used in this work. The characteristics of displacement, velocity, acceleration and pulse for this ratio are shown enlarged in Fig. 18.

To obtain the maximum slope for any part of the cam, the following procedure may be used. The slope of the cam is given by  $\frac{dy}{dx}$ , hence the maximum slope occurs when  $\frac{d^2y}{dx^2} = 0$ .

For example, for the clearing cam, (formula 3),

$$y = h_1 \left[ 1 - \frac{10}{3} \left( \frac{x}{d_1} \right)^2 + 5 \left( \frac{x}{d_1} \right)^4 - \frac{8}{3} \left( \frac{x}{d_1} \right)^5 \right]$$

Hence

$$y' = h_1 \left[ -\frac{20}{3d_1} \left( \frac{x}{d_1} \right) + \frac{20}{d_1} \left( \frac{x}{d_1} \right)^3 - \frac{40}{3d_1} \left( \frac{x}{d_1} \right)^4 \right]$$

and

$$y'' = h_1 \left[ -\frac{20}{3d_1^2} + \frac{60}{d_1^2} \left( \frac{x}{d_1} \right)^2 - \frac{160}{3d_1^2} \left( \frac{x}{d_1} \right)^3 \right]$$

The maximum slope occurs at the value of  $x$  obtained by equating the last expression to zero, i.e.

when

$$-\frac{1}{3} + 3 \left( \frac{x}{d_1} \right)^2 - \frac{8}{3} \left( \frac{x}{d_1} \right)^3 = 0$$

the solutions of this equation are

$$\frac{x}{d_1} = 1, \quad \frac{x}{d_1} = \frac{1 \pm \sqrt{33}}{16}$$

The first of these solutions refers to the point E (Fig. 18) where the velocity attains a mathematical maximum of zero, i.e. the slope is zero. The solution  $\frac{x}{d_1} = \frac{(1 - \sqrt{33})}{16}$  is

outside the range of the cam. The remaining solution  $\frac{x}{d_1} = \frac{(1 + \sqrt{33})}{16} = 0.4216$  gives the position of the maximum slope of the cam.

The maximum slope itself can be obtained from the equation for  $y'$  above. If  $\theta_m$  is the maximum angle we have

$$-\tan \theta_m = \frac{dy}{dx}$$

$$x = 0.4216d,$$

$$-\tan \theta_m = \frac{h_1}{d_1} \left[ -\frac{20}{3} (0.4216) + 20(0.4216)^3 - \frac{40}{3} (0.4216)^4 \right]$$

Since  $h_1 = 0.33$  and (for example)  $d_1 = 0.52$ , this gives

$$-\tan \theta_m = -1.115$$

$$\text{Hence } \theta_m = 48^\circ 6'.$$

When the stitch cam is considered, the polynomial equation is of the 6th order and, consequently, its second derivative is of the 4th order; in order to find the position of the maximum slope of the stitch cam it is therefore necessary to solve a quadratic equation and this is not usually considered worthwhile.

However, measurement of the clearing cam angle from Fig. 18 gives an angle of  $49^\circ$  which is within 2% of the calculated value for the clearing cam. It was considered, therefore, that the maximum cam angles of the non-linear cam could be obtained by accurate measurements.

From Fig. 18, the maximum angle measured for the clearing, stitch and upthrow cams were  $49^\circ$ ,  $57^\circ$  and  $49^\circ$ , respectively.

For this work it was desirable to compare the performance of three non-linear knitting cam systems when using different maximum angles. To obtain three non-linear cams it was convenient to plot the displacement points for ratio (ii) on alternatively a shorter or longer horizontal base.

The measured maximum angles of the clearing, stitch and upthrow cams for the non-linear cams designed for this work are given below :

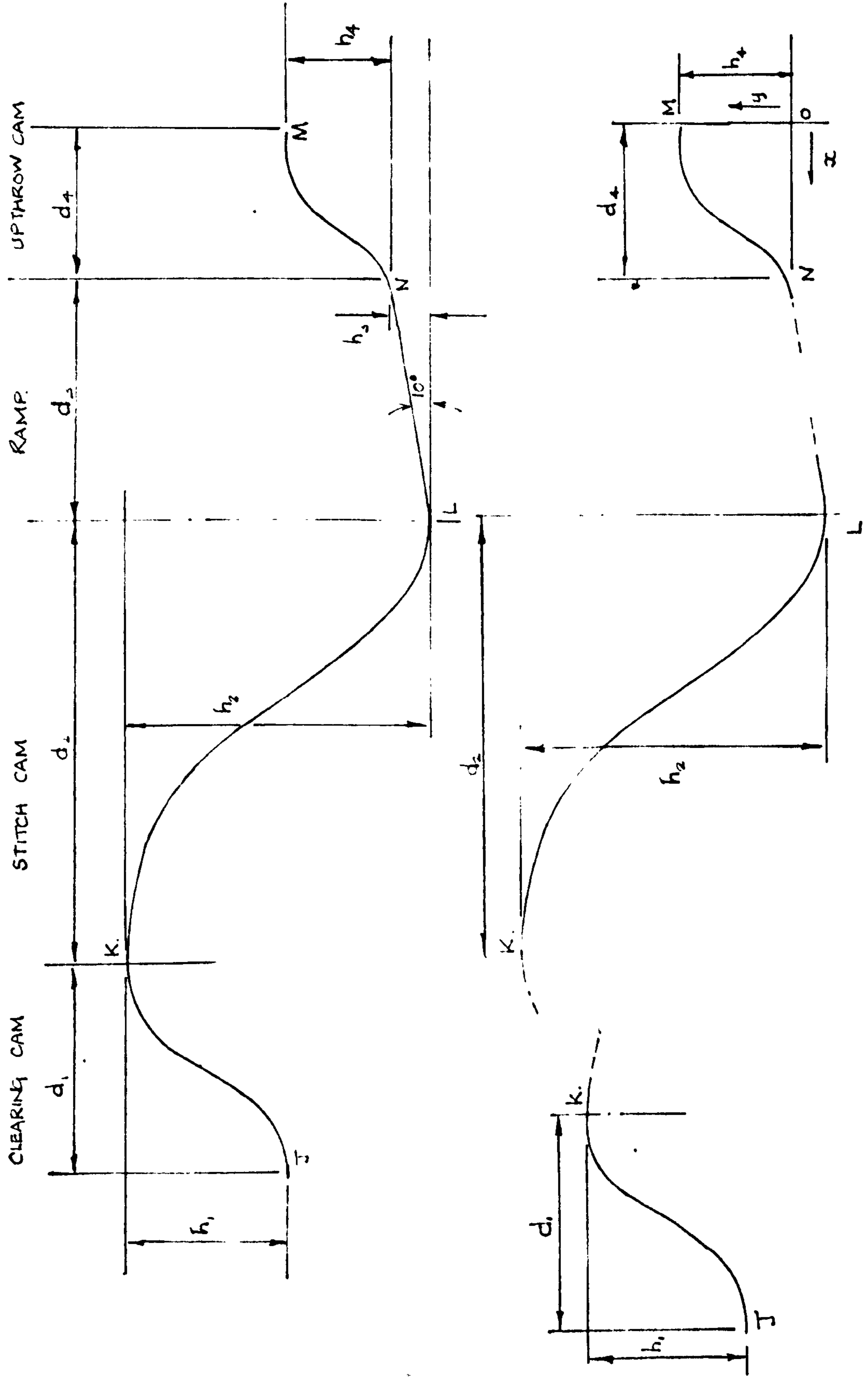
	<u>Clearing</u> <u>Cam</u>	<u>Stitch</u> <u>Cam</u>	<u>Upthrow</u> <u>Cam</u>
(a)	$44^\circ$	$52^\circ$	$44^\circ$
(b)	$49^\circ$	$57^\circ$	$49^\circ$
(c)	$54^\circ$	$62^\circ$	$54^\circ$

and for convenience the cams are subsequently referred to by the maximum angles of the stitch cam, i.e. (a)  $52^\circ$  non-linear (b)  $57^\circ$  non-linear, (c)  $62^\circ$  non-linear.

## 2. Calculations for the "straight and curved" cam using basic curves and polynomial equations.

The loop-forming shape for the "straight and curved" cam is shown in Fig. 19, and suitable basic curves and a polynomial curve were chosen to give a controlled acceleration curve.

FIG 19 MATHEMATICAL DESIGN FOR A 'STRAIGHT & CURVED' KNITTING CAM SYSTEM



Each cam portion was taken separately and the method of calculation is summarised below.

(i) Clearing Cam

The clearing cam is essentially a dwell-rise-dwell curve. It has been shown that for this motion excellent acceleration characteristics may be obtained using the cycloidal curve.

The equation for the cycloidal curve, given below, is well known :

$$y = \frac{h}{\pi} \left[ \frac{\pi\alpha}{\beta} - \frac{1}{2} \sin \frac{2\pi\alpha}{\beta} \right] \text{ ins.}$$

where  $\alpha$  = angle of cam rotation (degrees),

$\beta$  = maximum angle of cam rotation (degrees).

The displacement curve of the clearing cam (J K, Fig. 19) may be written in terms of  $x$  and  $d$ ,

$$y = \frac{h_1}{\pi} \left[ \pi \frac{x_1}{d_1} - \frac{1}{2} \sin 2\pi \frac{x_1}{d_1} \right] \dots (6)$$

The characteristics of velocity and acceleration for this curve have been shown previously in Fig. 9.

(ii) Stitch Cam

The stitch cam was provided with a low angled start for two reasons. If any inaccuracy occurred at the top of the clearing cam (K) the acceleration of the needle butt would not be zero, as predicted by the mathematics of the cycloidal curve, and the needles would tend to fly off the top of the clearing cam with a fairly high acceleration. It was important, therefore, for the

needle butts to meet the stitch cam with a low impact, and this would only be achieved if the start of the cam were very slow.

With a low angled start to the stitch cam there was no reason why a separate adjustable stitch cam should not be used, since the needle butt would meet the slow start when the stitch cam was set to different vertical depths.

If an adjustable stitch cam were to be fitted, then a  $10 - 15^\circ$  ramp could be used to start the curve; this could be joined by a stitch cam calculated from polynomial equations to provide the fall to the knitting point.

In the design used for this work a separate adjustable stitch cam was unnecessary and therefore it was more convenient to use a double harmonic curve to provide the slow start at K and the fall to the knitting point.

The double harmonic curve gives a slower start than any other basic curve, but has a finite acceleration value at the end which reduces its popularity as a dwell-rise-dwell cam for most applications.

However, in the particular case of loop-forming, experiments mentioned in Section A have shown that the tensions of the yarn, as a new loop is formed, are sufficiently high to hold the needle butt against the upper face of the stitch cam during the downward traverse, and are high enough to make the butts rise slightly after the knitting point. Consequently, it is expected that the finite acceleration values provided by the double harmonic cam will be counteracted by the yarn tensions

during knitting, and this will prevent the needle butts from flying off the stitch cam at the knitting point.

The double harmonic curve comprises the difference between two simple harmonic curves, one being  $\frac{1}{4}$  of the amplitude and twice the frequency of the other.

The equation for the double harmonic displacement curve is given as :

$$y = \frac{h}{2} \left[ \left( 1 - \cos \frac{\pi \alpha}{\beta} \right) - \frac{1}{4} \left( 1 - \cos \frac{2\pi \alpha}{\beta} \right) \right]$$

The displacement curve of the stitch cam (K L, Fig. 19), may be written in terms of  $x$  and  $d$

$$y = \frac{h_2}{2} \left[ \left( 1 - \cos \pi \frac{x_2}{d_2} \right) - \frac{1}{4} \left( 1 - \cos 2\pi \frac{x_2}{d_2} \right) \right] \dots (7)$$

The characteristics of the velocity and acceleration for this curve have been shown previously in Fig. 9.

### (iii) Uphrow Cam

When the needle butts leave the stitch cam it is important that they meet the upthrow cam with a fairly low impact force; the start of the upthrow cam is therefore provided with a low angled ramp. The ramp is also a necessity when a separate adjustable stitch cam is being used, as it will ensure the same low impact to the needle butts whatever the stitch cam setting.

The  $10^\circ$  ramp is shown in Fig. 19 with dimensions  $h_3$  and  $d_3$  where  $h_3 = 0.097$  ins. and  $d_3 = 0.543$  ins. and the ramp is joined by a curve calculated from polynomial

equations - it was easier to use polynomial equations than basic curves because of the initial velocity of the needles at the end of the ramp.

The upthrow cam MN is shown separately in Fig. 19 and the boundary conditions at M and N may be written down as follows :

$$\text{At M, } x = 0, y = h_4, y' = 0, y'' = 0;$$

$$\text{At N, } x = d_4, y = 0, y' = \frac{h_3}{d_3}, y'' = 0.$$

Polynomial equation is in the form,

$$y = C_0 + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + C_5x^5.$$

Substituting the boundary conditions and solving the resulting simultaneous equations gives the following displacement equation for the upthrow cam :

$$\begin{aligned} \frac{y}{h_4} = 1 - \left(10 + 4 \frac{h_3}{d_3} \cdot \frac{d_4}{h_4}\right) \left(\frac{x}{d_4}\right)^3 + \left(15 + 7 \frac{h_3}{d_3} \cdot \frac{d_4}{h_4}\right) \left(\frac{x}{d_4}\right)^4 - \\ - \left(3 \cdot \frac{h_3}{d_3} \cdot \frac{d_4}{h_4} + 6\right) \left(\frac{x}{d_4}\right)^5 \dots (8) \end{aligned}$$

For the design of the straight and curved cam values of  $h_1 = 0.33''$ ,  $h_2 = 0.66''$ , and  $h_4 = 0.238''$  were used in the respective formulae 6, 7 and 8, and values of  $d_1$ ,  $d_2$  and  $d_4$  were chosen to give a maximum angle of  $55^\circ$  for the clearing, stitch and upthrow cams.



### 3. Design of Commercial Type Cam Systems

It is known that commercial cam systems have maximum angles ranging between  $45^{\circ}$  and  $55^{\circ}$ ; It was therefore decided to design and make two linear cams with maximum angles of  $45^{\circ}$  and  $55^{\circ}$ .

Although commercial cams are usually fitted with separate adjustable stitch cams, it was more convenient to make these cams from complete pieces of steel; so that they could be fitted on to the same section as the non-linear and the straight and curved cams. A range of stitch length was obtained by lowering or raising the complete cam system in the slots of the cam section, in the same way as the curved cams.

The cams, in other respects, were similar to most commercial cams producing plain knitted fabric. They were designed with straight lines and slight curvatures at the leading edges.

The dimensions of the cams were obtained from the length of the needle latch and the distance from the running position to the top of the cylinder, as with the curved cams, (i.e.  $h_1$ ,  $h_2$  and  $h_3$  equalled  $1/3''$ ,  $2/3''$  and  $1/3''$  respectively), and the circumferential lengths were equivalent to those of commercial cams.

#### D. Method of Manufacturing Cams

##### 1. Introduction

Six cam systems were made for this work :

- (a) Three "non-linear" cams with maximum stitch cam angles of  $52^{\circ}$ ,  $57^{\circ}$  and  $62^{\circ}$ , respectively. The cam profiles were calculated from polynomial equations and were

designed to give continuous acceleration characteristics.

- (b) One "straight and curved" cam with a maximum angle of  $55^{\circ}$  for each cam portion. This cam profile was calculated from basic curves and polynomial equations, and was designed to give controlled acceleration characteristics.
- (c) Two "linear" cams with maximum angles of  $45^{\circ}$  and  $55^{\circ}$  respectively. These cam profiles are similar to commercial cams and were used for comparison with the curved cams; they were designed with straight lines and curved edges.

It was realised that the degree of success of these cams, as high speed systems, would largely depend upon the accuracy with which they were made. Consequently, the techniques for making the cams were chosen carefully to ensure the best possible accuracy with the facilities available. Even so, it was not expected to attain the accuracy necessary to give the calculated theoretical characteristics. It was suggested, however, that an inaccurately made polynomial curve, with theoretical continuous acceleration, would probably give better high speed performance than, for example, an inaccurately made simple harmonic motion curve, which already has a theoretical sudden change in acceleration at the ends. Another justification for using sophisticated theoretical designs for these cams, is that it seems likely that, over the next few years, more accurate camming machinery will become available, which will permit cams to be more accurately produced to a theoretical outline.

## 2. Making Master Cams

To obtain the best accuracy possible in constructing the cams, master cams six times actual cam size were made for all six cam systems. These were used on a profile milling machine with a 6:1 reduction, to reproduce the cam profiles. This technique is often used, as high accuracy can be achieved in producing a large master cam; also any errors which are made will be reduced six times when milling the actual cam profile.

To make the master cam, the mathematical displacement points for each of the six cam systems were plotted out on graph paper, thus providing the leading edges of the cam profile. The mating curves were then obtained by measuring off the width of the needle butt (six times actual size), from each leading edge at frequent intervals, and joining these points with a smooth curve. The upper and lower profiles were cut out accurately and attached to 1/32" aluminium sheeting.

A thin line was scribed around the paper profile in order to reproduce the profile accurately on to the aluminium sheet. The aluminium profile was cut out, using a band saw to follow the scribed line closely, and was then filed carefully to the line and finally polished with fine grain emery paper to give a smooth edge.

## 3. Material used for making Cams

A trial cam was made with mild steel and the profile edge was hardened by carbonising. With this method, however, it proved difficult to obtain a hard edge throughout the profile, owing to the problem of getting an even layer of carbon on to the sloping tracks.

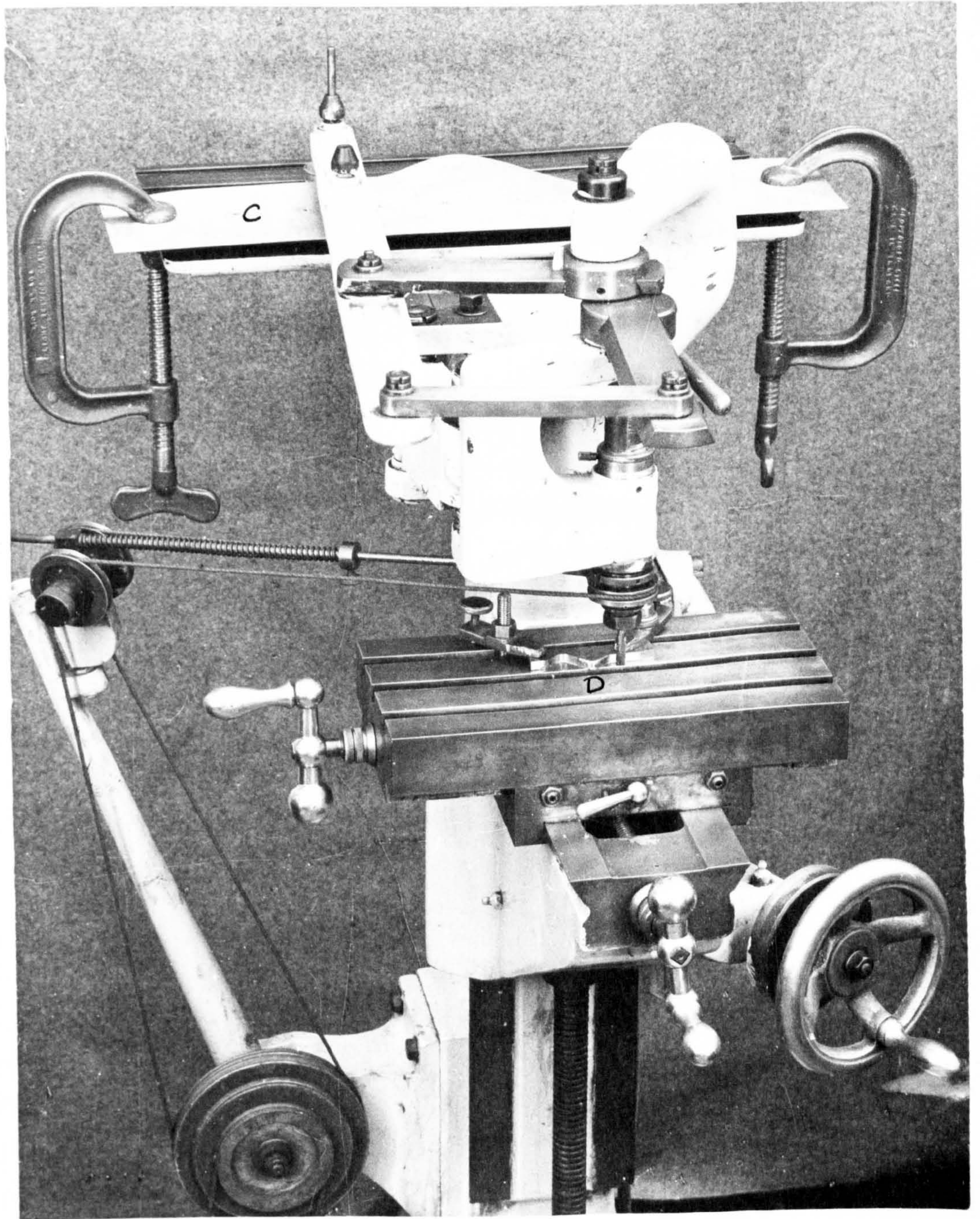
The material finally chosen for making the cam systems was a special oil-hardened cast steel. Although this material gave an adequate hardness to the profile edge, it also became hardened over the whole cam system, unlike the carbonising method which only hardened the actual profile edge. Consequently, great care had to be taken in be<sup>n</sup>ding the cams to the cam section, to reduce the chances of cracking when the cams were clamped down. To prepare the material for cam making, the cast steel was ground to a thickness of 0.38". Rectangles were then milled to give a suitable width and a correct cam length of 3.375 inches.

#### 4. Profile Milling

The ideal profile milling machine for making cams is one which functions without a master cam or even an operator. (14,15). All that is required is a punched tape with information of the displacement profile, as the machine has a built-in computer supplying data to a mechanism which continuously moves the cutting tool to its correct position. The positioning of the tool is extremely accurate, and any vibration of the cutter is corrected by a "feed back" unit.

A machine tool of this type, however, was not available for this work and the machine used for cutting the cam systems was a small profile milling machine, which had originally been designed for engraving work (see Fig. 20). The main disadvantage of this machine was that the vibration of the cutting tool became quite large when cutting the cam profiles. It was considered, however, that the accuracy of the cams would not be affected if only small cuts were taken on each traverse of the milling cutter.

FIG. 20



A wheel, which rotated freely on a bearing, was made six times the size of the milling cutter for following the profile of the master cam. For cutting the cast steel cams a special high speed Wimet milling tool ( $\frac{1}{4}$ " diameter), was acquired, as the ordinary cutters tended to wear very quickly.

The master cam, C, and cast steel rectangle, D, were clamped to their respective tables of the milling machine. The profile linkage mechanism was set to give a 6:1 reduction, and checked by ensuring that the milling tool and follower would touch each end of the rectangle and the master cam respectively.

The technique of cutting the profile was to traverse the follower, by hand, around the profile edge of the master cam until the milling cutter had described the outline of the profile on the surface of the rectangle. In order to reduce milling time, the main bulk of the material was then removed by sawing carefully around the outline of the profile. This rough profile was then ground by hand, leaving as little excess material as was possible without cutting through the profile outline.

The cam profile was reclamped to the milling machine and its edge milled until perfectly formed. An important factor, when cutting this profile, was to make sure that the follower moved continuously and slowly on the edge of the master cam. If the movements of the follower were not continuous, it was likely that the profile would not be smooth, and the acceleration characteristics, therefore, seriously affected.

The width of the profile edge (0.38"), was then reduced, as only a small contact area of needle butt to cam track was necessary. This was achieved by cutting back

approximately 0.050" on the profile edge, but leaving approximately 0.030" thickness of the edge untouched to act as the needle butt track. The profile was then ready for bending.

##### 5. Bending Cams

The mathematical characteristics of the cam profiles are unaffected by bending. However, if two mating curves are bent differently, then these curves will not fit together accurately. This is to be expected as the arcs of the curves will be different and consequently the displacement points on two mating curves will not be exactly opposite.

As the radius of the knitting cylinder used in this work was 4", it was necessary to bend the cams to this same radius. Two cast iron blocks - one with a concave radius and the other a convex radius of 4" - were used in a hydraulic press, to bend the cams. The cams were heated to 800°C, then quickly positioned between the two cast iron blocks. Pumping the lever of the hydraulic press brought the blocks together and therefore bent the cams to a 4" radius. As a standard for bending, a pressure of 1,000 lbs/sq.in was attained for all the cams before the blocks were released.

It proved to be very difficult to obtain a perfect bend over the complete length of the cam, even after several reheating and bending attempts, this being mainly the result of trying to bend particularly long cam profiles. It was observed that needle butts would not pass through the profiles with the correct clearance if the two cams had been bent differently; it was essential, however, that this clearance should be correct with the non-linear cams, as a positive control was required over the needle butts.

In order to rectify the bending errors, it was considered worthwhile to try and re-machine the bent cam profiles.

#### 6. Re-machining Cam Profiles

It was found convenient to mount the bent cam profile on the concave block used in bending, and to position this block on the table of the milling machine. The master cam profile was fixed to a curved plate of 24" radius and a long follower was made, so that contact with the master cam was maintained along the curved length of its profile. Only a few traverses of the follower were required to eliminate the bending errors and it was found, on matching two mating profiles, that they now fitted very closely, with the correct clearance.

The mathematical characteristics of each curve were not affected by this re-machining procedure since both the master cam and actual cam were bent to respective radii of 24" and 4", which ensured that the horizontal and vertical displacement points were correct.

There was a slight discrepancy, however, when machining cams which were already bent. After bending, the cam tracks acted radially inwards and this gave a correct motion for the needle butts, as they were held in radial tricks in the knitting machine cylinder. Re-machining these bent cams with a vertical milling tool eliminated the radial slope of the cam tracks, which meant that the actual contact area for the needle butts was reduced. However, this was not too important as the needles would "bed-in" and provide an adequate contact area after a period of running-in.



## 7. Hardening the Cams

After drilling and tapping screw holes in the cams, the cams were reheated in a controlled oven to a temperature of 800°C, and quenched in whale oil to give the necessary hardness to the cam tracks. The cams became completely hard throughout and those that had been bent imperfectly were likely to break when screwed down on the cam section - this, in fact, happened with the first cams that were made. It was important, therefore, to bed the cams perfectly to the cam section. This was done by relieving material from the back of the cams, apart from an area around the two screw holes. These areas were then polished with a hand stone until a perfect seating with the cam section had been achieved.

## 8. Running In

The cam tracks were polished carefully with fine emery paper to remove the scale developed during hardening.

The six cam systems, which are shown in Fig. 21, were run in with a full set of needles in the knitting machine; after a period of approximately 60 hours, the leading edges became shiny and smooth and were considered to be sufficiently well run in.

Many needle butts broke in the initial periods of running in with the steeper angled cams. After a period, the tracks became smooth and needle breakages became less frequent until no breakages occurred at all. Additional needle breakage also occurred with the non-linear cams, but this was due to a variation in size of the needle butts - the larger butts jamming in the positive track. Eventually these larger butts were eliminated and no further breakages occurred.

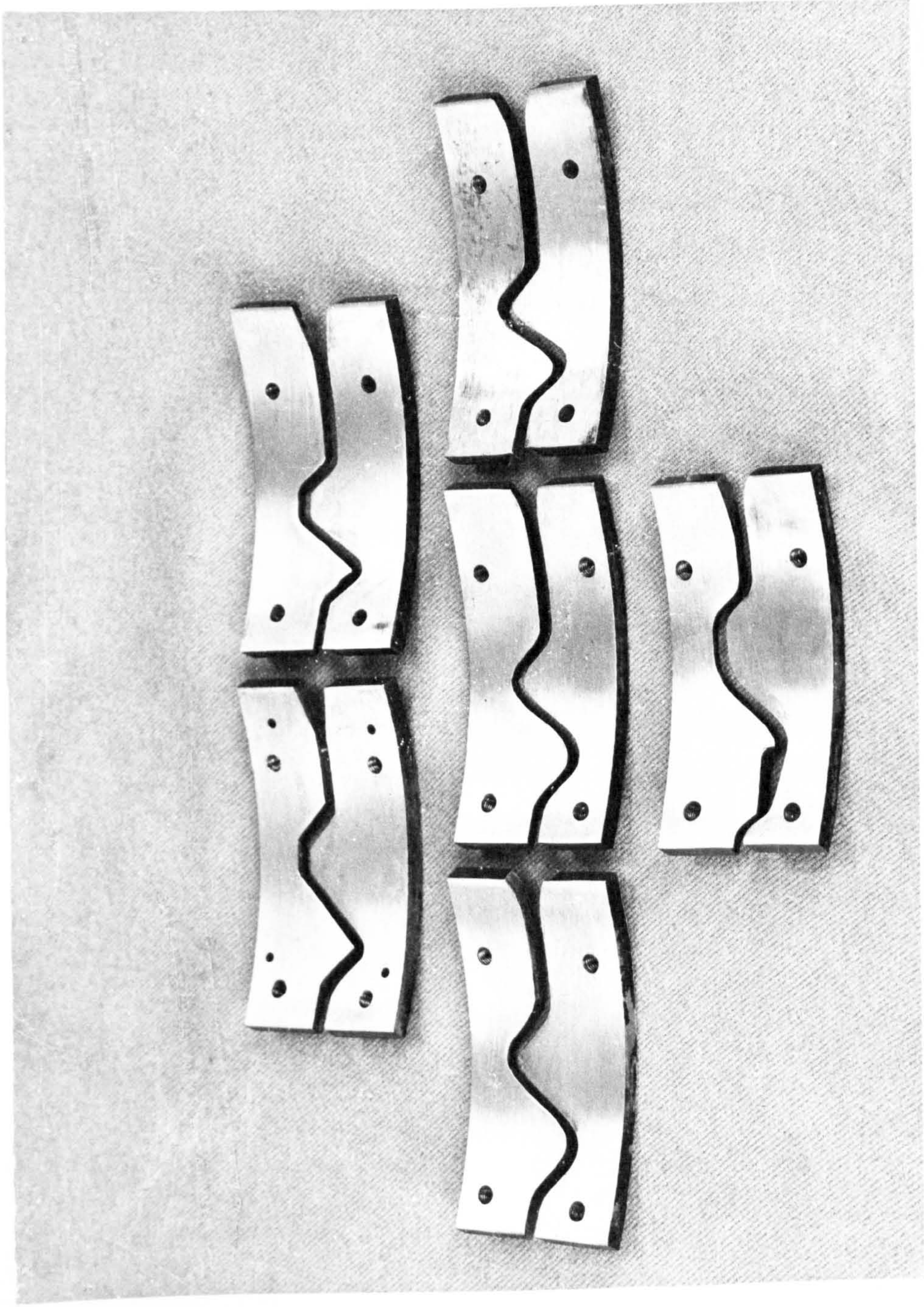


FIG. 21

Quite a lot of damage can be done by needle butts; on one or two occasions during running in, a butt would bend into an adjacent butt, and this set up a chain reaction until as many as twenty butts were bent beyond repair and the damage done to the trick walls was severe.

It was therefore decided to harden the needle butts so that they were particularly brittle and, consequently, would shear off rather than bend and damage the tricks. A window, made in the cam section so that the needle butts could be observed, provided an excellent escape for the broken butts, as they tended to fly out of the window rather than tangle with the butts moving through the cam track. This worked particularly well, and the hardened butts were invaluable for running in, and also for later work when extreme conditions of knitting, such as high yarn tension, high speeds, etc., were used, as inevitably many needle butts were broken.

CHAPTER 3

A DESCRIPTION OF THE KNITTING APPARATUS AND  
MODIFICATIONS TO THE MACHINE TO ENABLE HIGH SPEED  
TESTS AND MACHINE TORQUE MEASUREMENTS TO BE MADE

CHAPTER 3

A DESCRIPTION OF THE KNITTING APPARATUS AND  
MODIFICATIONS TO THE MACHINE TO ENABLE HIGH SPEED  
TESTS AND MACHINE TORQUE MEASUREMENTS TO BE MADE

Preamble

It has been shown in the previous chapter how various high speed cam systems were designed and manufactured.

In Chapter 5 it will be shown how these cam systems were subjected to many different knitting conditions including knitting tests at very high speeds. Measurements of machine torque, forces on cams and stitch length were also made, so that an analysis of the performance of the cams could be achieved.

The purpose of this chapter is to describe the modifications necessary to a commercial knitting machine to suit the particular requirements of this work. The first part is concerned with a description of the machine and some of the important components, and the second part describes a major modification to the machine bearing system, to make it suitable for high speed tests and machine torque measurements.

A. Description of the Knitting Apparatus

(a) Experimental knitting machine

An 8" diameter weft knitting machine, of the rotating cylinder type, was supplied by A. Kirkland & Co. Ltd, for use in this experimental work. The machine, fitted with eight feeders, was originally intended for producing patterned fabric using the pattern wheel system; however, to suit our requirements it was supplied with a spare set of blank

cam sections so that cams could be made to our own specification and design. For the experimental work, the machine was reduced to a single feed, and the various cam shapes designed and manufactured for this work were attached separately to one cam section; the remaining cam sections were maintained with horizontal running cams.

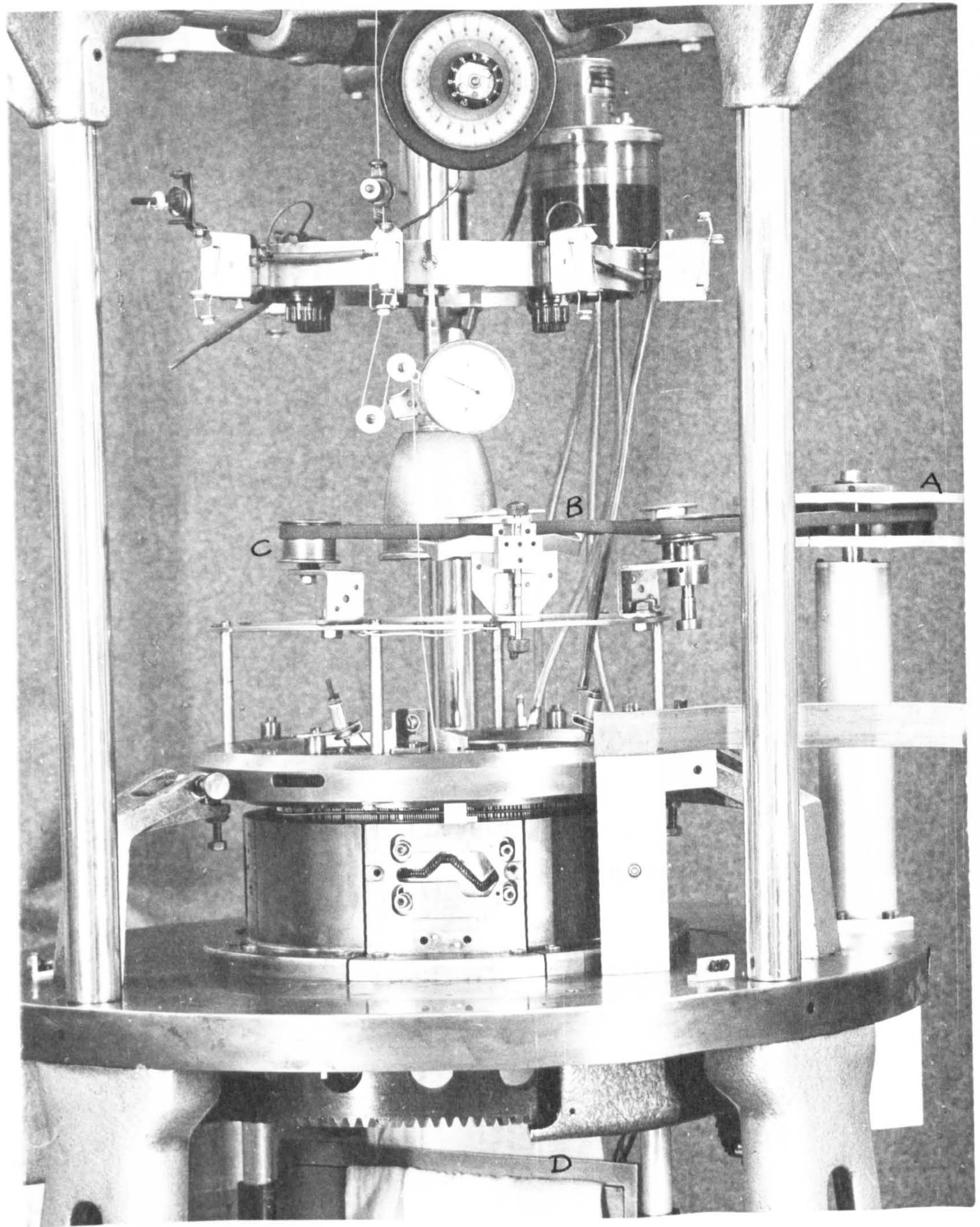
The machine is shown in Fig. 22, and some of the important features, such as the yarn tension device, positive feed and fabric take-down unit are described in detail below. The machine drive and bearing system is described in the second part of this chapter.

#### (b) Yarn Tension Devices

It is well known that the yarn input tension (the tension of the yarn prior to its entering the needles), must be constant if a good quality fabric is to be obtained when knitting without positive feed. It has also been shown, by Nutting (16), that the variation in stitch length is greater at low yarn input tensions than at high input tensions. To obtain constant tension values a reliable tension control device is necessary, more especially when knitting at tensions below 5 gms.

Many different types of tension control devices are available, but some are far more effective and reliable than others. Kalkman (17), did some extensive tests on more than fifty different types of tension device and related them to steadiness and "sensitivity to speed". He found that the three most efficient devices were the Hysteresis brake, the Gate tension and the Disc tension with a compensator, in this order.

FIG. 22



The Hysteresis brake and the Gate tension device, however, are not normally used in the knitting industry. The Hysteresis brake is expensive and there is a tendency for the yarn to wrap around the feed wheel when the machine stops, which would obviously be inconvenient on a multifeed machine.

The Gate tension device has the disadvantage of being a rather large and cumbersome arrangement and, as the technique of adding or subtracting weights to alter the tension value would be a lengthy procedure on a multifeed machine, this tension device is also unsuitable for commercial use.

The Disc tension with compensator, which was Kalkman's third most efficient device, is fitted as standard equipment on many commercial machines - including the machine supplied for this work. It is compact, easy to use and gives satisfactory performance.

For a large proportion of the experiments the disc tension device was used, since it was considered important that an assessment of the cam's performance should be made with the tension unit most likely to be used in commercial practice. With certain experiments, however, the hysteresis brake was used to check the results.

(c) Positive Feed System

A positive feed system supplies a fixed quantity of yarn to the needles and therefore variables such as yarn input tension, cam setting, yarn count, etc., have no effect on the stitch length of the fabric.



For this work a "trip-tape" system of positive feed was fitted to the machine. From Fig. 22 it will be seen that a main control wheel (A), geared to the machine drive, rotates a continuous belt (B) which is tensioned and guided by jockey pulleys (C). The yarn from the package passes between a freely rotating wheel and the roughened surface of the belt and is consequently fed at a constant rate to the needles. The diameter of the control wheel can be varied to alter the speed of the belt and therefore the speed of the yarn; in this way a wide range of stitch lengths can be obtained. In commercial practice the inevitable change in yarn tension between the belt and the needles, caused by altering the stitch length, is compensated for by an adjustment to the stitch cam. However, in this work, the main purpose of the positive feed was to determine the maximum range of stitch lengths that could be obtained from a fixed setting of a non-linear cam, (i.e. without compensation for changes in tension). The results of the tests are shown later in Chapter 5.

(d) Fabric Take-down Tension

Applying a tension to the fabric during the knitting process is an important function of any knitting machine, as this ensures reliable clearance of the loops from the needles. The techniques of applying the fabric tension varies for different machines; on hand driven machines the tension may be effectively applied by simply hanging weights on a fabric clip, but on circular power driven machines a complicated roller mechanism is necessary, as the tension must be applied continuously. If the machine is the rotating cylinder type, where the fabric revolves,

an extra mechanism is needed to roll up the fabric as it is being knitted.

Although provision is made for changing the tension in the fabric, the take-down mechanisms on power driven machines are not usually calibrated and there are no simple techniques for measuring the tension in the fabric.

It was considered important, in this work, to use a take-down unit that could be calibrated accurately so that the effects of known fabric tensions on stitch length and needle forces, could be compared when using different cam systems.

The existing fabric take-down tension mechanism on the machine used in this work offered no scope for calibration and, rather than attempt to modify it to suit our requirements, it was considered more convenient to replace it with a simple gravity take-down unit.

. . . A gravity take-down unit was designed in a similar manner to the one used by Knapton (8) and is shown in Fig. 22. It is a simple arrangement comprising two pillars mounted between two discs and supporting a freely sliding fabric clip (D). The top disc was screwed to the knitting cylinder and the bottom disc mounted on a self-aligning bearing, which was fixed to the floor; this gave a good alignment of the take-down unit as the cylinder rotated. The fabric was attached to the clip and provision was made so that a large range of weights could be attached to it.

This system would obviously be unsuitable for commercial machines as it was necessary to stop the machine and raise the fabric when it reached the floor; however, for the relatively short tests in this work the system proved to be quite adequate.

FIG 23 . DIAGRAM SHOWING ORIGINAL PLAIN BEARING

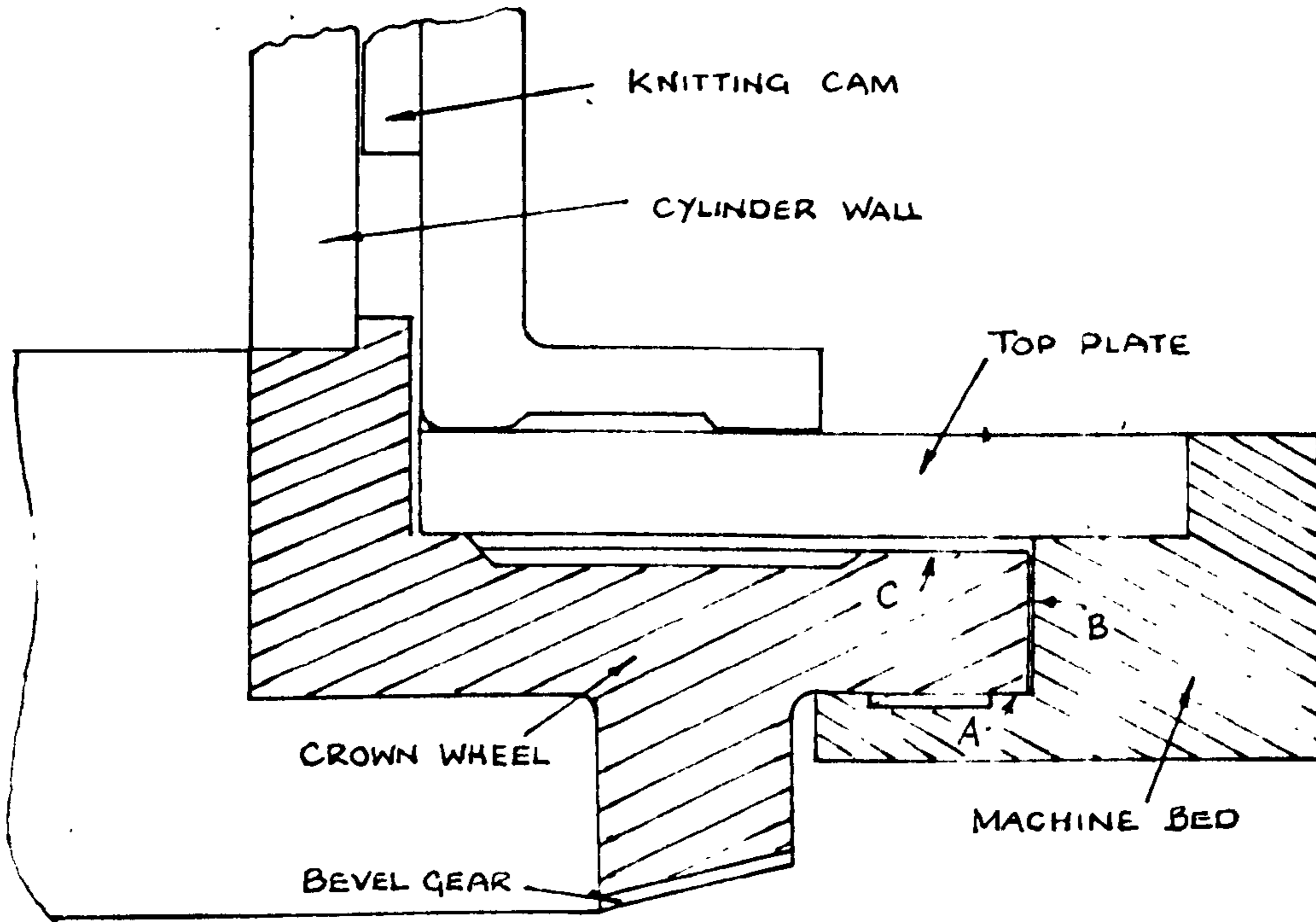


FIG 24

MODIFICATION USING BALL RACES

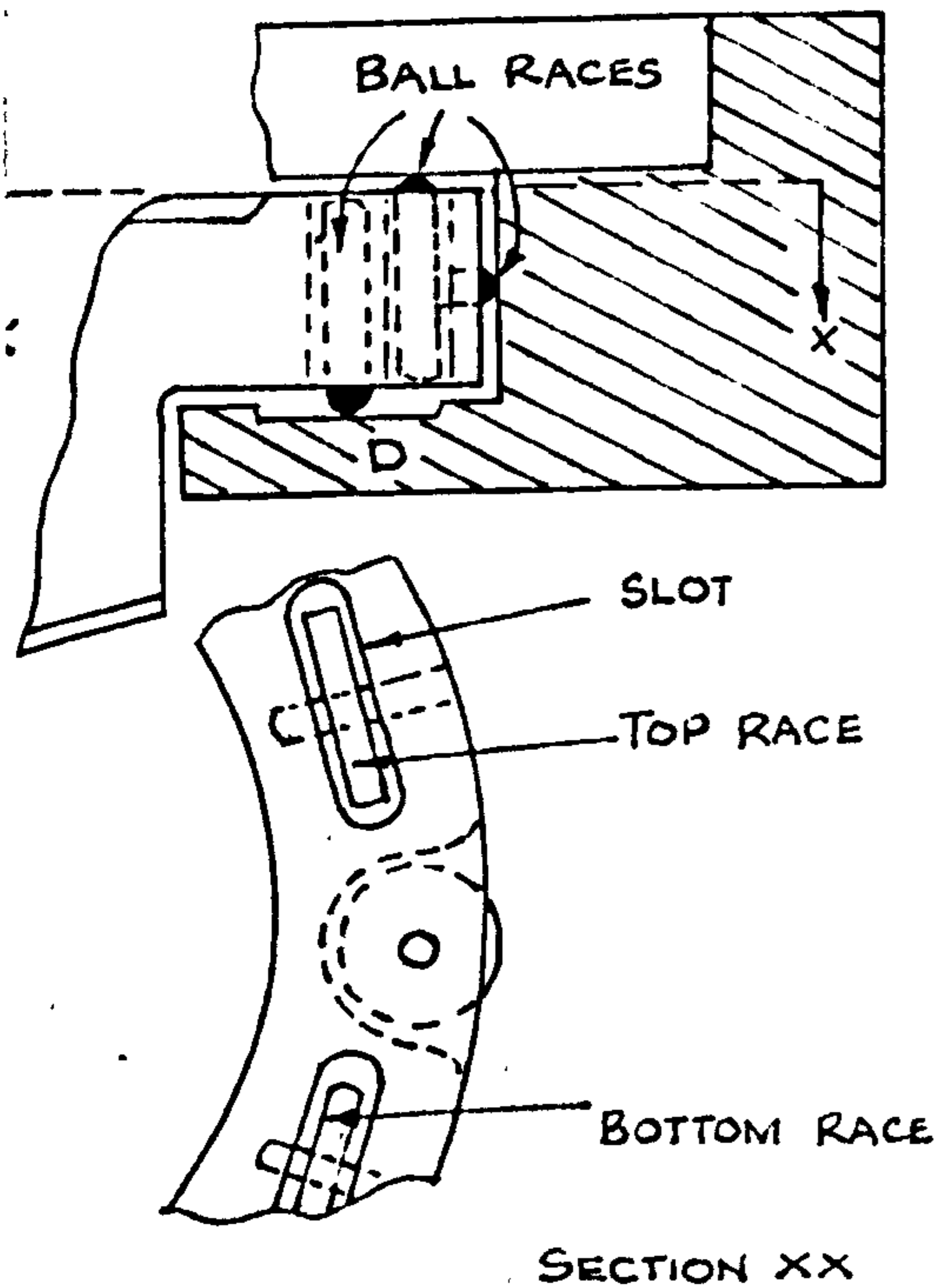
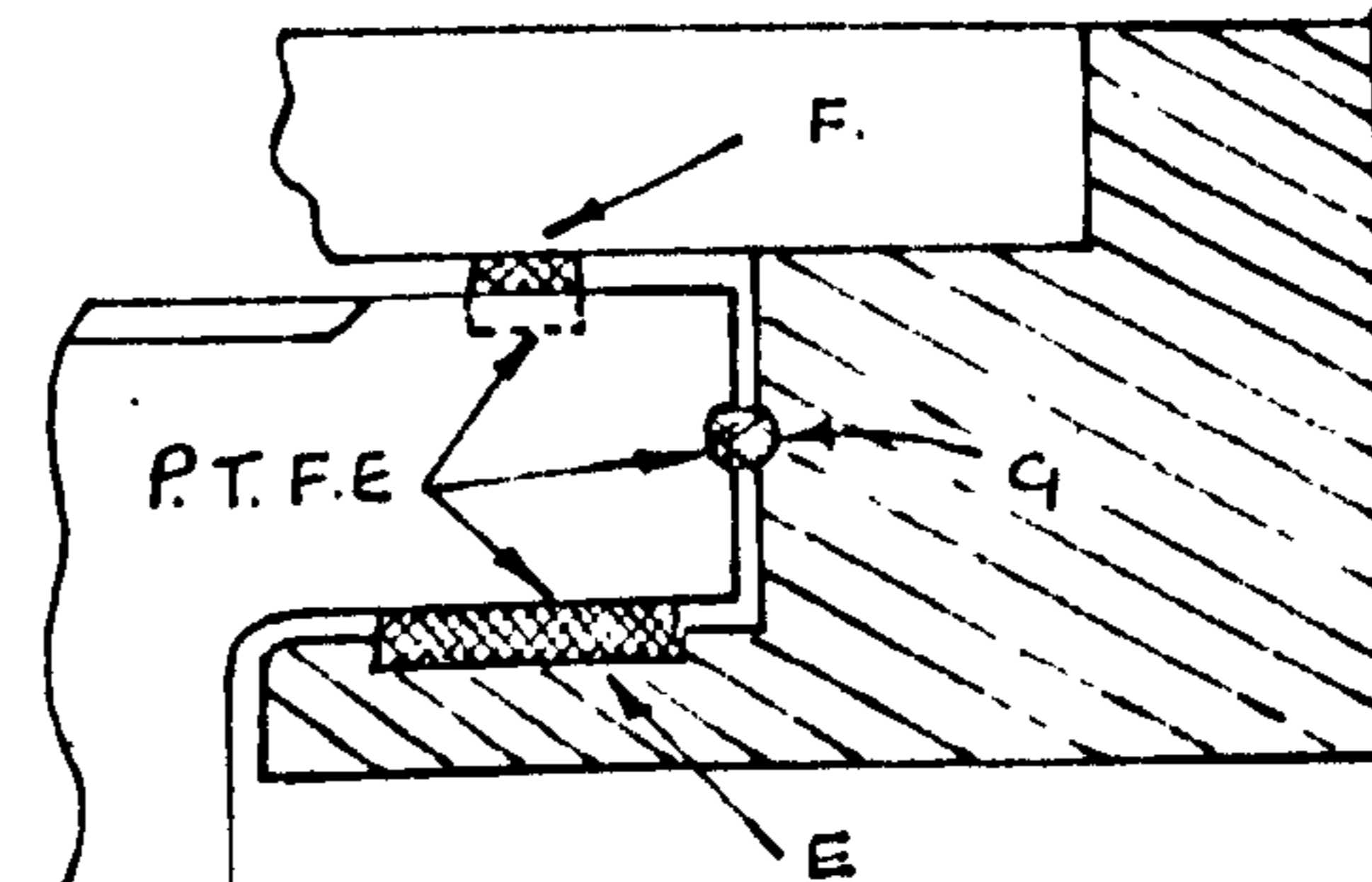


FIG 25

MODIFICATION USING P.T.F.E.



B. Modifications to the Machine to enable satisfactory High Speed Tests and Machine Torque Measurements to be made

1. Examination of the moving parts of the Machine

To provide a complete analysis of the various high speed cam systems, it was necessary to adapt the knitting machine to allow high speed tests and reliable machine torque measurements to be made.

Examination of the original moving parts of the machine (see Fig. 23) shows that the knitting cylinder, complete with sinker ring, is screwed to a bevel geared crown wheel, and this assembly is fitted into a bearing system in the machine bed. The main driving shaft for the machine, which is fitted with a bevel gear and is belt driven from a 1 h.p. motor, rotates the crown wheel, knitting cylinder and sinker ring assembly.

Examination of the bearing system (Fig. 23) reveals that it is essentially a plain bearing, i.e. the crown wheel has direct metal to metal contact with the bearing surfaces of the machine bed. The total weight of the crown wheel, cylinder and sinker ring act on the bottom bearing surface (A), the periphery of the crown wheel exerts thrust forces on the radial bearing surface (B), and the tendency of the crown wheel to lift, due to the upward thrust motion of the bevel gear drive, is prevented by the bearing surface (C) of the machine top plate.

On small diameter knitting machines, ranging from approximately 2 - 8 ins. diameter, this type of bearing system is found to be adequate for commercial purposes, since the speeds of knitting machines are reasonably slow.

In this work, however, where speeds much higher than commercial speeds were required and reliable measurements of torque were essential, it seemed likely that the existing bearing system would prove inadequate.

Preliminary high speed tests and static torque measurements were made to assess the suitability of the plain bearing system.

## 2. Preliminary High Speed Tests and Static Torque Measurements

The machine was fitted with a high speed motor and a variable speed system, and run at speeds approximately three times that of commercial speed, without a cam in position.

Observations from short running tests showed that most moving parts functioned satisfactorily; for example, the sinker ring unit, although running on a plain cast iron radial bearing, gave no signs of over-heating, and the sinkers functioning on linear cam systems moved in and out without damage or breakage. This satisfactory performance was due to small radial friction forces and the relatively low angles of the sinker cams.

The bevel gearing systems on the main drive to the crown wheel and on the positive feed also proved to be satisfactory at the high speeds.

As expected, the main adverse effect was that of the machine bed which, after only short periods of running at high speed, became quite hot. It was realised that continuous running could lead to bearing seizure and, therefore, damage to other parts of the machine. Although

this problem might be solved by improving the lubrication system, the most direct solution was to reduce the high friction forces by improving the bearing system.

Measurements of static machine torque were achieved by attaching a lever arm, one foot long, to the main drive of the machine and hanging weights on the arm until the knitting cylinder started to rotate. It was found that the torque required to start rotation of the crown wheel and cylinder on the plain bearing was 10 lbs.ft.

With the 45° linear cam system in position and a moderately high tension applied to the yarn, the extra torque required to turn the machine was found to be approximately 0.25 lbs.ft.

These simple tests showed clearly that the main proportion of the total torque was involved in rotating the crown wheel assembly on its bearing, and only a small proportion was due to the cam system. If a dynamic machine torque measuring device were used with this bearing system, the torque range of the meter would need to be approximately 0 - 10 lbs.ft. to accommodate the high values of the bearing torque. This wide range would mean that only a small sensitivity would be available for detecting the small changes in torque caused by the cam effects.

Examinations of the bearing system showed that large thrust forces were also likely to occur on the periphery of the crown wheel and on the machine top plate. The thrust forces would lead to variation in the torque of the bearing system and these inconsistent values might be large enough to obscure the torque effects when changing from one cam to another, or when altering the knitting variables on a particular cam.

It appeared, therefore, that for suitable measurement of dynamic torque, a reduction of friction forces in the bearing torque was necessary, so that a lower range and, therefore, more sensitive torque device could be used. The bearing torque must also be reasonably constant.

From these preliminary investigations it was clear that to obtain satisfactory high speed running and reliable measurements of machine torque, a major modification to the bearing system would have to be made.

### 3. Modifications to the Bearing System

#### (a) Introduction

The ideal bearing system for this machine would be a complete ball race system, which would contain and support the large crown wheel. Such a bearing would solve both high speed and torque measuring problems. The ball races would be perfectly aligned to ensure even loading of the crown wheel and to give free movement, and this would give large reductions to the friction forces and therefore no overheating of the bearing would occur at high speeds. The reduced friction forces would result in a lower running torque, which would allow a low range, sensitive, torque measuring transducer to be used and, as the thrust forces would be reduced to a minimum, a constant torque value would be obtained.

During discussions with bearing manufacturers it was pointed out that a complete ball race system of this type would have to be made specially to suit this particular crown wheel; alternatively the existing crown wheel could be redesigned to suit the standard bearing. Either of

these modifications would necessitate a redesign of the machine bed, as examinations revealed that there was insufficient cross section material to fit the housing of the roller race bearing system.

After consideration of the expense and the time required to provide the machine with this ideal bearing system, it was decided that a compromise should first be considered; a bearing system, for example, that could be fitted to the existing parts of the machine and yet would allow short high speed tests and reliable measurements of dynamic torque to be made.

If it became evident that certain cam systems were particularly successful for high speed knitting, it might be considered worthwhile rebuilding a knitting machine so that prolonged knitting tests could be made at high speeds. Such a machine could be designed to incorporate several other factors apart from the complete ball race system: For example, an improved bearing for the sinker ring unit; a more compact gear system completely immersed in oil; a take-down unit with a mechanically-balanced, fabric roller mechanism; faster stop motions; automatic yarn package changes, etc.

(b) Design of a compromise ball race bearing system

As an alternative to the ideal bearing system, it was considered that an improvement to the plain bearing might be achieved by making use of separately mounted ball races.

To achieve this, slots were cut in the crown wheel so that ball races could be mounted freely, to provide a low friction running edge on the bottom, top and sides of the





bearing surfaces. The positioning of the slots and the mounting technique for the ball races are illustrated in Fig. 24 (Page 63). Only three ball races were fitted to each running surface, as a larger number would have presented extremely difficult problems of alignment.

It was realised, from the dimensions of the bearing cross section (D, Fig. 24), that a ball race of diameter larger than  $9/16$ " could not be used unless major modifications were made to the machine bed. A small increase in ball race diameter could have been achieved by machining the surfaces of the machine bed, but for substantial increases in the bottom and top ball race diameter a new machine bed with thicker cross section would be necessary. Calculations of the loading weight of the crown wheel and estimated maximum speeds, however, indicated that a ball race of  $9/16$ " diameter would be satisfactory. These races were therefore mounted in position and the crown wheel bearing edges were relieved by  $1/32$ " so that the crown wheel was free from the bearing surfaces but made contact through the ball races.

The ball races which acted on the top and bottom bearing surfaces were fitted with carbonised mild steel tyres, to avoid the scuffing of the bearing that would otherwise have occurred.

#### Static torque measurements and high speed tests

Preliminary static torque measurements showed a considerable improvement over those of the original plain bearing system. The torque now required to start the machine was only 4 lbs.ft. and therefore an approximate torque range of 0 - 4 lbs.ft. may be used to give more

than twice the sensitivity. It was also considered that a constant value of running torque would be achieved, due to the control on the crown wheel periphery.

High speed tests, however, were not so satisfactory, and the limitations of the compromise bearing system became evident. Examinations of the ball race tyres after a period of high speed running showed that considerable wear was taking place and, after further periods of running, the tyres became distorted and damage to the ball races occurred. Experiments without tyres produced excessive scuffing, as had been expected, and consequently the ball races were damaged.

It became evident also that certain conditions of machine running could result in breakage of the ball races and it was noticed that, on one or two occasions, sudden acceleration or deceleration of the machine caused sufficient thrust forces to break a side ball race. It was also observed, during preliminary experiments on the effects of knitting variables, that breakage of any of the ball races could result from a high yarn count or heavy take-down tension. It was realised, therefore, that although this bearing system was probably suitable for reliable dynamic torque measurements, the excessive thrust forces caused by varying knitting conditions and the sudden speed changes were exceeding the safe limit of the ball races: this together with the ball race tyre problem made the system unreliable.

These problems could be overcome but would necessitate major modifications to the machine. For example, the problem of breakage due to thrust forces could be overcome by using much larger ball races, and more robust tyres.

Alternatively, a system of separately mounted ball bearings could be used as these would overcome the problems of scuffing and may give satisfactory high speed performance. However, if these adaptations were to be applied, it might well be debatable whether to go a stage further and fit a complete ball race system which, as discussed previously, would clearly be the ideal bearing system.

Before any major modifications were made to the machine to fit either of these bearing systems, it was considered worthwhile to try a low friction material such as P.T.F.E. as a bearing surface, since this would not involve machine modifications and yet might provide the desired requirements for high speed tests and reliable dynamic torque measurements.

(c) Design of a P.T.F.E. Bearing System

P.T.F.E. is the best known of the fluorated plastic materials and has certain properties which are made use of in the chemical and electrical industries. The mechanical properties of P.T.F.E. are limited compared with other plastic materials, but it has one outstanding property that is frequently made use of by the engineer; it has the lowest value of friction of any known solid. For this reason it was considered that it might be used successfully as a plain bearing surface in the knitting machine, although the suitability of certain of its mechanical properties, such as compression strength and resistance to wear at high speeds, could only be determined by a trial run.

The application of P.T.F.E. on the existing bearing

surfaces of the machine bed was a relatively simple procedure (Fig. 25, Page 63). The oil groove (E) which runs in the centre of the bottom bearing surface, is  $\frac{1}{2}$ " wide and  $\frac{1}{32}$ " deep and was found to be a convenient place to mount P.T.F.E. washers. The washers were cut from sheet P.T.F.E. material to give dimensions of  $\frac{1}{16}$ " x  $\frac{1}{2}$ " x 6" and were fitted adjacently in the oil groove to give a continuous P.T.F.E. surface protruding  $\frac{1}{32}$ " above the crown wheel bearing edge. To provide a low friction surface from the top edge of the crown wheel, three pads were fitted into the slots that had been made for the roller races. The pads were clamped by screws and protruded  $\frac{1}{32}$ " to give the correct distance for contact with the machine top plate (F).

To provide a low friction surface for the crown wheel periphery, P.T.F.E. rod of  $\frac{1}{16}$ " diameter was acquired. A semi-circular groove was cut in the crown periphery and a continuous length of P.T.F.E. rod was pressed into the groove, thus providing a  $\frac{1}{32}$ " protrusion to give the correct distance for contact with the radial bearing surface (G).

#### High Speed Tests and Static Torque Measurements

Preliminary high speed tests proved to be extremely successful, with maximum speeds being maintained for fairly long periods and no signs of the bearing overheating.

Micrometer measurements of the thickness of the P.T.F.E. material showed that no change had occurred after these extensive speed tests, showing clearly that the mechanical properties, such as compression strength and resistance to wear, were quite satisfactory for this

particular application. Other tests, such as sudden changes in acceleration of the machine and substantial increases in yarn count or fabric take-down tension, showed no adverse effects on the P.T.F.E. bearing materials.

From these preliminary tests it was evident that P.T.F.E. provided a more reliable bearing system than the ball race system, and also appeared to be completely satisfactory for the high speed experiments necessary in this work.

Static torque measurements showed that this bearing system compared favourably with the ball race system, as the static torque required to start the rotation of the crown wheel on the P.T.F.E. bearing was 5 lbs.ft. These results indicated that a torque measuring device with a range of 0 - 5 lbs.ft. could be used, and this range would probably cover all the values of dynamic running torque obtained in these experiments. This range would also provide twice the sensitivity compared with the range of 0 - 10 lbs.ft. that would have been necessary with the existing cast iron bearing system.

Provided that reasonably constant values of running torque were obtained, which seemed likely as positive control of the crown wheel periphery had been achieved, then this P.T.F.E. bearing should give reliable dynamic torque measurements.

#### 4. Modifications to the machine drive to suit a dynamic torque measuring transducer

A torque transducer and an appropriate meter to amplify the signal were acquired from Saunders-Roe, specialists in

the design and manufacture of torque measuring devices. A complete description of these units will be given in the following chapter.

Preliminary tests, with the torque transducer mounted in various positions on the machine, showed that the most suitable drive arrangement for satisfactory dynamic torque measurements was a direct alignment of the motor, torque transducer and crown wheel bevel gear. To achieve this the original casting and machine drive were removed, and a rigid frame was made to support the electric motor, a variable speed system and the torque transducer - the frame being bolted to the bed of the machine and to the floor. These units were mounted and aligned to the frame, and gave direct drive to the crown wheel gear, (Fig. 26). The transducer was mounted between two castings, fitted with self-aligned race bearings, and the shaft was supplied with a split pin to protect the transducer against any excessive torque value, such as might result from a needle seizure.

The electric motor used in this work was a 2 H.P. 3 phase motor with a maximum shaft speed of 1250 r.p.m. As the bevel gear ratio from the main drive shaft to the crown wheel was a 6:1 reduction, the maximum speed of the knitting cylinder was 200 r.p.m. thus providing a maximum knitting speed of approximately three times commercial speeds.

A reliable variable speed system was a necessity for this work for a number of reasons. It enabled the machine to be brought to maximum speeds gradually, thereby avoiding excessive starting torque; it allowed a series of speed tests to be made from well below commercial speed to the

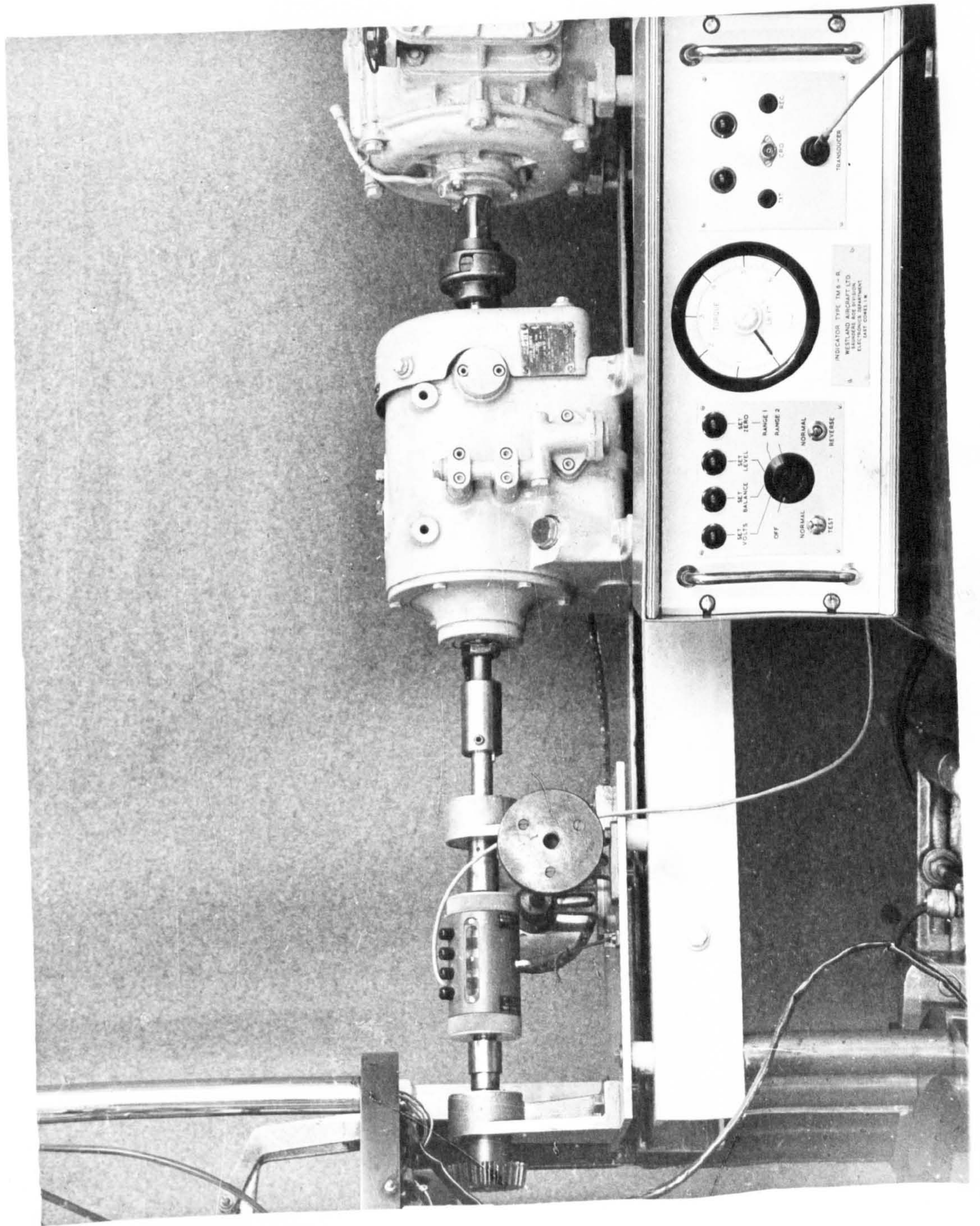


FIG. 26.

maximum speed; most important of all it was used to determine the precise speeds at which the various cam systems failed to produce satisfactory fabric or caused needle butts to break.

A Carter hydraulic variable speed gear was used which proved to be extremely accurate and reliable. It gave constant speeds at any speed between 0 - 1250 r.p.m. and developed a 55 lbs.inches torque at any speed, thereby ensuring that no changes in torque on the knitting machine would affect the speed of the gear.



CHAPTER 4

DESCRIPTION OF THE METHODS USED TO MEASURE  
THE PERFORMANCE OF THE VARIOUS CAMS

CHAPTER 4

DESCRIPTION OF THE METHODS USED TO MEASURE  
THE PERFORMANCE OF THE VARIOUS CAMS

Preamble

In the previous chapter it has been shown how the knitting machine was modified so that high speed tests and machine torque measurements could be made.

This chapter is concerned with the methods used to measure the performance of the various cam systems.

In Section A, a description of the torque measuring device is given and a few torque values are examined. Section B describes a force device which was designed and constructed for measurements of the forces directly on the cam system. This force device was used partly to supplement the torque value results, but also to provide more detailed information of non-linear and linear cam systems.

A. Method of Measuring Dynamic Machine Torque

1. Introduction

The measurement of dynamic torque is frequently used as a means of determining the performance of a variety of machine applications in the engineering field, and many industrial firms specialise in the manufacture of different types of accurate torque measuring devices.

It has been shown in Chapter I that both Knapton (8) and Wignall (10) designed and developed systems for measuring dynamic torque, and their results indicate that the measurement of torque is a suitable means of obtaining information about the forces on knitting cams and the effect of knitting variables on torque, in circular weft knitting machinery.

The measurement of dynamic torque is usually obtained by measuring the angle of torsion in a certain length of driving shaft, and this angle of twist is proportional to the transmitted torque, viz.

$$\frac{T}{J} = \frac{G\theta}{l}$$

where T = torque transmitted through the shaft;

J = polar moment of inertia of cross section of shaft (i.e.  $J = \frac{\pi r^4}{512}$  for a solid shaft).

G = modulus of rigidity ( $12 \times 10^6$ );

$\theta$  = angle of twist;

l = length of shaft.

This principle forms the basis of many different types of torque measuring device, but the methods of measuring the angle of twist ( $\theta$ ) are various. For example, the value of  $\theta$  may be measured from a change in :

- (i) resistance of strain gauges mounted on a shaft,
- (ii) capacitance between two condenser plates,
- (iii) relative positions of a slotted disc observed by photo-electric cells,
- (iv) air gap of a small differential transformer.

Examinations of a number of commercial products, showed that Saunders-Roe included in their torque measuring equipment, units which were specifically designed to measure very low torque values - the measurement of  $\theta$  being obtained from a change in resistance of strain gauges. A torque transducer, with a choice of two torque ranges of 0 - 6 lbs.ft. and 0 - 3 lbs.ft., and an indicating unit were obtained, therefore, from this manufacturer (shown in Fig. 26).

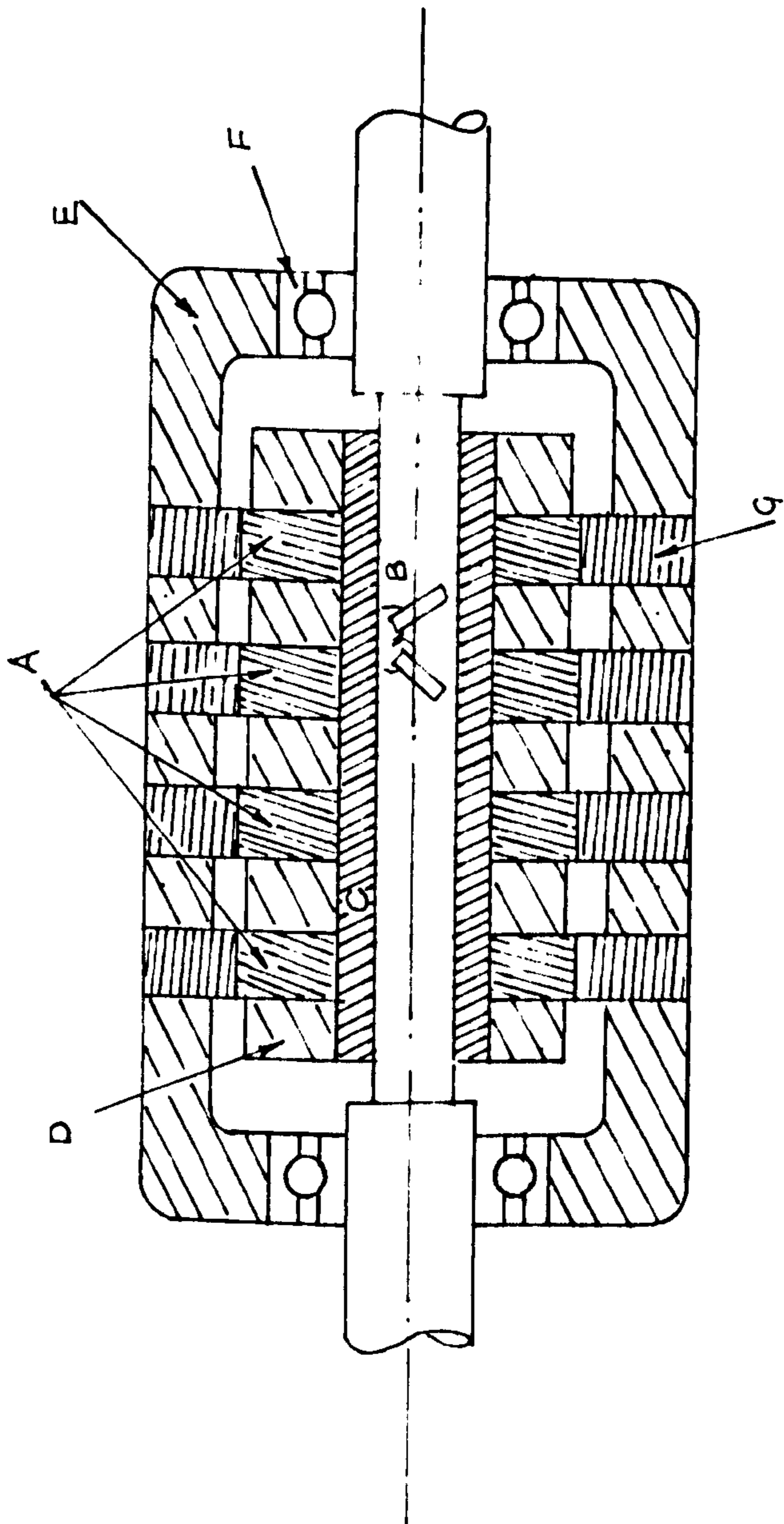
## 2. Description of the Torque Transducer

An assembly drawing of the torque transducer is shown in Fig. 27.

Torque is transmitted through a high tensile steel shaft, to which is cemented a network of bonded foil torsion strain gauges. The strain gauges are connected in such a way that an electrical bridge unbalance is created by torsional strain. The arrangement is such that the shaft strains due to bending, thrust and temperature are self-cancelling and do not contribute to

FIG. 27

TORQUE TRANSDUCER.



the electrical output. The strain gauges are protected by a tough, resilient, resin film and are covered by a rotor tube which carries a set of four silver slip rings.

From an examination of Fig. 27 it may be seen that the slip rings (A) are connected to four terminals of the fully active strain gauge bridge network (B), and are fixed to the rotor tube (C) by a spin moulding technique in a dielectric (D) consisting of a thermo-setting resin with glass fibre. The resulting one-piece rotor construction has excellent mechanical and electrical properties, and cannot become loose or run eccentrically.

The rotating part supports the stator housing (E) of the transducer, which is also fabricated as a one-piece glass fibre spin moulding, on a pair of high speed light section bearings (F). The stator carries a set of eight silver-graphite brushes (G) giving two parallel contact paths per ring, the brushes being directly connected to an integral length of four core cable, which provides connection to the instrumentation system.

The very small angular deformation of the torque shaft, resulting from the torsional stress, causes a change in the electrical resistance of the strain gauges and produces an electrical output voltage from the bridge circuit, which is directly proportional to the transmitted torque.

The stator is prevented from rotating by a light cord attached to the stator and a fixed part of the machine.

Very little maintenance was necessary with this transducer apart from removing the brush dirt and

occasionally relubricating the ball bearings. The slip rings and brushes had been designed to last at least  $10^8$  revolutions before wear was sufficient to warrant renewal of these parts.

### 3. Description of the Torque Indicator

The torque indicator was used to amplify and record the voltage signal from the strain gauges. To achieve this the indicator uses a D.C. amplifier to drive the moving coil cirscale meter from the low level transducer output. The D.C. amplifier employs six transistors which are mounted in a temperature controlled block. Temperature control is effected by a circuit employing thermistors and a three-transistor balanced amplifier which maintains the temperature within very close limits.

The accurate temperature control combined with the carefully balanced design of the amplifier enables high gain and high stability to be achieved, together with freedom from long term drift. The indicator also includes a stabilised power supply which delivers power both to the amplifier circuit and to the externally connected transducer.

### 4. Preliminary Experiments with the Dynamic Machine Torque Apparatus

A few experiments were carried out to determine whether the modified knitting machine enabled reliable machine torque measurements to be made, and whether the measurements provided adequate information for analysing the performance of the various cam systems.

An immediate observation, on measuring the dynamic torque of the machine when knitting with a 45° linear cam under normal conditions of speed, yarn tension, etc., was that the torque gradually dropped as the machine began to warm up, and this decrease continued for approximately two hours before the torque value eventually settled at a constant value. It was considered that this rather long transition period of torque change may be an effect of the properties of the P.T.F.E. material, resulting in a slight change in frictional properties of the material when the temperature increased. Although it was realised that the various torque experiments could be made providing the machine had been sufficiently warmed up, it was decided to investigate the torque change to see if it was directly related to temperature changes in the machine bearing system.

(a) Measurement of the temperature of the P.T.F.E. bearing system

One of the most widely used temperature measuring instruments is the thermo-electric pyrometer. An instrument of this type was used by Seedhom (18) for his researches on spinning machines, and was made available for this experiment.

The basic principle of the thermo-electric pyrometer is that when two dissimilar wires are joined to form a complete electric circuit and two junctions are maintained at different temperatures, an electro motive force (e.m.f.) is set up, due to the algebraic sum of an e.m.f. developed between two different metals placed in contact and an e.m.f. developed between the ends of a homogeneous wire when one end is heated.



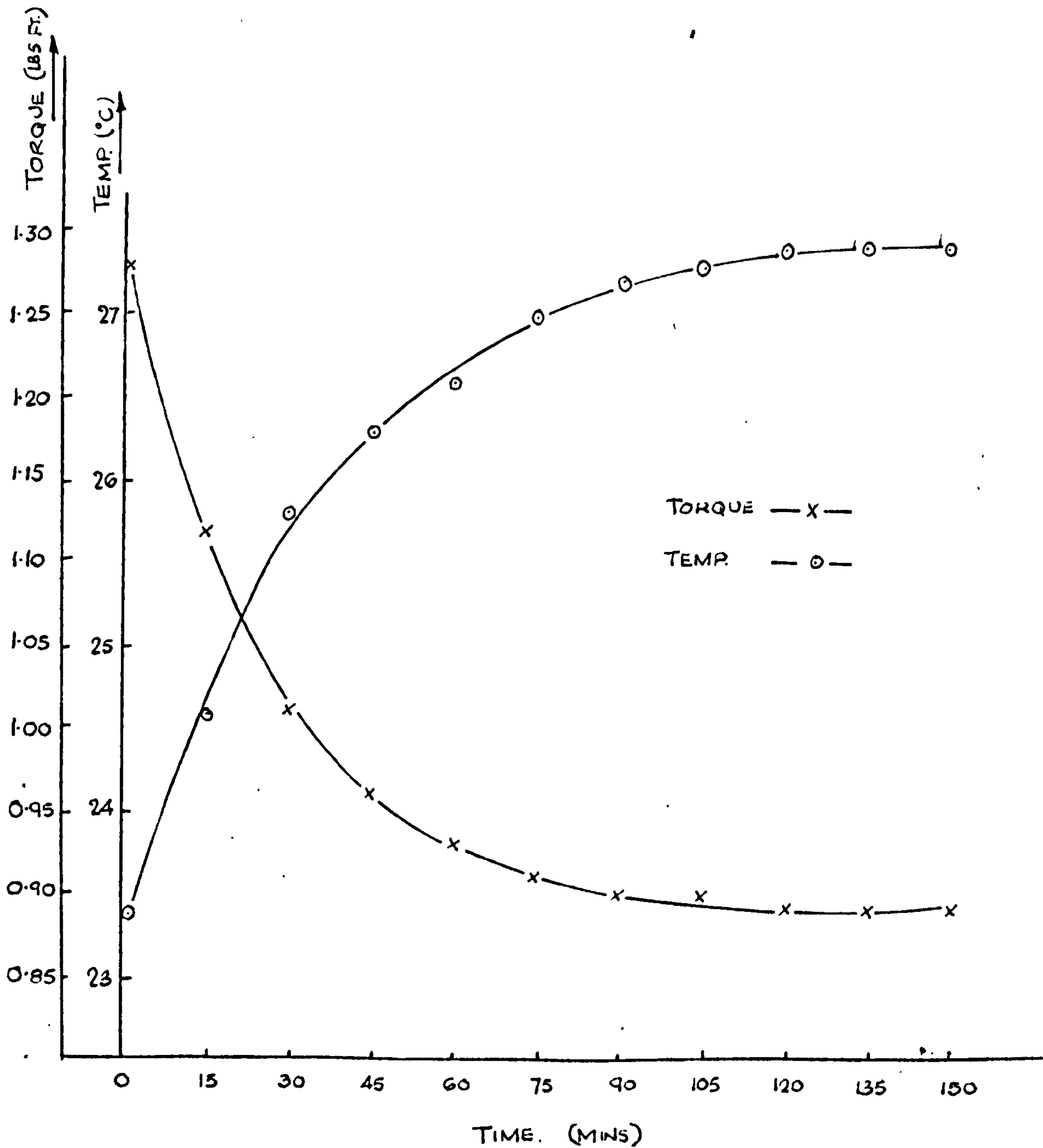
The magnitude of the e.m.f., therefore, depends upon the temperature difference of the junctions and the metal used. If the cold junction is held at a uniform temperature then the e.m.f. developed can be used to determine the temperature of the hot junction; this arrangement of conductors is called a thermocouple. The complete pyrometer consisted of the following parts,

- (i) thermocouples which were fitted into the knitting machine bearing,
- (ii) provision for controlling the temperature of the cold junction,
- (iii) an instrument for measuring e.m.f.

The thermocouples used for measuring the temperature on the P.T.F.E. bearing surface were made by twisting two dissimilar wires together and welding the ends with an electric arc system. Three very small holes were drilled in the 1/16" thick P.T.F.E. material to provide close proximity temperature measurements on the bottom bearing surface, and three holes were drilled in the cast iron periphery of the machine bed in close proximity to the P.T.F.E. The thermocouples were electrically insulated and fitted into these holes; the couple leads were brought out of the bearing and fixed to sockets that were mounted on the machine top plate.

Seedhom's compact apparatus was used to measure the temperature changes from the thermocouple sockets. The apparatus included an arm for making electrical connection to the socket positions, a controlled temperature cold junction comprising a thermoflask filled with melting ice, and a potentiometer for measuring the e.m.f.

FIG. 28.



To determine the relationship of torque and temperature of the machine bearing system, the crown wheel, without the knitting cylinder, was rotated at a constant speed of 60 r.p.m. from cold, and the torque and temperature measurements were noted every fifteen minutes. The results are shown in Table 15 and plotted in Fig. 28.

From Fig. 28 it was evident that the torque and temperature were directly related; the torque decreased as the temperature increased and after a period of two hours running, the temperature and torque became constant. It appeared, therefore, that reliable torque measurements could be made after the machine had been rotating for two hours.

(b) The effect on machine torque when the knitting conditions were varied.

A series of preliminary experiments were conducted to see whether dynamic torque measurements provided adequate information for analysing the performance of the various cam systems.

These experiments were designed to show :

- (a) The effect of a change in knitting variable on machine torque when using the  $45^{\circ}$  linear cam;
- (b) The difference in torque between a  $45^{\circ}$  linear and a  $52^{\circ}$  non-linear cam system;
- (c) The proportion of torque due to the machine's moving parts and the needle/cam reaction.

The machine was run for a period of time to ensure that the constant torque value of the bearing system had been achieved. The measurements were then taken from the

meter scale of the torque indicator, using the 0 - 3 lbs.ft. range (where 1 division = 0.05 lbs.ft.).

The torque results, shown in Table 2 below, represent the average value taken from the meter. A fluctuation of 0.05 lbs.ft. between the maximum and minimum positions of meter needle was noted at a machine speed of 60 r.p.m.

TABLE 2

Exp.	Knitting Conditions					Torque (lbs.ft.)
	Cam System	Yarn Tens. (Gms)	Take-Down (lbs)	Count (W <sup>o</sup> )	Speed (R.P.M)	
(a)	45° linear	5	2	1/28	60	1.55
		10	2	1/28	60	1.60
		15	2	1/28	60	1.70
		5	6	1/28	60	1.65
		5	10	1/28	60	1.70
		5	2	2/32	60	1.70
		5	2	2/28	60	1.75
		5	2	1/28	80	1.70
		5	2	1/28	100	2.13
		(b)	45° linear	5	2	1/28
52° non-linear	5		2	1/28	60	1.53
(c)						
(i)	No cam	-	-	-	60	1.33
(ii)	45° linear	No Yarn			60	1.50
(iii)	45° linear	5	2	1/28	60	1.55

The results from experiment (a), Table 2, showed that a change in knitting variable provided only small changes in dynamic machine torque. However, when it was considered that this was the effect of one feeder on a machine that would normally be fitted with at least eight feeders, it was clear that a change in knitting variable would have a distinct effect on the total dynamic torque on a multi-feed machine.

The results from experiment (b) were rather surprising as they show little difference in the average dynamic torque between the linear and non-linear cam system. It had been considered, previously, that the reduced impact forces of the non-linear cam system would have resulted in a more obvious decrease in average dynamic torque.

The results from experiment (c) showed that the main proportion (i.e. 85%) of the total dynamic torque was due to the moving parts of the machine (i.e. bearing system, sinker ring, gears), and further examinations showed that most of this was caused by the crown wheel and knitting cylinder acting on the bearing surface, despite the large reductions in friction produced by introduction of P.T.F.E. material.

As the additional effect of knitting with a  $45^{\circ}$  linear cam represented such a low proportion (15%) of the total dynamic torque, it was evident that the small changes in torque caused by, for example, changing from a linear to a non-linear cam, would be very difficult to obtain with any real accuracy. Attempts to amplify these results further, using a pen recorder or an oscilloscope did not provide much improvement as the high

proportion of bearing torque restricted any substantial increases in sensitivity.

Examinations of Knapton's dynamic torque results, obtained on a 4" diameter knitting machine, showed that the proportion of machine moving parts to needle/cam force was in the order of a 50-50 ratio, and this was satisfactory for reasonably sensitive and accurate measurements of torque to be made. It was evident from the experiments, therefore, that the 8" diameter machine with its much heavier crown wheel and larger friction contact areas in the bearing, was not as suitable for detecting small changes in dynamic machine torque as a small diameter machine.

It was also considered that even if a complete ball race system were fitted to the machine (discussed in Chapter 3) it would not reduce the machine parts ratio sufficiently for sensitive measurements of dynamic torque to be made. The answer to successful dynamic torque measurements on large diameter machines might be to use an air bearing, as this would eliminate the bearing forces completely. It was realised that as these bearings are still in the early stages of development, the fitting of such a bearing would represent a research project of its own.

Despite the lack of sensitivity the torque measurements were useful when certain knitting variables were increased, as the results which are shown in Chapter 5 provided adequate information to serve as a guide to the performance of various cam systems. However, for more detailed information, especially as experiment (b), (Table 2), seemed to indicate that there is little difference

between the average torque of a  $45^{\circ}$  linear and a  $52^{\circ}$  non-linear cam system, it was considered worthwhile to develop a more sensitive and direct method of measuring the needle forces on the cams, which would not involve the measurements of the machine's moving parts.

B. A direct Method for Measuring the Needle Forces -  
Strain-Gauge Force Device

1. Introduction

A direct measurement of the needle forces was obviously more desirable than measurements using a dynamic torque(machine torque)device, as the direct force results would be completely independent of and, therefore, unaffected by, the machine's bearing system and other moving parts of the machine.

It has been explained in Chapter 1 that Wignall measured the forces directly on separate parts of a linear cam system using a mechanical spring balance arrangement, and with this device obtained interesting information about the proportions of average forces at different stages of loop forming.

In designing an apparatus for this work it was considered necessary to go beyond the spring balance system devised by Wignall, and to design and construct an apparatus which would indicate the fluctuations in the force value as well as the average force values. From this it was hoped to be able to compare the fluctuations produced by linear and non-linear cam assemblies, and also to detect the effect of individual needles operating upon the cam systems.

## 2. Description of the Strain Gauge Force Measuring Device

### (a) Design of the Mechanical Apparatus

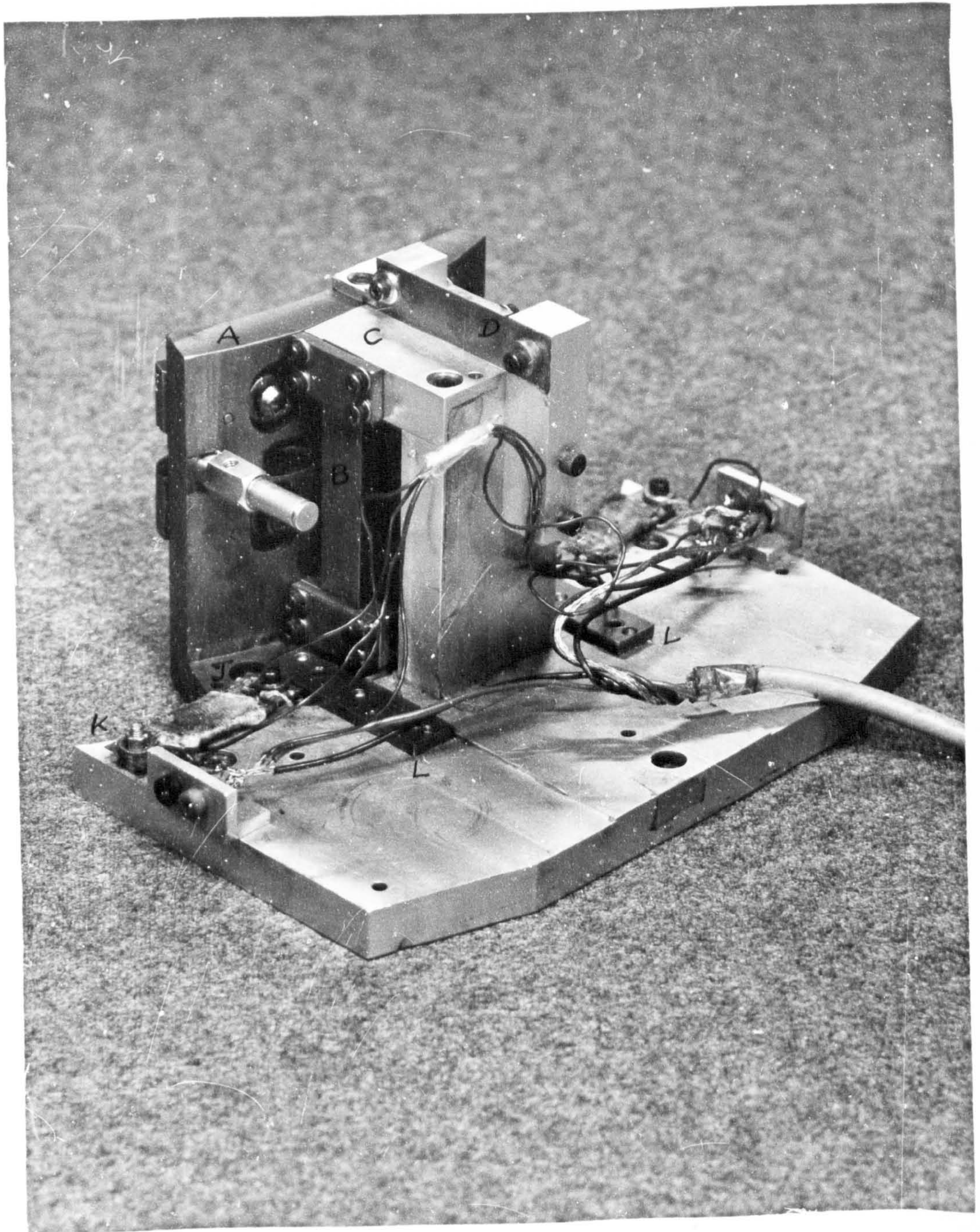
To obtain a direct measurement of needle force, it was considered that the cams could be suspended so that no friction|contact would occur between the cam and other parts of the machine. In this way the needle butts would move the cam in a horizontal direction and the measurement of force could be obtained from the cam displacement.

To avoid the complication of suspending individual cam systems with free movement, it was realised that a convenient method would be to suspend the complete cam section assembly - thus the cams could be mounted separately in the normal way. The jig designed and constructed to hold the cam section is shown in Fig. 29 and Fig. 30.

From Fig. 29 it may be seen that the cam section (A) is suspended by two 0.010" feeler strips (B) from the jig (C) - the feelers are clamped to the jig at the top, and to the cam section at the bottom. A third 0.010" feeler strip (D) is mounted horizontally at the top of the jig and cam section. The arrangement of the three feeler strips allowed horizontal movement of the cam section, but restricted any vertical or radial movement. To ensure horizontal freedom, 0.010" clearances were provided between the cam section and jig at positions E and F, and a 0.005" clearance between the base of the cam section and the machine bed (G), Fig. 30. This arrangement satisfied the condition that no friction|forces of cam to parts of the machine should occur.



FIG. 29.



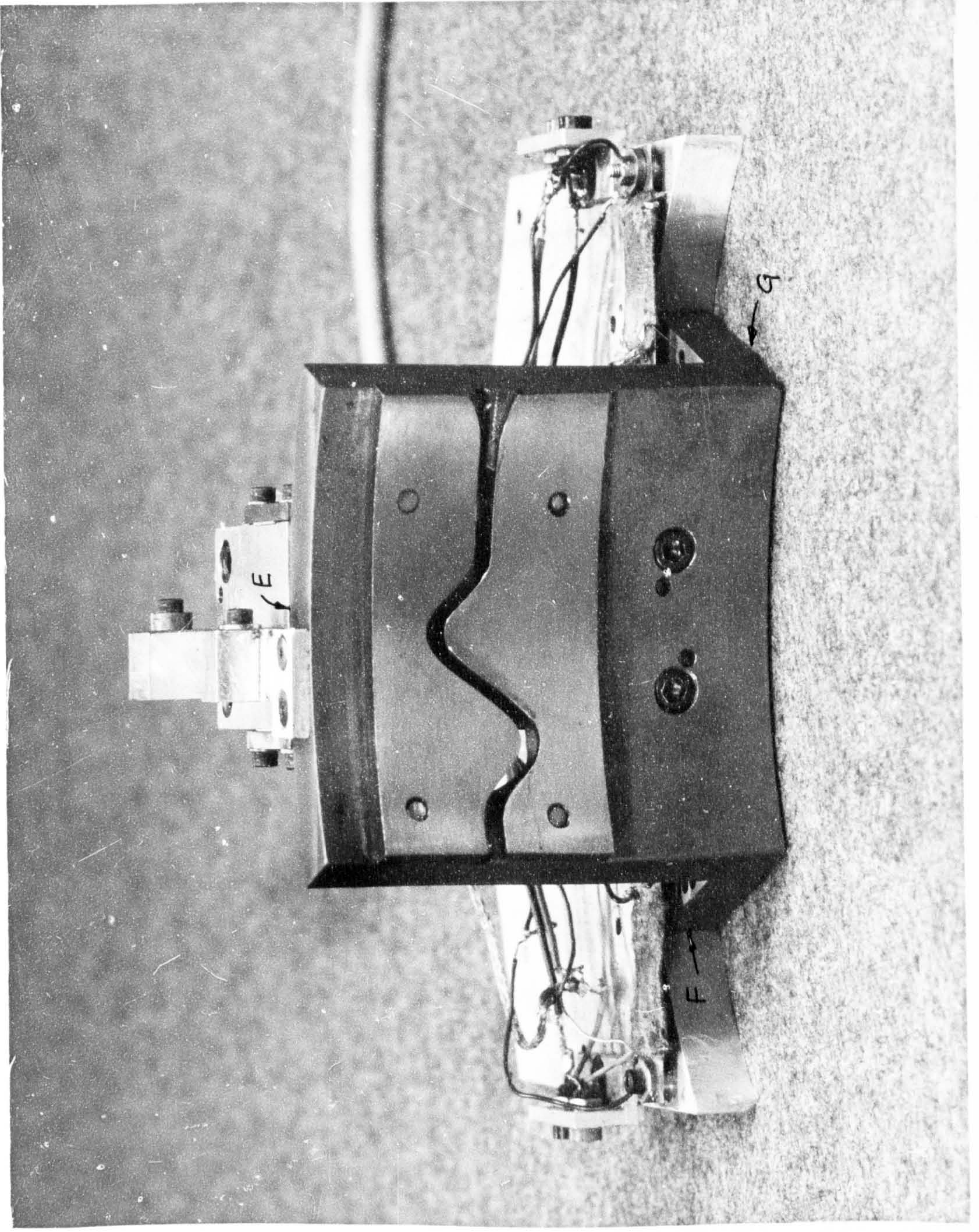


FIG 30

Three location pegs were accurately fitted to the jig and machine bed when the apparatus was correctly aligned, with a 0.005" clearance between the cam face and the knitting cylinder: This guaranteed accurate replacement of apparatus after it had been removed.

It was realised that although the horizontal displacement must be measured accurately, only approximately 0.005" horizontal movement of the cam section was possible before frictional contact occurred between cam face and knitting cylinder.

In order to obtain accurate measurements of force from this short displacement movement, it was decided to mount 0.004" steel feeler strips between the cam section and the fixed jig (J and K, Fig. 29), and to fit strain gauges to the feelers to measure the strains resulting from the needle forces.

In this way the complete cam section was maintained virtually rigid in all directions, but the strain gauges could detect sensitively and accurately the movements within the 0.004" feeler strips.

The complete apparatus of jig and cam section could easily be removed from the machine for cam setting adjustment or for changing the cam system; to guard against the possibility of upsetting the strain gauge feeler strips when changing a cam system, keeper plates (L, Fig. 29), were fitted which ensured complete rigidity of the cam section and the jig; these keeper plates could easily be removed when actually carrying out a knitting experiment.

(b) Description of electrical apparatus

In the last twenty years the strain gauge has become a valuable engineering tool and in many applications it has enabled rapid advancement in project development. It is cheap, easy to make, its mass is negligible and it is so thin that it can be regarded as part of the surface of a material. In addition, the dimensions of the strain gauge can be adjusted to suit a wide variety of applications - for example, extremely small strain gauges have been used to measure the stress at the bottom of a screw thread; on the other scale larger strain gauges have been used to examine the deformation of a submarine hull under shock.

In this work the size of the strain gauge was determined by the space available for mounting the feeler strips, and relatively small strain gauges ( $\frac{1}{2}$ " x  $\frac{1}{4}$ ") were found to be suitable.

The strain gauge is a small strain sensitive element, and is made by arranging an electrical conductor (i.e. thin resistance wire) into a grid and then mounting the wire grid on to thin paper.

The strain gauges were fixed to the feeler strips by an even layer of durafix glue and were allowed to set for two days under a one pound weight. In this way uniformity of gauge to feeler was obtained and after setting had occurred the strain gauge and feeler would behave as a single unit, in that dimension changes made to the feeler strip would equally affect the dimensions of the strain gauge.

The application of needle force to a cam system would tend to move the cam section in a horizontal direction, resulting in one feeler strip being in tension and the other

in compression. The changes in dimension would, therefore, be directly transferred to the actual strain gauges and would result in resistance changes, according to the following relationship

$$R = \frac{\rho l}{a}$$

where R = resistance in ohms;  
 $\rho$  = specific resistance of conductor material;  
l = length of conductor;  
a = area of conductor.

To ensure that the recorded changes in resistance were due entirely to the horizontal compression or tension strains, it was necessary to guard against other factors which were known to affect the resistance of a strain gauge, (19), viz. changes in temperature, bending motions, and moisture content.

(1) Compensation for Temperature Changes

It is known that strain gauges react not only to strain but also to variations in temperature. Temperature changes can raise the resistance of the gauge wire, expand the wire or expand the feeler strip, all of which would alter the recorded strain and cause large errors in the results.

To compensate for these effects, "dummy" strain gauges were fixed at right-angles to the active strain gauges, on both the compression and tension feeler strips. In this way the temperature compensating gauges would not record the horizontal strain and yet they would be close enough to record the same

temperature and, by connecting dummy and active gauges in series, any changes in temperature would result in both gauges changing resistance together and therefore no errors would be incurred.

(ii) Compensation for bending

Another factor which would affect the validity of the recorded resistance would be slight bending movements of the feeler strips when the needle force was applied to a cam system. To compensate for this it was necessary to fix extra strain gauges to the feeler strips diametrically opposite the active and temperature gauges. By connecting these extra gauges in series, any change in resistance of a gauge due to bending would be completely compensated for by the opposite gauge - the horizontal strain would now be recorded by two active gauges on both compression and tension sides of the force device.

(iii) Protection against moisture

Moisture from the atmosphere can have detrimental effects on the strain gauge, as it causes the cement to swell and reduces the insulation between the windings of the filament. Both these effects could cause large changes in the resistance of the strain gauge, which could exceed the variation in resistance due to strain.

The strain gauges were protected from moisture by first drying them thoroughly with a fan heater and then coating them with several layers of digel wax.

CIRCUITRY FOR THE FORCE MEASURING DEVICE

FIG 31. THEORETICAL CIRCUIT.

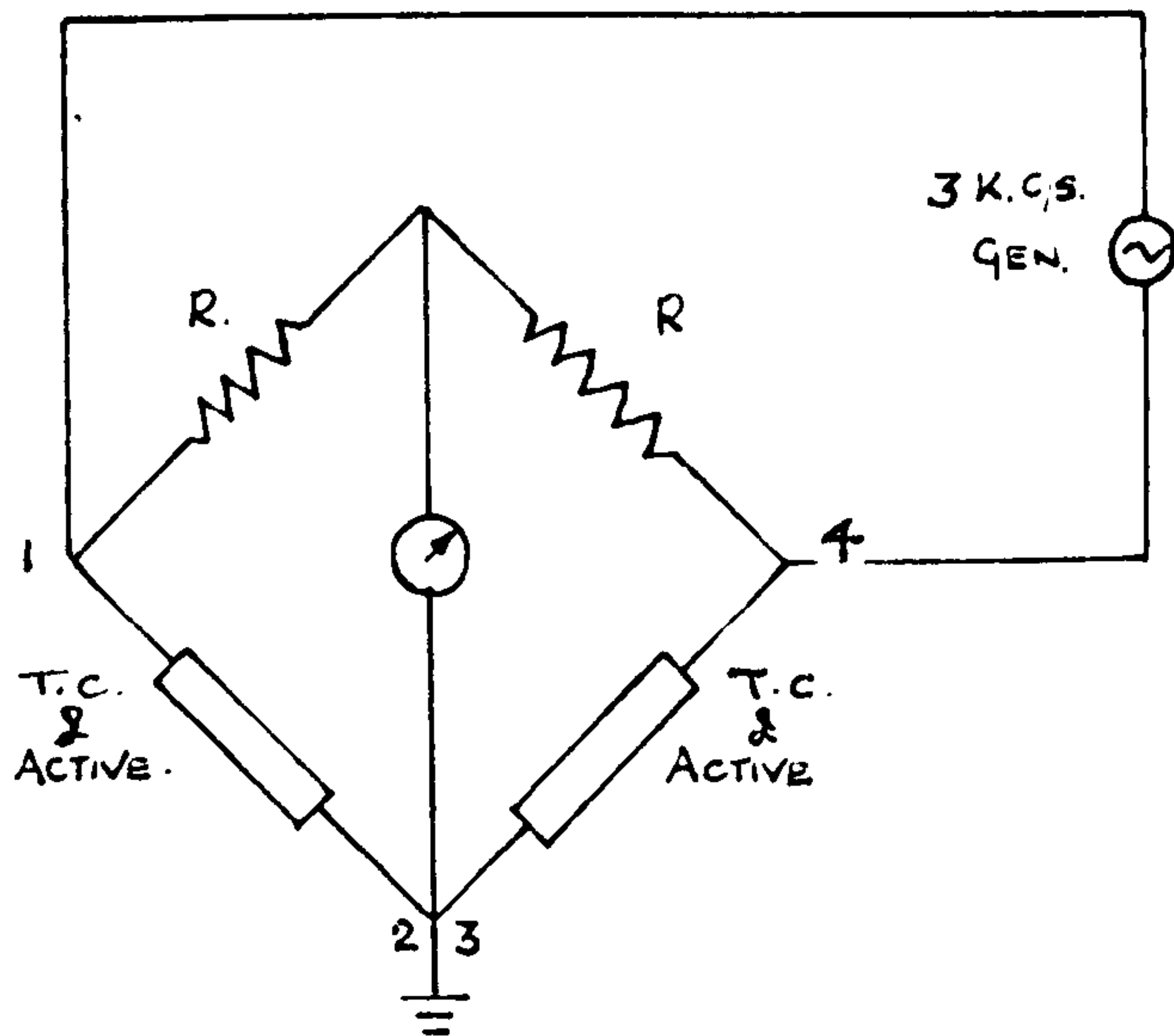
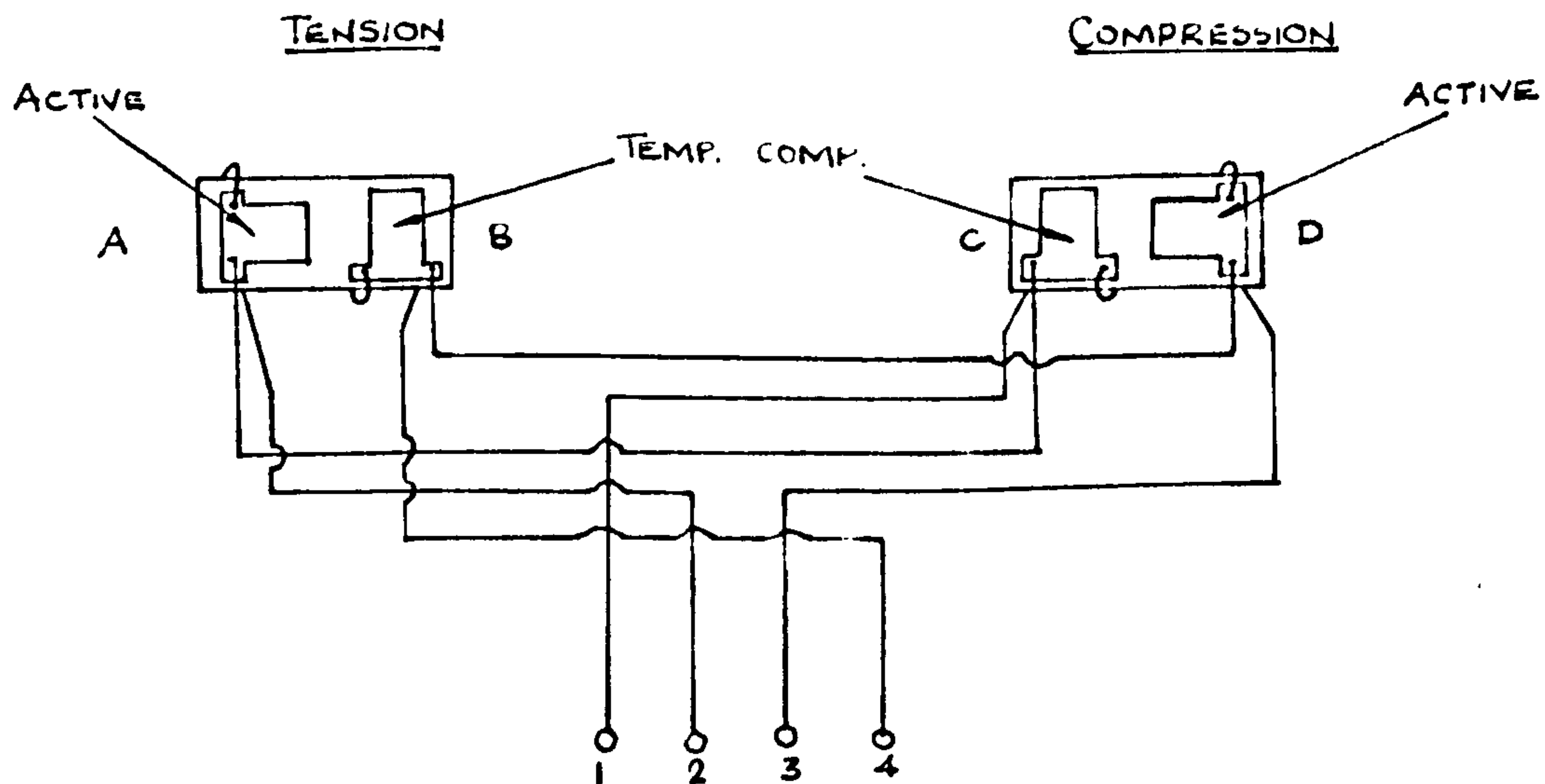


FIG 32 STRAIN GAUGE CONNECTIONS



(c) Measurement of the Strain Gauge Output

With the strain gauges attached to the feeler strips in such a way as to measure horizontal strain with compensation for temperature and bending effects and protected against moisture, it was then necessary to connect the strain gauges into a suitable circuit in order to measure the strain gauge output.

A circuit was required that would supply no signal when the gauges were unstrained but would supply a signal proportional to the change in gauge resistance when the strain was applied - such a circuit is the well known Wheatstone Bridge. The strain gauge leads were therefore connected in this way; the theoretical circuit used for this particular application is shown in Fig. 31, and the strain gauge connections are shown in Fig. 32.

The apparatus used to feed the Wheatstone Bridge circuit and to amplify and measure the voltage output from the strain gauges was a Bruel and Kjaer instrument (type 1516) which is specifically designed for use with strain gauges.

This instrument uses a 3 Kc/s oscillator for feeding a Wheatstone Bridge, and the signal output voltage from the bridge is amplified in a 3 stage amplifier, the output of which, together with a sample of the oscillator voltage, is fed to a phase sensitive demodulator circuit. A centre zero indicating meter, connected to the output of the demodulator, indicates the magnitude and sense of the bridge unbalance and hence the strain. The advantages of the circuitry used are, high sensitivity to signal, insensitivity



to hum (the demodulator responds only to the oscillator frequency), and phase or sense discrimination permitting tension and compression to be displayed respectively as positive and negative deflections on the meter.

A Wheatstone Bridge will always be in slight unbalance due to small divergencies between the components, the capacitance influence of the wiring, etc. With the extreme sensitivity necessary for this kind of measurement any unbalance in the bridge has to be completely eliminated before taking measurements. In order to achieve perfect balance, both an R and C balance are incorporated in the instrument, consisting of a set of resistors and capacitors across the bridge supply. Output supply points are fitted so that auxilliary equipment such as a pen recorder or an oscilloscope can be used.

Initial trials were made with the strain gauge force measuring device, to determine the most suitable means for obtaining sensitive and accurate dynamic results.

A pen recorder was found to give high sensitivity but would not respond to the frequency changes from the force device output signal. It is known that pen recorders are suitable for detecting frequency change below 100 c/s, but the power required to drag the pen against the friction of the paper is too great to follow frequency response above 100 c/s.

Although a pen recorder could have been used to determine the average force, it was decided that the most suitable equipment for this purpose was the Cathode Ray Oscilloscope used in conjunction with a moving film camera,

VIBRATION

FIG. 33

A. 60 R.P.M.

B. 20 R.P.M.

C. 100 R.P.M.

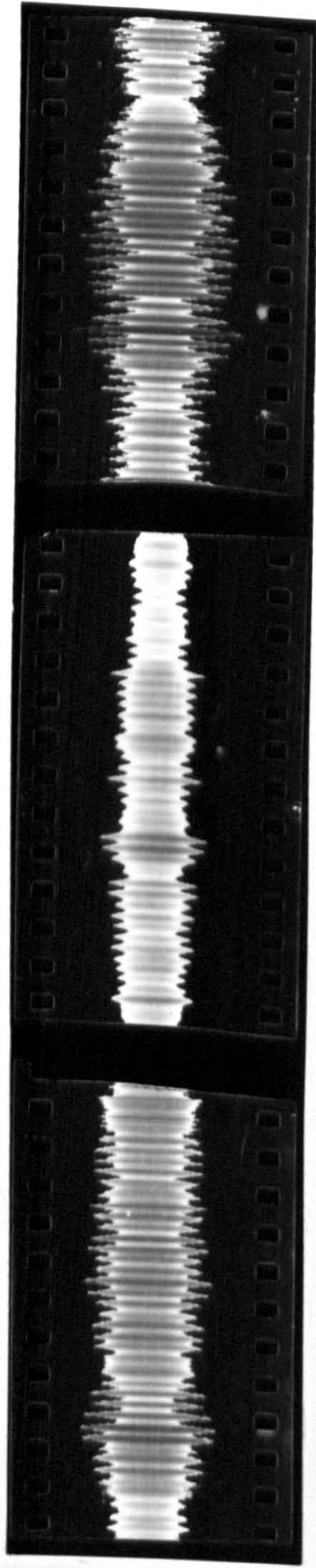
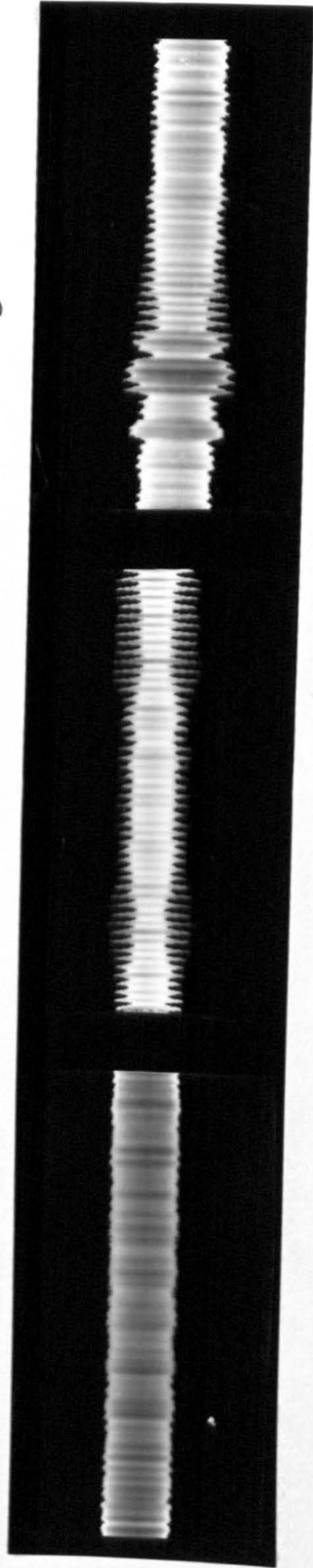


FIG 34

A

B

C



as this frictionless apparatus could be used to detect not only accurate average force value, but also to reveal detailed information of force fluctuations.

A Cosser double-beam oscillograph model 1035 mk III comprising a cathode ray tube with a post deflection anode, and an amplifier of variable gain, was used in conjunction with an oscilloscope camera model 1428 mk IIA, capable of recording continuous wave forms. Film traces of the strain gauge output signal were obtained for many of the experiments made with the force measuring apparatus.

### 3. Preliminary Experiments with the Strain Gauge Force

#### Device

Some experiments were made to determine the reliability of the measured output from the strain gauge force measuring device, and to determine if the output was related directly to the needle forces.

The  $45^{\circ}$  linear cam system was fixed to the cam section, and the strain gauge force measuring device was located and clamped into its correct position on the machine top plate.

A film trace was taken, with an oscilloscope camera speed of 10 ins/sec, when the knitting machine was producing fabric at a speed of 60 r.p.m. (A, Fig. 33). Examination of the film trace showed a clearly defined frequency of 19 cycles/inch (i.e. 190 c/sec.).

The precise nature of this frequency was not immediately obvious; it could not be related to a needle force as there were 468 needles in the cylinder and at this machine speed of 60 r.p.m., an individual needle would have produced a frequency of 468 c/sec or 46.8 needles/inch. It was

considered that this frequency might be caused either by the indirect motion of some other part of the machine, such as the crown wheel gear teeth or by the natural vibration of the force measuring device. In order to obtain more information, further film traces were recorded at a camera film speed of 10 ins/sec with the machine running at 20 and 100 r.p.m. These traces, B and C, Fig. 33, provided the same frequency of 190 c/s and therefore it seemed likely that this frequency was a result of natural vibration, since a change in frequency would have been observed were it related to a function of the moving machine parts.

(a) Dampening the natural vibrations

It was considered that this vibration force and frequency were probably obscuring the desired traces of needle forces. The vibration would therefore have to be eliminated or considerably reduced if accurate, detailed information of the needle forces were to be observed.

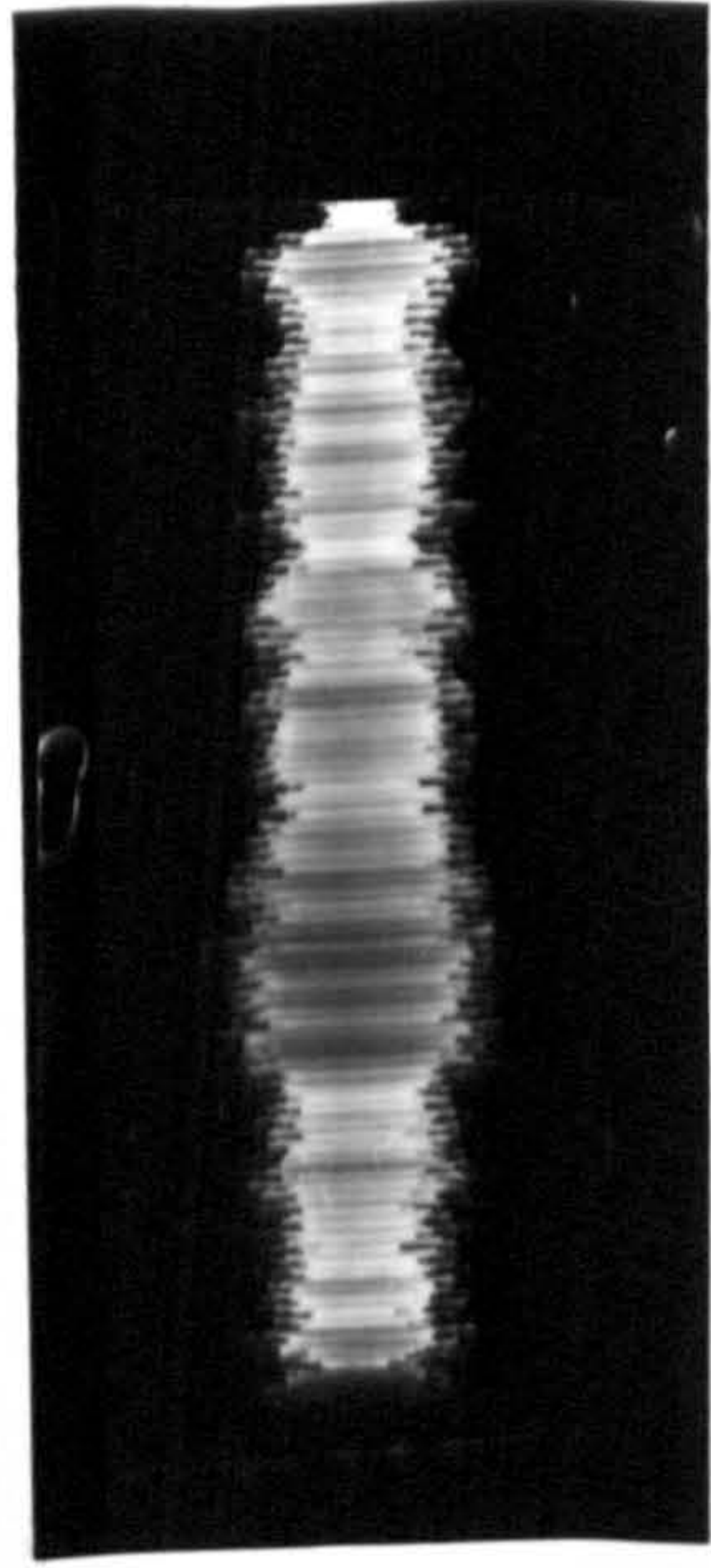
It seemed likely that the needle butts produced the natural vibration of the force system, although it might also have been produced by the vibrations of the machine.

To determine more precisely the causes of this strain gauge vibration, film traces were taken with a camera speed of 10 ins/sec at machine speeds of 60 and 100 r.p.m. without needles in the cylinder. It was observed from the film traces at 60 r.p.m. (A, Fig. 34), that no vibration force was evident, whereas the film trace at 100 r.p.m. (B, Fig. 34) showed clearly the same vibration frequency as that observed with previous traces taken with needles in position. It appeared, therefore, that both needle butts and machine vibrations could produce the natural vibration, although the

NEEDLE FORCE

A. FIG 35

60 R.P.M

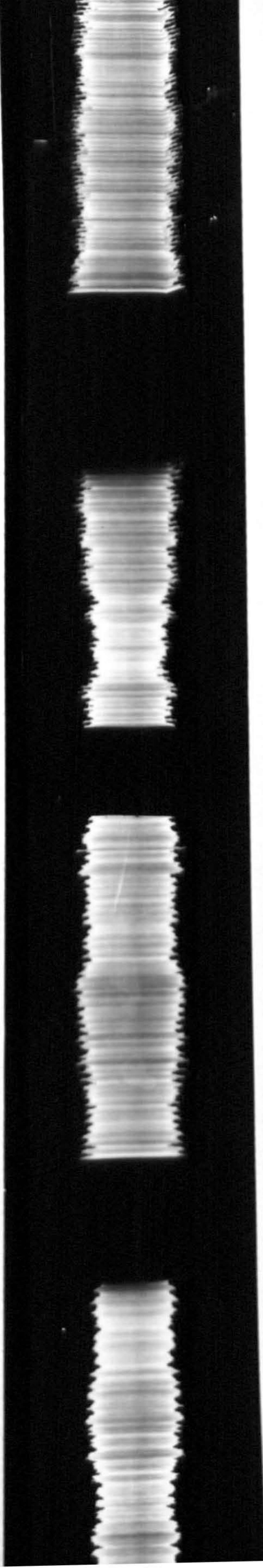


B. FIG 35

30 R.P.M

40 R.P.M

50 R.P.M



machine vibration only affected the force device at higher speeds. Another experiment was conducted to illustrate the natural vibration. A film trace was taken at a camera speed of 10 ins/sec of one needle acting on the cam track, at a machine speed of 60 r.p.m. (C, Fig.34); this film trace clearly shows the vibration force after the needle had left the cam track.

A number of trials were made in an attempt to dampen or reduce the magnitude and frequency of this natural vibration. These trials, which involved modifying the force device (Fig. 29), included : substituting the 0.010" vertical feeler strips (B) with 0.060" feeler strips; increasing the 0.010" top feeler strip (D) to a 0.100"; clamping large weights to jig fixture (C); attaching weights to each side of the cam section with the weights immersed in oil; fixing cork and rubber washers under the strain gauge feeler strips etc. The film trace recordings of the above trials showed no substantial changes to the natural vibrations, and the force device was altered to its original state.

Further trials were made with thin sheet rubber mounted between the base of the cam section and the machine top plate (i.e. position G, Fig. 30). A film trace was taken at a camera speed of 10 ins/sec and a machine speed of 60 r.p.m. with yarn supplied to the needles in the 45° linear cam system. The film trace (A, Fig. 35) shows a significant change in the frequency of the trace, and a close examination shows that the new frequency is approximately 45 c/in., i.e. 450 c/sec., which agrees closely with the number of needles passing through the cam system in one second.

These results suggested that the introduction of the thin sheet rubber had been sufficient to completely eliminate the vibrational effects, and the traces obtained under these conditions were recording accurately the strains on the cam, due to the impact and movement of the needles upon it.

To confirm this, film traces (B, Fig. 35) were taken at a camera speed of 10 in/sec for different machine speeds, i.e. 20, 30, 40 and 50 r.p.m. Examination of the frequencies from the traces showed a distinct change with speed, and these were found to be directly related to the speed of the needles.

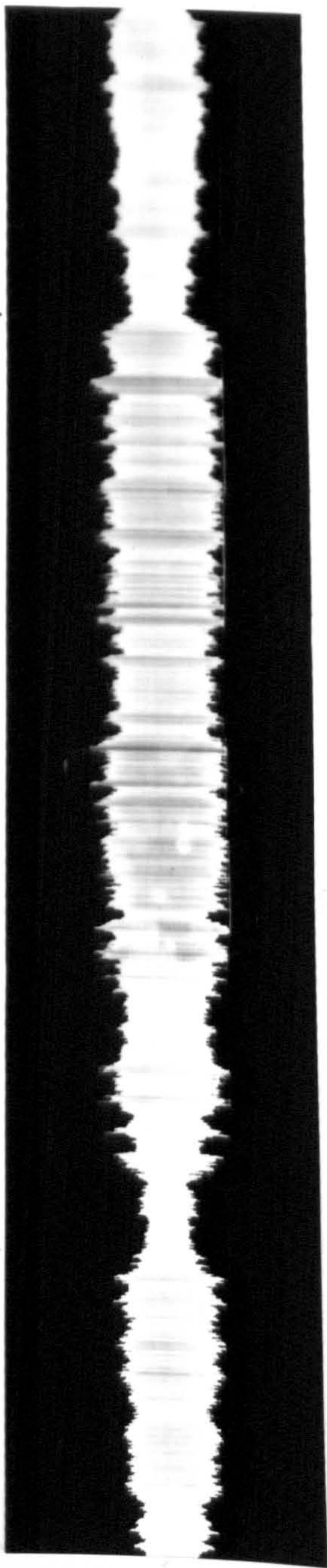
Since the application of thin rubber had clearly dampened the natural vibration, it was necessary to determine the effect of the rubber on the overall sensitivity when the force system was subjected to a change in applied force. Experiments showed that although the overall sensitivity had been reduced approximately 30% by the rubber, the sensitivity was still very high - approximately four times the sensitivity obtained from the torque device.

It was evident that the strain gauge force measuring device could now be used to determine accurately and sensitively the forces of needles in the cam system. By obtaining film traces of the strain gauge output signal, not only could an accurate average needle force be measured but also individual needle forces and variations in forces could be obtained. It will be shown that these latter features provided entirely new and more detailed information on many aspects of the forces in knitting, which could not

ONE REVOLUTION OF THE NEEDLES

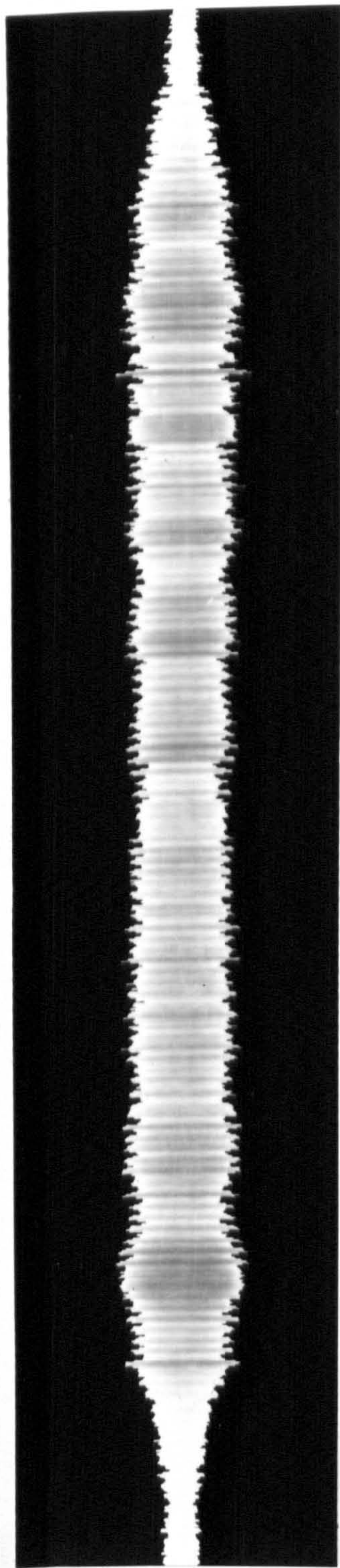
PREPARED

SECTION



B. FIG 3G

PREPARED SECTION





be obtained from a machine torque measuring device, or any other system which measured the average force effects.

(b) Selecting a certain section of needles for filming

Film traces of needles passing through a  $45^{\circ}$  linear cam system showed that a variation in force amplitude occurred over a complete revolution of the needles - probably due to a variation in stiffness from one needle to another. A section of 110 needles and tricks were straightened and polished, to ensure that the needles would slide freely in the tricks, and a film trace was taken for a complete revolution of the machine without yarn supplied -(see A, Fig. 36); (film speed 5 ins/sec, machine speed 60 r.p.m.). It is evident from this trace that a considerable improvement had been achieved by polishing the needles and tricks, as not only had the force amplitude of the specially prepared section decreased, but also the variation between needles had been substantially reduced. It was also far easier to obtain an average force value from the specially prepared section of this trace. It was realised that although an improvement could be made to the remaining needles and tricks, it was unlikely that a perfectly level amplitude would be obtained throughout the complete cycle, particularly as a number of the other needle tricks had been slightly damaged in previous experiments. To ensure accurate and consistent force results the prepared section of needles and tricks was therefore used for the filming of most of the knitting experiments. A film trace (camera speed 25 ins/sec) of the prepared section when needles were knitting in the  $45^{\circ}$  linear cam is shown in B, Fig. 36.

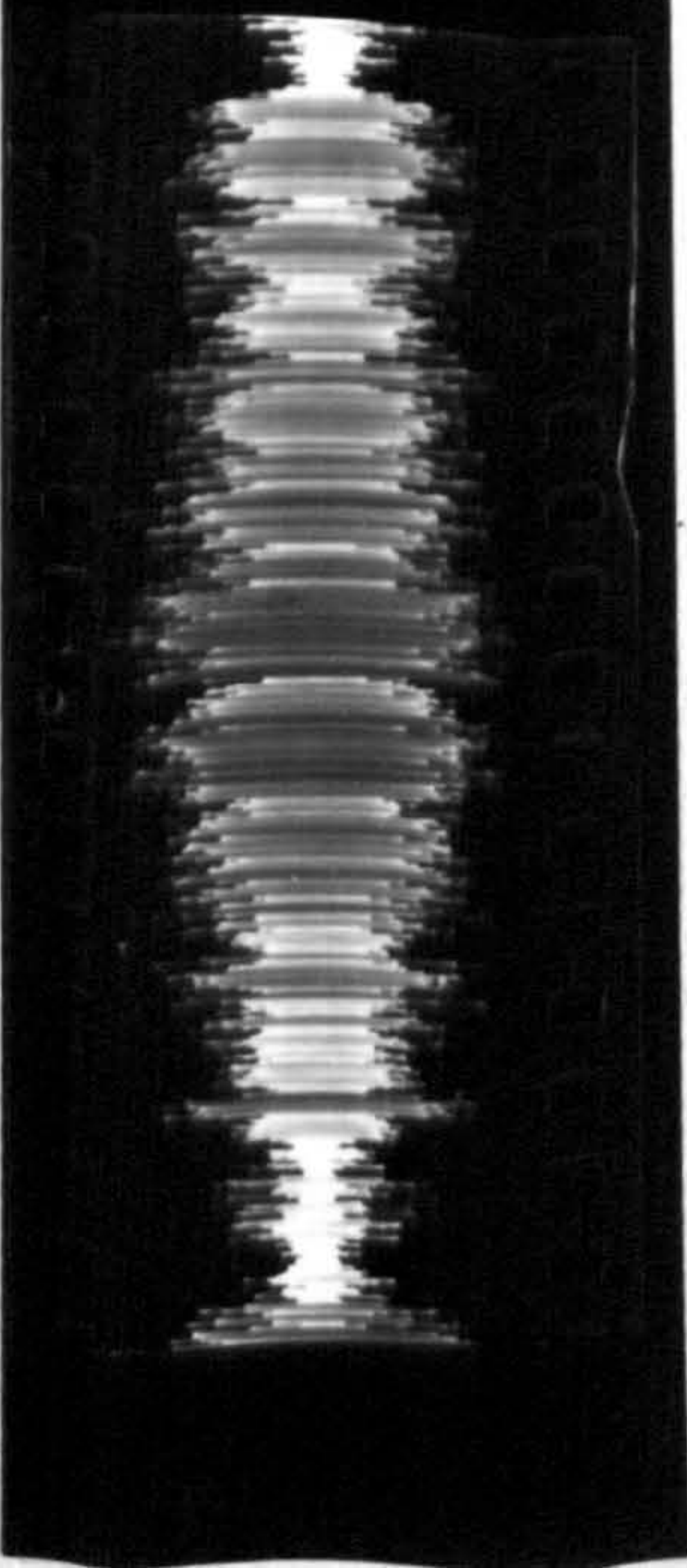
It was considered advantageous to remove a number of needles on each side of the prepared section as shown in B, Fig. 36 since the following factors would then be made possible.

- (i) It enabled the strain gauge device to be calibrated after each series of experiments, without having to remove needles or unclamp and remove the strain gauge apparatus from the cylinder.
- (ii) The missing needles provided an accurate means of identifying the start and end of the prepared section on the film traces. To avoid wasting film, the traces were filmed from just prior to the polished needles entering the cam system to just after their leaving it.
- (iii) The machine was always started and stopped without any needles being in the cam section, thus avoiding the initial high forces that could upset the strain gauge balance - especially with the steeper angled cams. An alternative method would have been to fit in the rigid keeper plates before starting and stopping the machine, but for short experiments this would be inconvenient.

Under abnormal knitting conditions, such as high yarn tensions, large yarn diameters, etc., the thin feeler strips tend to buckle and upset the balance; although this limitation would not restrict the scope of the present work, since the torque apparatus was available and used to measure forces under these severe knitting conditions, a possible method of overcoming this problem was to employ a more sensitive strain gauge fitted to thicker feeler strips.

FIG 37

NYLON YARN



WORSTED YARN

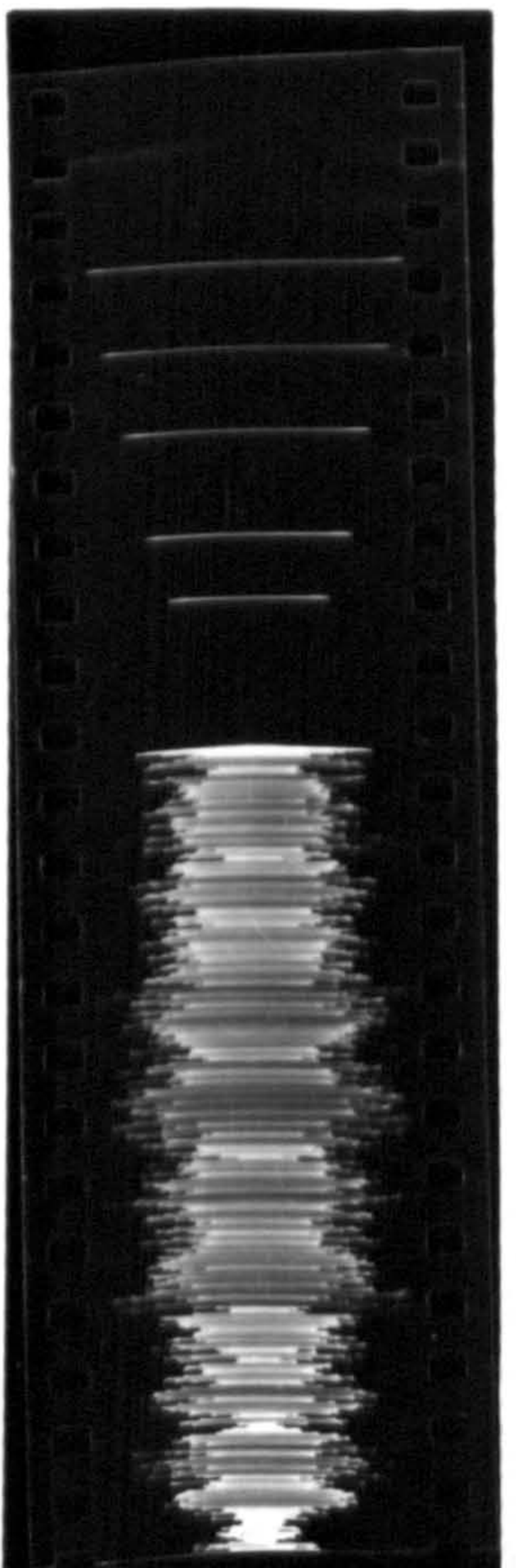
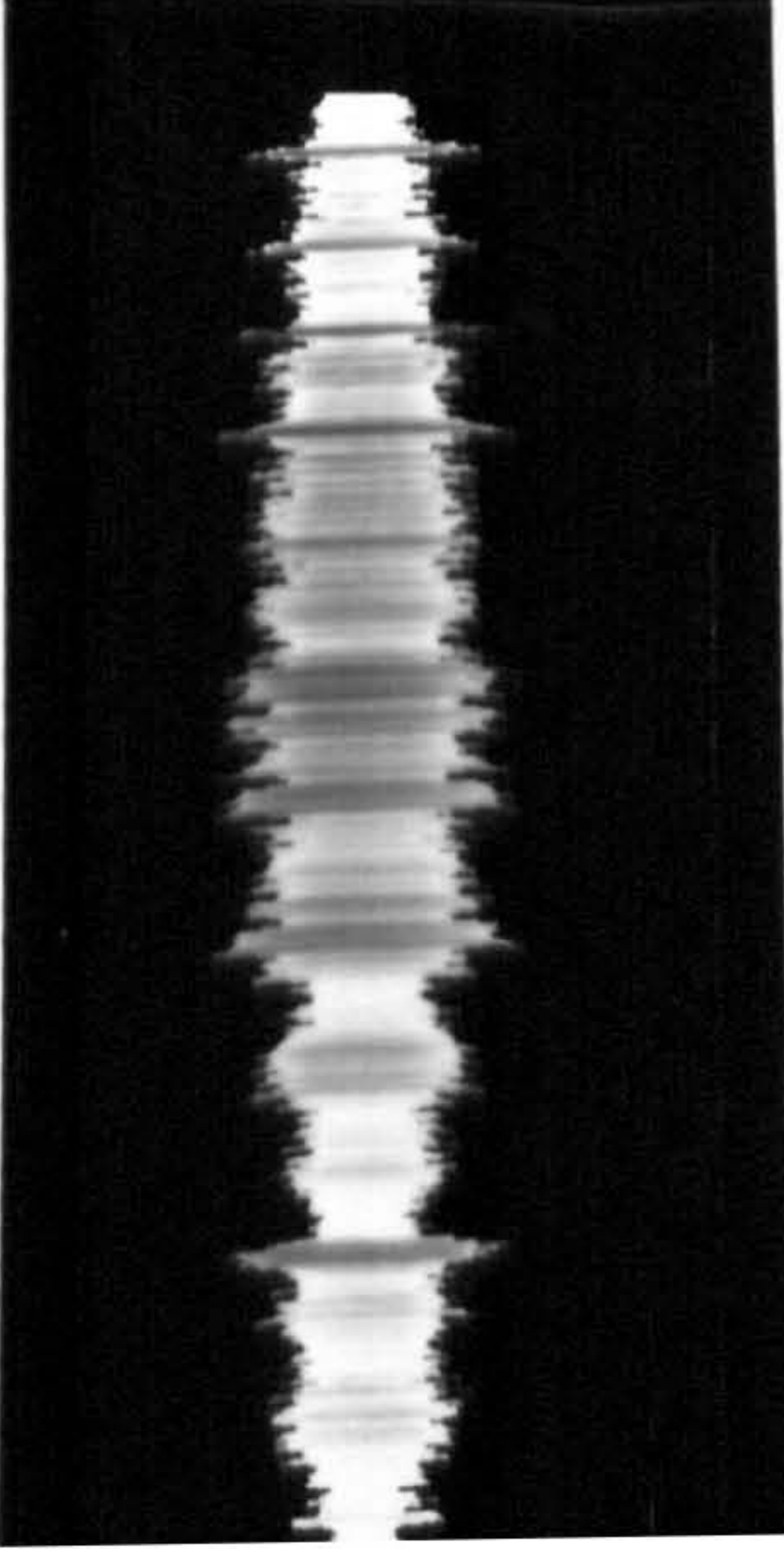
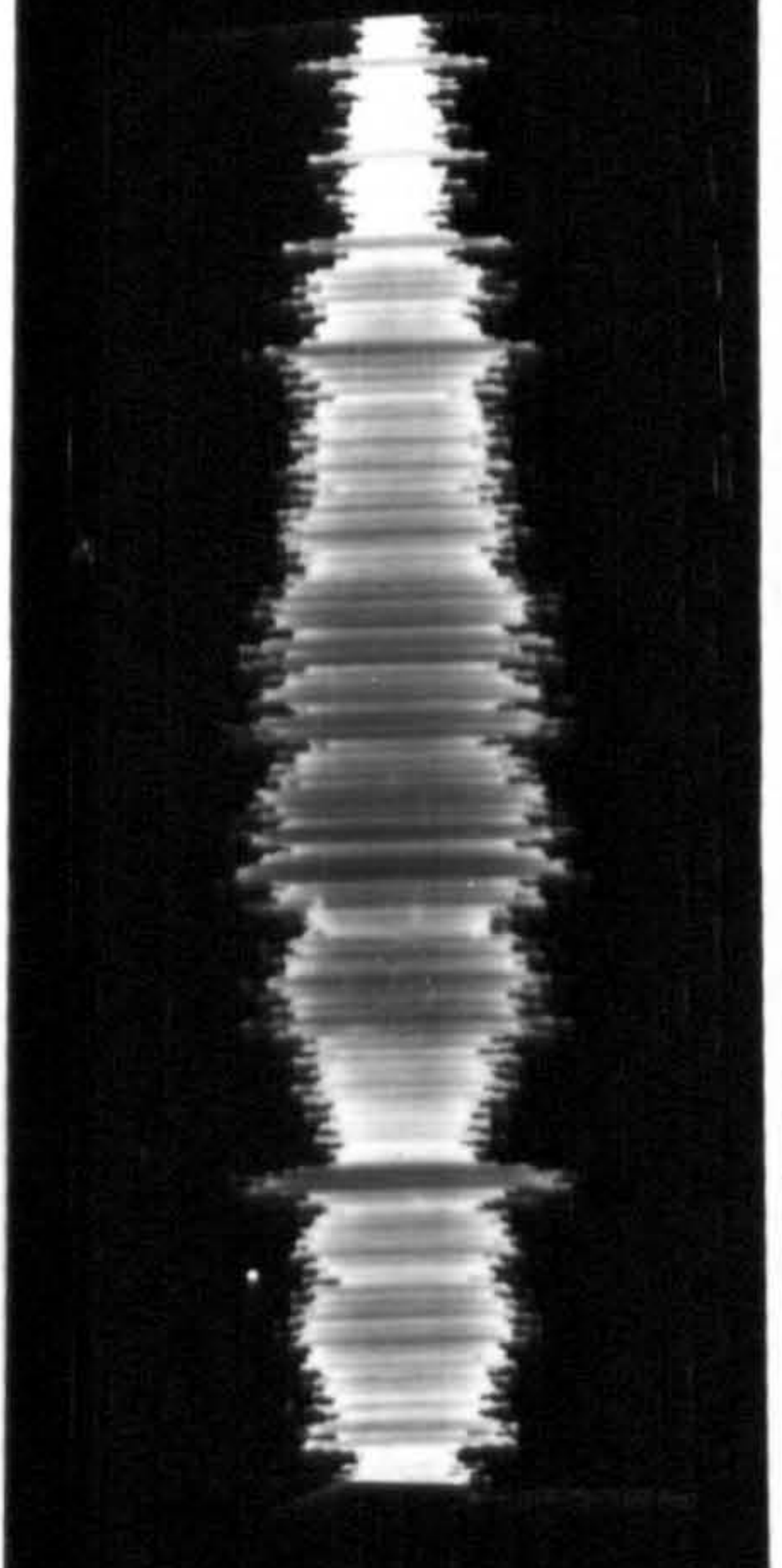


FIG 38

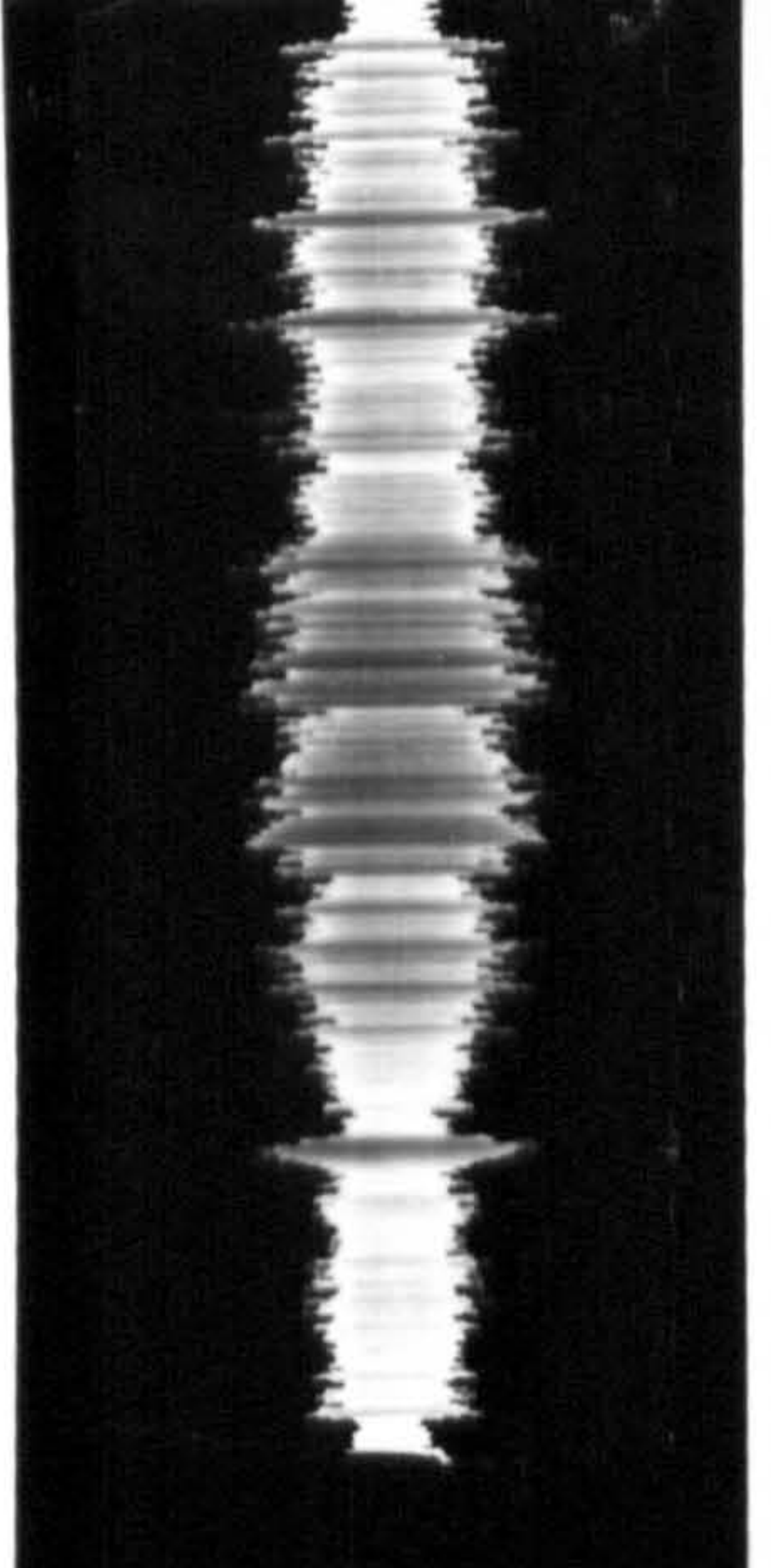
HYSTERESIS BRAKE



POSITIVE FEED



DISC TENSION



(c) Calibration and measurement of forces

Calibration of the strain gauge force apparatus was made by hanging weights on a tray, which was suspended from the apparatus. Weights were added in 4oz. increments and a film was taken of the calibration; a typical calibration and film are shown in Fig. 37.

To obtain the values of forces, film traces were passed through an enlarger and the image was copied accurately on to a roll of recording paper. Average amplitudes could then be drawn easily, and values of force could be obtained from the corresponding calibration.

(d) Effect of different types of yarn on the force characteristics

Using the 45° linear cam system, film traces were taken when knitting at 60 r.p.m. with a 1/28 worsted yarn and a 300 denier continuous filament nylon yarn - the yarn input tension for both yarns was controlled at 5 gms. The traces which were taken at a camera speed of 10 ins/sec and at a higher sensitivity than previous traces, are shown in Fig. 37, and it may be observed that there is virtually no difference in the force amplitude. Measurements of yarn friction, under the same conditions of yarn input tension, gave only slight differences in values of friction, i.e. 0.302 for the 1/28 worsted yarn and 0.290 for the 300 denier nylon yarn.

(e) Effect of different mechanical control of the yarn on the force characteristics

It is known that the hysteresis brake device, although not satisfactory for commercial purposes, gives better and more uniform control of yarn tensions than the disc tension,

and that positive feed gives the most accurate control of the stitch length. To determine whether these systems have any effect on the force characteristics, film traces were taken at a camera speed of 10 ins/sec and a machine speed of 60 r.p.m. The control of the yarn, which was supplied to needles acting in the  $57^{\circ}$  non-linear cam, was achieved with (i) hysteresis brake, (ii) positive feed, (iii) disc tension with compensator, and the film traces are shown in Fig. 38.

It may be seen from these traces that no significant difference in force amplitude or individual peak force occurred and therefore no advantage can be gained by using the hysteresis brake or positive feed rather than the disc tension for the force results.

CHAPTER 5

INVESTIGATIONS INTO THE PERFORMANCE OF THE

VARIOUS SHAPED CAM SYSTEMS

CHAPTER 5

INVESTIGATIONS INTO THE PERFORMANCE OF THE  
VARIOUS SHAPED CAM SYSTEMS

Preamble

It has been shown in previous chapters how the cam systems, knitting machine, torque and force measuring devices have been prepared for an experimental analysis. The purpose of this chapter is to compare the performance of the various cam systems when the needles acting in each cam are subjected to a wide range of knitting conditions.

The performance of each cam (i.e.  $45^{\circ}$  and  $55^{\circ}$  linear,  $52^{\circ}$ ,  $57^{\circ}$  and  $62^{\circ}$  non-linear,  $55^{\circ}$  "straight and curved") will be assessed from the stitch length, force and torque results which were obtained when the knitting conditions such as cam setting, yarn tension, take-down tension, yarn count and speed were varied.

In the final part of this analysis the needle butt performance and the fabric appearance will be examined, at knitting speeds much higher than commercial speeds, to see which cam systems will provide increases in rate of fabric production.

A. Experimental Procedure

(a) Providing for "Standard" Knitting Conditions

So that the different cam systems could be compared under precisely the same knitting conditions, it was important to establish certain "standard" machine and yarn conditions. The standard conditions chosen were those that would normally be used to produce a satisfactory commercial fabric on the 8" diameter machine, viz.

Yarn input tension = 5 gms.

Yarn count =  $\frac{1}{28}$  worsted or 300 den.

Take-down tension = 2 lbs.

Speed = 60 r.p.m.

(b) Choice of yarn

A worsted yarn was used for most of the experimental work rather than a more uniform man-made fibre yarn, as it has a lower tensile strength. It was considered that if the various cam shapes were effective under all knitting conditions, when using this yarn, then there would be no difficulty with most other yarns.

The breaking load and the percentage elongation of the  $\frac{1}{28}$  worsted yarn were measured on the Ulster load tester. An average breaking load of 228.4 gms. and an elongation of 8.7% were obtained over 20 separate tests.

(c) Cam Setting and Stitch Length

Three specific stitch lengths of 0.196", 0.176" and 0.156" were used for these experiments, to give respectively slack, medium and tight fabric conditions.



To obtain the above stitch lengths, each cam system was adjusted vertically and the stitch length was measured from fabric knitted at the "standard" conditions. When a cam system was located at the correct depth to give a required stitch length, a vernier height gauge was used to measure the cam setting; this vernier height gauge measurement was used to re-locate the cam setting for subsequent experiments. Sample fabrics were knitted for each experiment and, after they were allowed to dry relax, accurate measurements of the stitch length were made. This was achieved by unroving several complete course lengths from the 8" diameter fabric and measuring the uncrimped length on a H.A.T.R.A. course length tester. The stitch length values were obtained from

$$l = \frac{\text{Average course length } L \text{ (ins)}}{\text{Nos. of needles in cylinder } n}$$

where  $n = 468$  for the 8" diameter machine.

B. Comparison of Stitch Length Results obtained with the Various Cam Systems

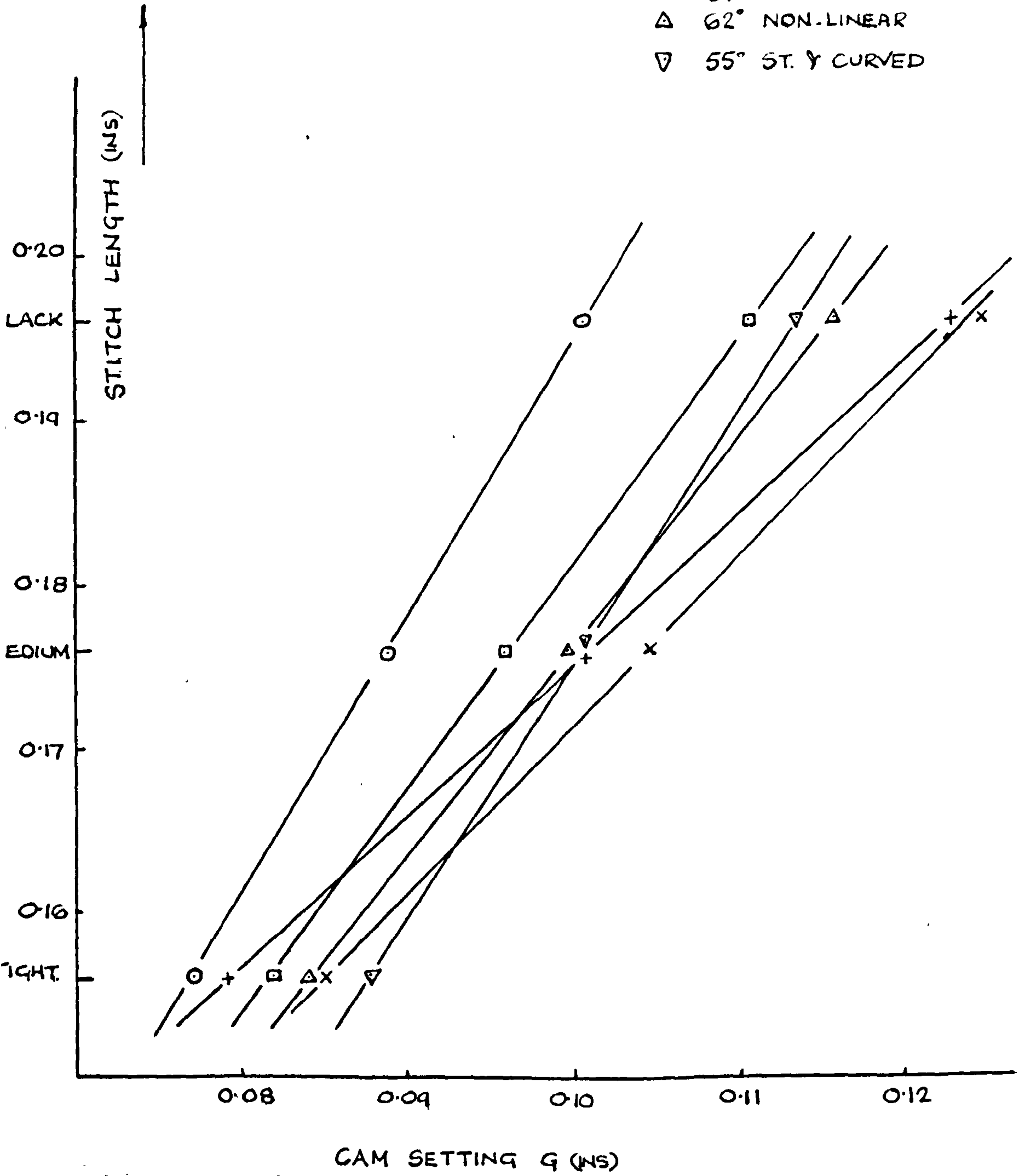
1. Relationship between Cam Setting, Stitch Length and Robbing Back

(a) Effect of Cam Setting on the measured stitch length

Each cam system was set to provide the slack, medium and tight stitch lengths and, for each setting, the maximum needle depth, from the sinker knock-over surface to the needle hook, was measured with a depth gauge; these results were added to the 0.018" diameter of the needle hook to provide a value for the total cam setting. This total cam setting will be subsequently referred to as cam setting (G).

Fig 39.

- X 45° LINEAR
- + 55° LINEAR
- 52° NON-LINEAR
- 57° NON-LINEAR
- △ 62° NON-LINEAR
- ▽ 55° ST. Y CURVED



The cam setting (G) results, together with the corresponding measured stitch length values, are shown in Table 16\*, and plotted in Fig. 39.

It may be observed from Fig. 39 that at any stitch length there is a wide variation in cam setting. For instance at the slack stitch length the cam setting (G) for a 52° non-linear was 0.101", whereas the 45° linear cam setting was 0.125". At the medium and slack stitch lengths the linear cams require a longer cam setting than the non-linear and the 55° "straight and curved" cams; no such pattern is to be observed at the tight setting, and there is also less variation in cam setting.

The relationship of cam setting and stitch length for every cam system is linear; the slopes for non-linear cams, however, are all steeper than those of the linear cams. Therefore, in changing from slack to tight structures, a linear cam requires more vertical adjustment than the non-linear cam system. For instance a 45° linear cam requires 0.040" adjustment between slack and tight structures, whereas a 52° non-linear cam requires only 0.024" adjustment, i.e. 40% less vertical movement is required for this cam system. It is of interest to note that the 55° "straight and curved" cam behaves in a similar manner to the non-linear cam system.

(b) Calculation of theoretical stitch length ( $l_t$ ) and the percentage robbing back values

In order to explain the differences between non-linear and linear cam systems, described above, it was necessary to calculate the percentage robbing back values for each cam at each setting. It has been mentioned in

\* See Appendix

Chapter 1 that Knapton, in his work with linear cam systems, found that at all knitting conditions the actual stitch length ( $l_a$ ), measured from the fabric, was always less than the theoretical stitch length ( $l_t$ ) predicted from the geometry of the knitting elements. Nearly all this difference in stitch length was a result of the descending needle robbing yarn from the previously knitted loop, (i.e. robbing back).

The percentage robbing back was calculated from

$$\% \text{ R.B.} = \frac{l_t - l_a}{l_t} \times 100.$$

Knapton calculated the theoretical stitch length ( $l_t$ ) of a  $45^\circ$  linear cam system from the geometrical arrangement of the longitudinal axis of the yarn (see Fig.40), using the relationship

$$l_t = 2\sqrt{d^2 + \frac{a^2}{4}}$$

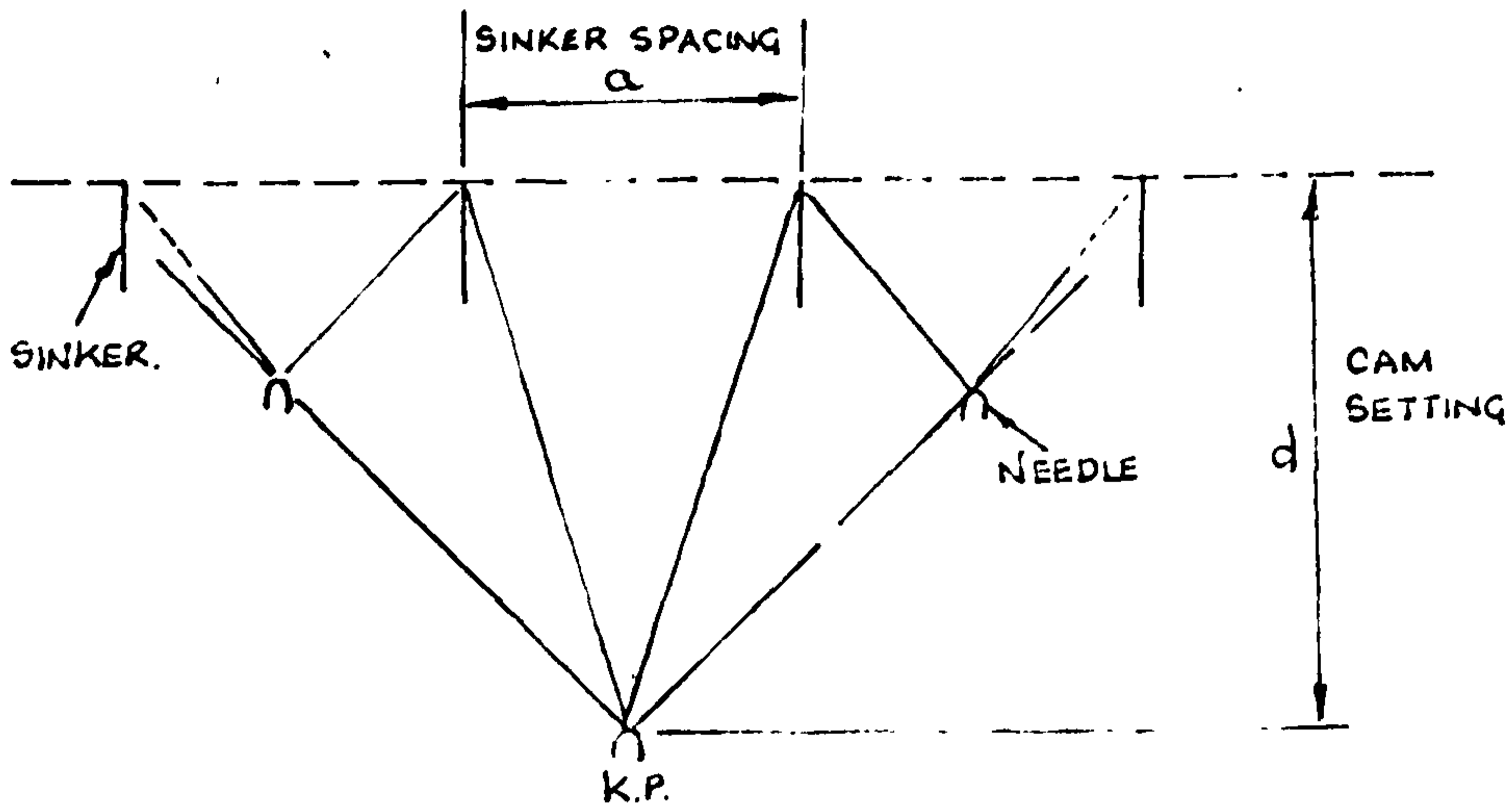
This expression, however, assumes that the yarn diameter and knitting element dimensions are negligible compared with the sinker and needle spacing.

For the purpose of this present work it was considered more accurate to take into account the actual dimensions of the yarn diameter and knitting elements for the calculation of the theoretical stitch length. These dimensions were measured in the following manner.

Using a standard microscope with 100 times magnification, the yarn diameter was measured under low and high tensions; at 5 gms. tension the yarn diameter was 0.013" and at tensions above 20 gms. it was 0.010".

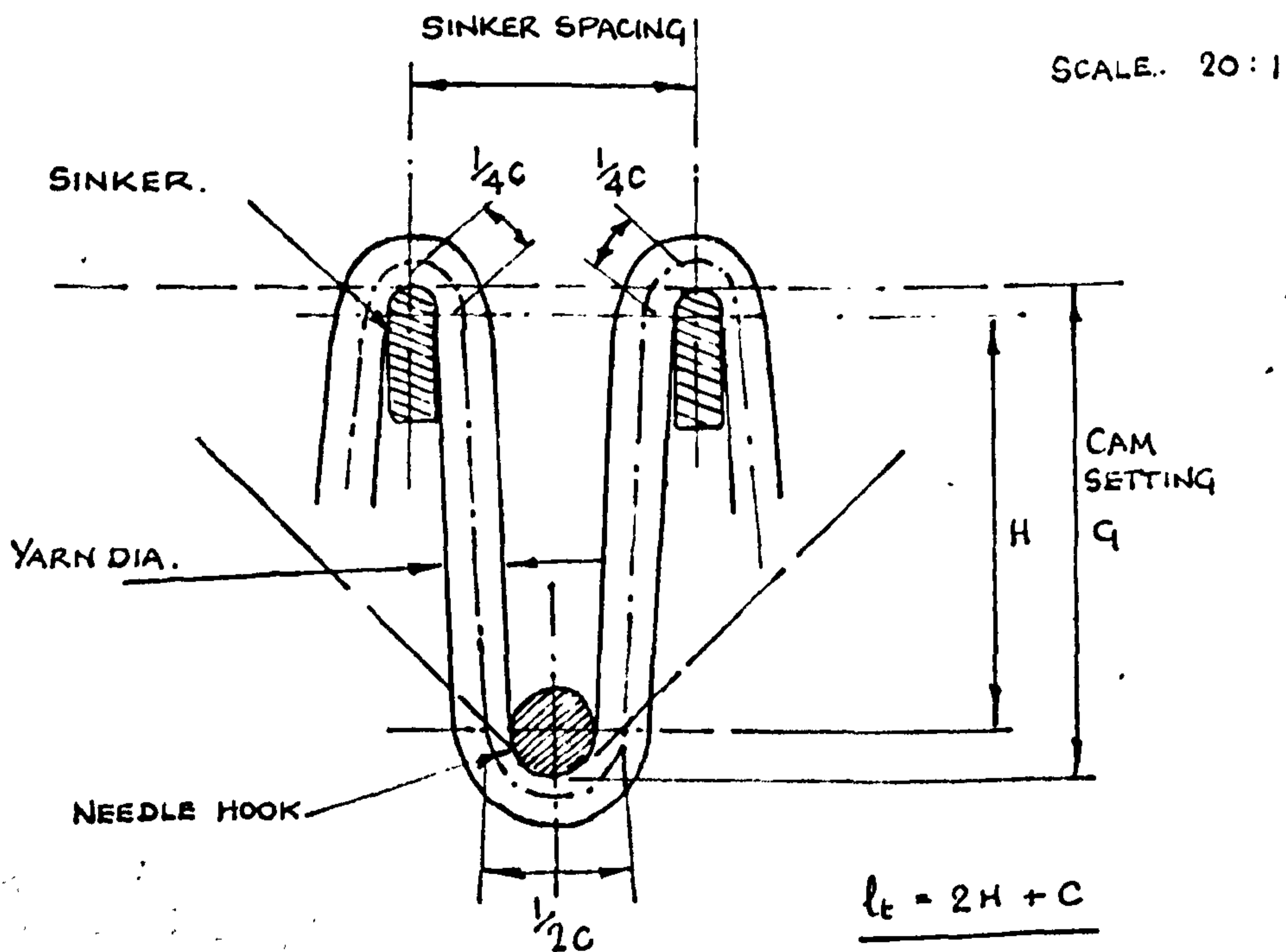
DETERMINATION OF THEORETICAL STITCH LENGTH ( $l_t$ )

FIG 40



$$l_t = 2\sqrt{d^2 + \frac{a^2}{4}}$$

FIG 41



$$l_t = 2H + C$$

The dimensions of the knitting elements, measured with a micrometer, were: sinker width = 0.008", needle hook diameter = 0.018", and the sinker spacing, calculated from the circumference of the machine, was 0.054". These values, which were drawn to scale (Fig. 41), illustrated that the path of the yarn provided an angle of wrap around the knitting elements of approximately  $180^{\circ}$ ; with Knapton's method the angle of wrap was  $150^{\circ}$  for the same cam setting.

The values of  $l_t$  from this more accurate method comprised the vertical length of yarn between sinker and needle centre line (i.e.  $2H$ , see Fig. 41), plus the yarn passing around the radii of the needle and sinker elements (i.e.  $C$ , Fig. 41).

$$\text{Thus, } l_t = 2H + C \quad \dots (V.1)$$

The value of  $H$  was dependent upon the cam setting  $G$ , and was obtained by subtracting the sum of the sinker and needle radii from the value of  $G$ ,

$$\text{i.e. } H = G - 0.013" \quad \dots (V.2)$$

In obtaining a value for  $C$ , it was not obvious whether this length of yarn was determined from the actual radii of the knitting elements, or whether it was determined from somewhere near the radius of the neutral axis of the yarn. It was important to determine this length of yarn accurately, as calculations showed that a wrong choice of  $C$  would provide a significant error in the  $l_t$  values. For instance, if the knitting element radii were assumed to be correct, dimension  $C$  would equal 0.040", and if the neutral axis of yarn were correct  $C$  would equal 0.072".

An experiment was conducted to establish the precise value of C. Ten needles were set to give a cam setting G of 0.203" and yarn was passed around ten sinkers and needle hooks - the yarn was therefore under high tension as during loop formation.

The yarn was marked and cut at each of the end sinkers and then removed from the knitting elements and measured accurately under 10 gms. tension. This method provided consistent lengths of yarn of 4.30" when the experiment was repeated several times. This length of yarn represented the total length of yarn in ten equivalent  $l_t$  values and therefore the length of yarn in one  $l_t$  value was 0.43". A value of  $2H = 0.38"$  was obtained from formula (V.2) where  $G = 0.203"$ ; therefore, from formula (V.1) the value of  $C = 0.050"$ .

It was noted that this value of C was inbetween the values calculated from the knitting element radii and the neutral axis of the yarn. The theoretical stitch length at any cam setting may now be calculated from

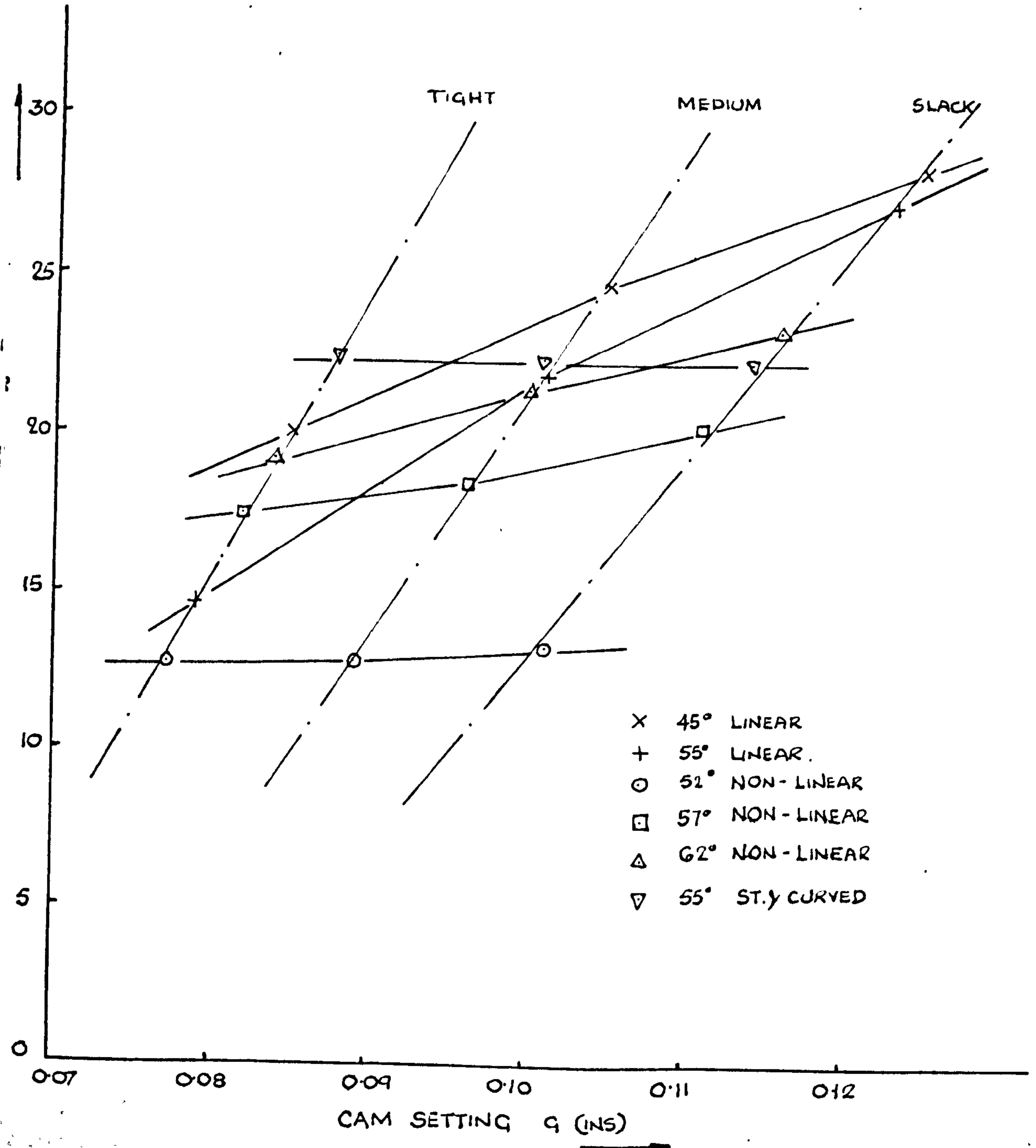
$$l_t = 2H + 0.050" \quad \dots (V.3)$$

where H is dependent upon the cam setting G.

It was interesting to compare the values of  $l_t$  obtained for the  $45^\circ$  linear cam system at slack, medium and tight settings, when using this system, with those obtained using Knapton's method, (see below).

<u>Cam Setting</u>	<u><math>l_t</math> (ins) This work</u>	<u><math>l_t</math> (ins) Knapton</u>
Slack	0.274	0.257
Medium	0.234	0.217
Tight	0.194	0.178

Fig 42





(c) Relationship between cam setting and robbing back

Calculations of  $l_t$  and the percentage robbing back for each cam system at each cam setting, are shown in Table 16, and cam setting G and robbing back were plotted, (see Fig. 42).

It is observed from this relationship that at the slack setting all the non-linear cams have a lower percentage robbing back than the linear cams; this agrees with the relationship already established between cam setting and measured stitch length (see Fig. 39), where it was observed that a longer cam setting was required for linear cams to produce a slack stitch. It was clear that this longer cam setting was necessary to compensate for the larger percentage of robbing back that occurred.

The robbing back relationship also showed that as the cam setting was increased from tight to slack, the percentage robbing back increased; although the increase was only slight with the non-linear and the straight and curved cams, it was quite substantial with the linear cams. This also explained why, when examining the measured stitch length/cam setting relationship, the linear cams required a larger range of cam setting movement to give a specific change of stitch length. It was clear that since more robbing back occurred with the linear cams as the stitch cam setting was increased, then more vertical movement was required to compensate for the robbing back. With non-linear cams only sufficient cam movement to alter the stitch length was required, as robbing back only changes by a very small amount.

It was noted that at a tight cam setting the 55° linear cam gave less robbing back than the 57° and 62° non-linear cams; this presumably occurred because at this short stitch length, the needles in the 55° linear cam found it easier to draw yarn from the package than to rob yarn back.

Since non-linear cams required less depth below the sinker level to provide a given range of stitch lengths, it was realised that the circumferential lengths of non-linear cams could be reduced, which would compensate slightly for the extra lengths of such cams. However, a comparison of circumferential lengths, shown later, gave only small decreases in length.

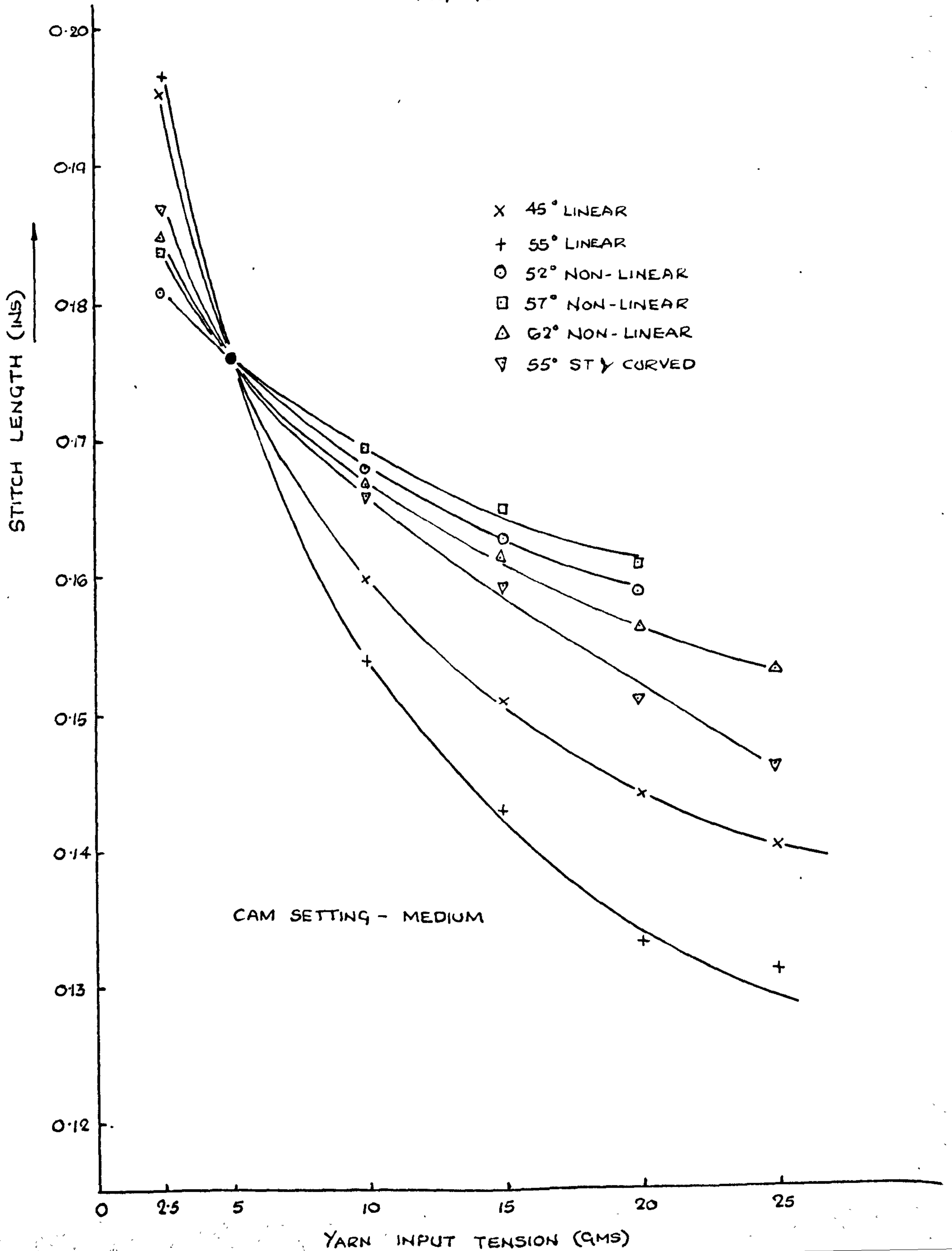
## 2. Relationships Between Yarn Tension, Stitch Length and Robbing Back

### (a) Effect of yarn tension on stitch length

Each cam system was set in turn, under standard conditions of knitting, to provide the medium, slack and tight stitch lengths. For each cam system the yarn input tension was adjusted, in small increments between 2.5 gms and 30 gms, and sample fabrics from each test were unraveled and the stitch length measured. The results for the medium setting are shown in Table 17\*, and stitch length against yarn input tension is plotted in Fig. 43. The results for the slack and tight settings were found to give similar relationships of stitch length to those obtained at the medium cam setting, and are therefore not shown.

\* See Appendix.

FIG 43



An examination of Fig. 43 shows a significant difference between the various cam systems. The linear cam systems give large decreases in stitch length with increases in tension, whereas the non-linear cam systems show relatively small decreases in stitch length with increases in tension. The compromise cam system (55° straight and curved) gives a stitch length/tension relationship curve which lies inbetween the linear and non-linear cam systems.

With all cam systems the greatest change in stitch length, for a given change in tension, occurs between 2.5 gms. and 5 gms. and this change is greatest with the linear cam systems. It will be noted that the non-linear cam curves do not extend beyond 20 gms. input tension ; this is because holes appeared in the fabric when knitting at tensions above 20 gms.

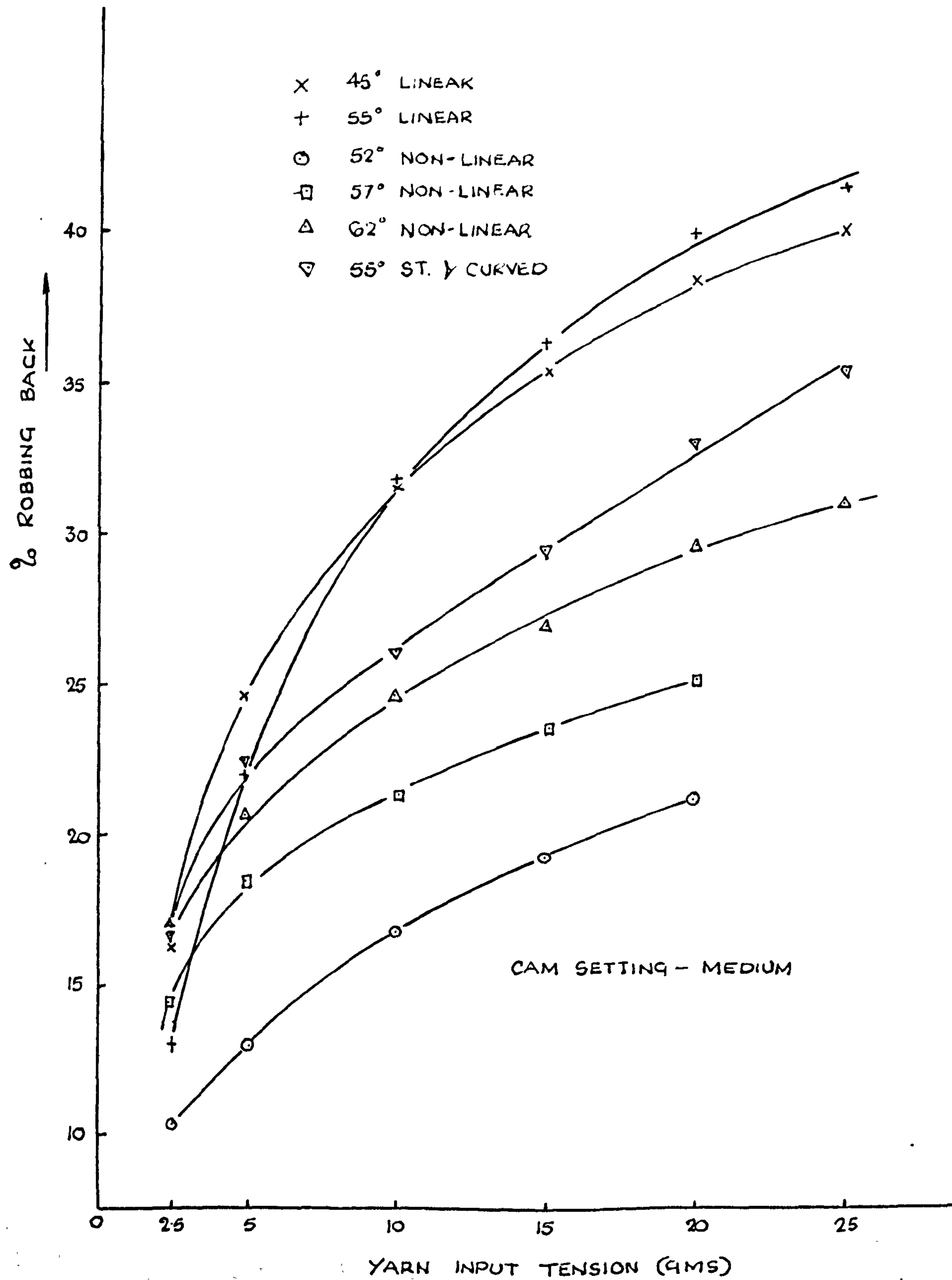
These results show that an approximate 20% change of stitch length is obtained from an increase of 5 - 20 gms. input tension with linear cam systems, and only approximately 10% change with non-linear cams.

(b) Effect of yarn tension on percentage robbing back

To help explain the above-mentioned differences between the linear and the non-linear cam systems, the percentage robbing back values were calculated for each of the measured stitch length values; these are shown in Table 17, and the yarn tension/robbing back is plotted in Fig. 44.

Examinations of Fig. 44 show that with an increase in tension the percentage robbing back increases rapidly with the linear cam system , particularly at the lower tension range. The non-linear cam systems also give an increase in

FIG 44



robbing back with increase in tension, although with these cams the percentage increase in robbing back is far less than with the linear cam systems.

Another feature of interest is that at yarn tensions of 5 gms and above, non-linear cams give a lower percentage of robbing back than the linear cams.

(c) Positive feed as a stitch length controller

Before a more detailed examination of the differences between non-linear and linear cams is given, a practical issue resulting from the above observations will be considered.

Since the non-linear cams have no facilities for separate stitch cam adjustment, it was considered that positive feed might be used as an alternative method of obtaining a range of stitch lengths.

To determine whether this was possible, a trip-tape positive feed system was used to supply different yarn course lengths to the needles in the 57° non-linear cam system, when this cam was fixed at the medium setting. Experiments showed that it was possible to knit a 12% range of stitch length without either causing yarn overfeed or increasing the knitting tension so high as to cause yarn breakage.

The stitch length and corresponding tension values are shown below in Table 3.

TABLE 3

57° Non-Linear Cam System	
Pos. Feed Stitch Length (ins)	Resulting tension (gms)
0.180	2
0.174	5
0.166	10
0.159	20

These results may be compared with those given in Chapter 2, where it was shown that a 26% range of stitch lengths could be obtained when using positive feed with a fixed  $45^{\circ}$  linear cam system.

Since the stitch cam adjustment on commercial machines normally permits at least a 30% range of stitch lengths, it may be concluded that positive feed is an inadequate stitch length controller with fixed non-linear cams. However, it was suggested, in Chapter 2, that if positive feed were unsuitable as a stitch length controller, then certain alternative methods might be used. For example, a modification to provide a vertical adjustment of either the cylinder or the complete cam system or, on dial and cylinder machines, an adjustment of the height of the dial - a provision already available on these machines.

It is clear from the above experiments that if non-adjustable non-linear cams are to be used on commercial machines, then one of these alternative methods must be used to obtain a suitable range of stitch lengths.

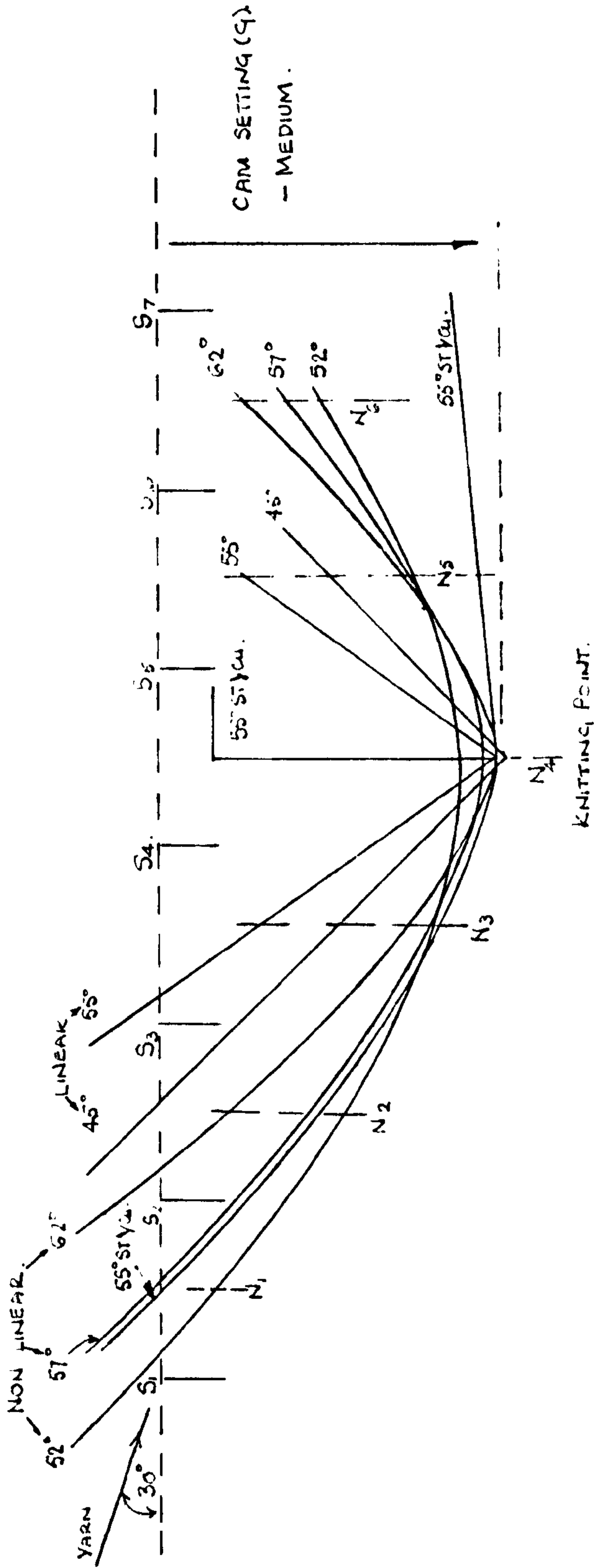
### 3. Calculations of Maximum Tensions within the Loop-Forming Portions of the Various Cam Systems, and their Relation to Robbing Back and Stitch Length Values

#### (a) Examination of the loop-forming portions

It was considered important to make a more detailed examination of the loop-forming portions (i.e. the knitting point areas) of the cam systems, since it seemed likely that this portion determined, to a large extent, the knitting performance of a cam system.

Fig 4-5

LOOP FORMING PORTIONS OF THE CAM SYSTEMS



SCALE, 18:1

N - NEEDLE  
S - SINKER.



Examinations of an enlarged scale diagram of the six loop-forming shapes (Fig. 45) illustrates the low change in displacement of the non-linear cams at the knitting point. The curved shape of a non-linear cam is a necessary feature for continuous acceleration of the needle butt, but it inevitably means that more knitting elements are involved in loop formation than with linear cams. The cam shapes may also be used to show the path of the yarn during loop formation. However, to avoid confusion, the yarn is not shown in Fig. 45 but the correct positions of the needles and sinkers are given. The yarn would wrap around the sinkers and needles as indicated previously in Fig. 41.

It is known from Amonton's Law (given below) that when yarn passes over a metal surface an increase in tension will occur; it is likely, therefore, that with a non-linear cam the build up in yarn tension during loop formation would be far greater than with the linear cam, due to the extra number of yarn/metal contacts, (i.e. knitting elements).

$$\frac{T_2}{T_1} = e^{\mu\theta}$$

where  $T_2$  = yarn output tension (gms)

$T_1$  = yarn input tension (gms)

$\theta$  = angle of <sup>wr</sup>lap (radians)

$\mu$  = coefficient of yarn/metal friction.

Knapton showed a method for calculating the build up in tension within the loop-forming portion, from the geometry of the knitting elements and the known value of yarn/metal friction; with this information he was able to explain a number of knitting effects observed when knitting with linear cams.

FIG 46

MEASUREMENT OF YARN / METAL FRICTION

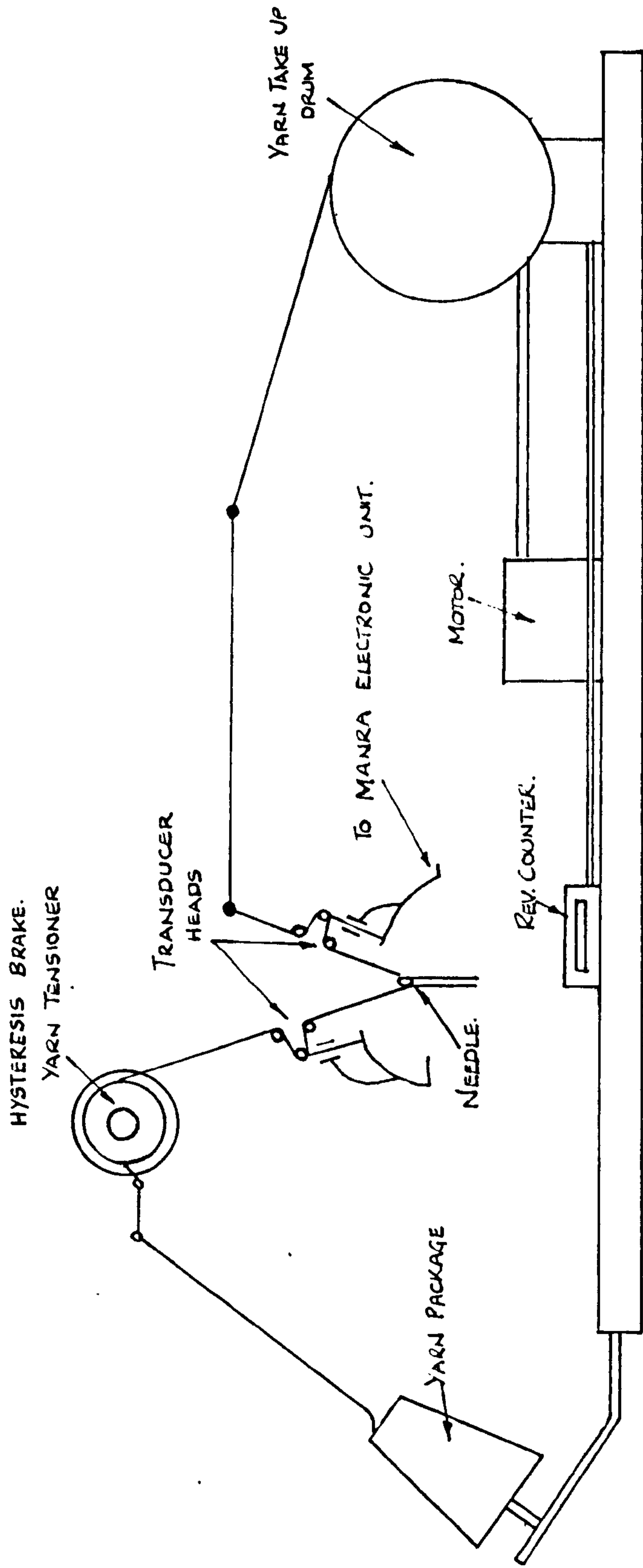
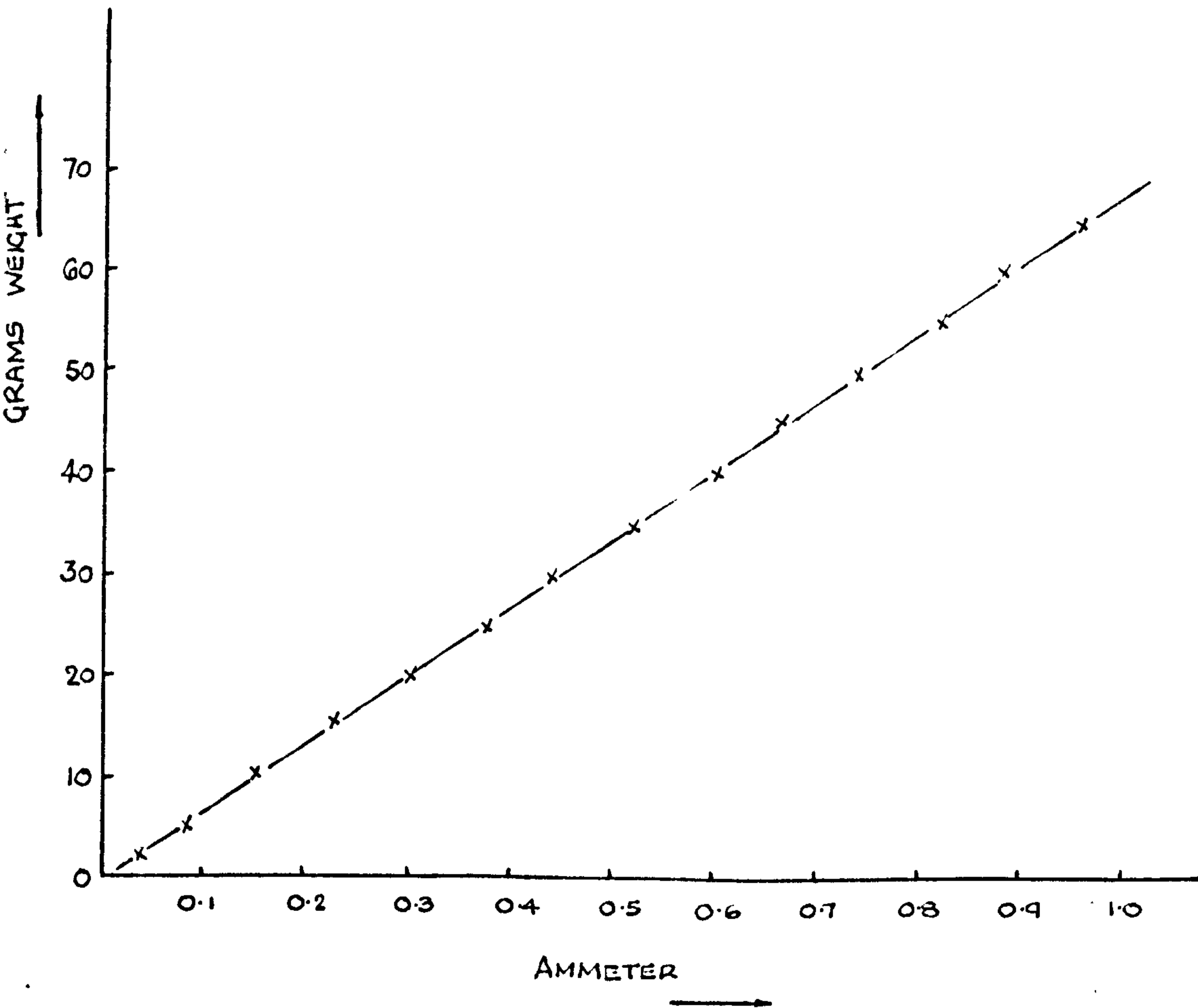


FIG 47

CALIBRATION FOR TRANSDUCER / ELECTRONIC UNIT N°1.



(b) Measurements and calculations of yarn/metal friction

In order to calculate the values of maximum tension within the loop-forming portions of a cam system, it was first necessary to establish a value of yarn/metal friction, under similar conditions to those present when actually knitting a yarn. An apparatus for obtaining the yarn/metal friction values was developed by Knapton (8) and was used in this work.

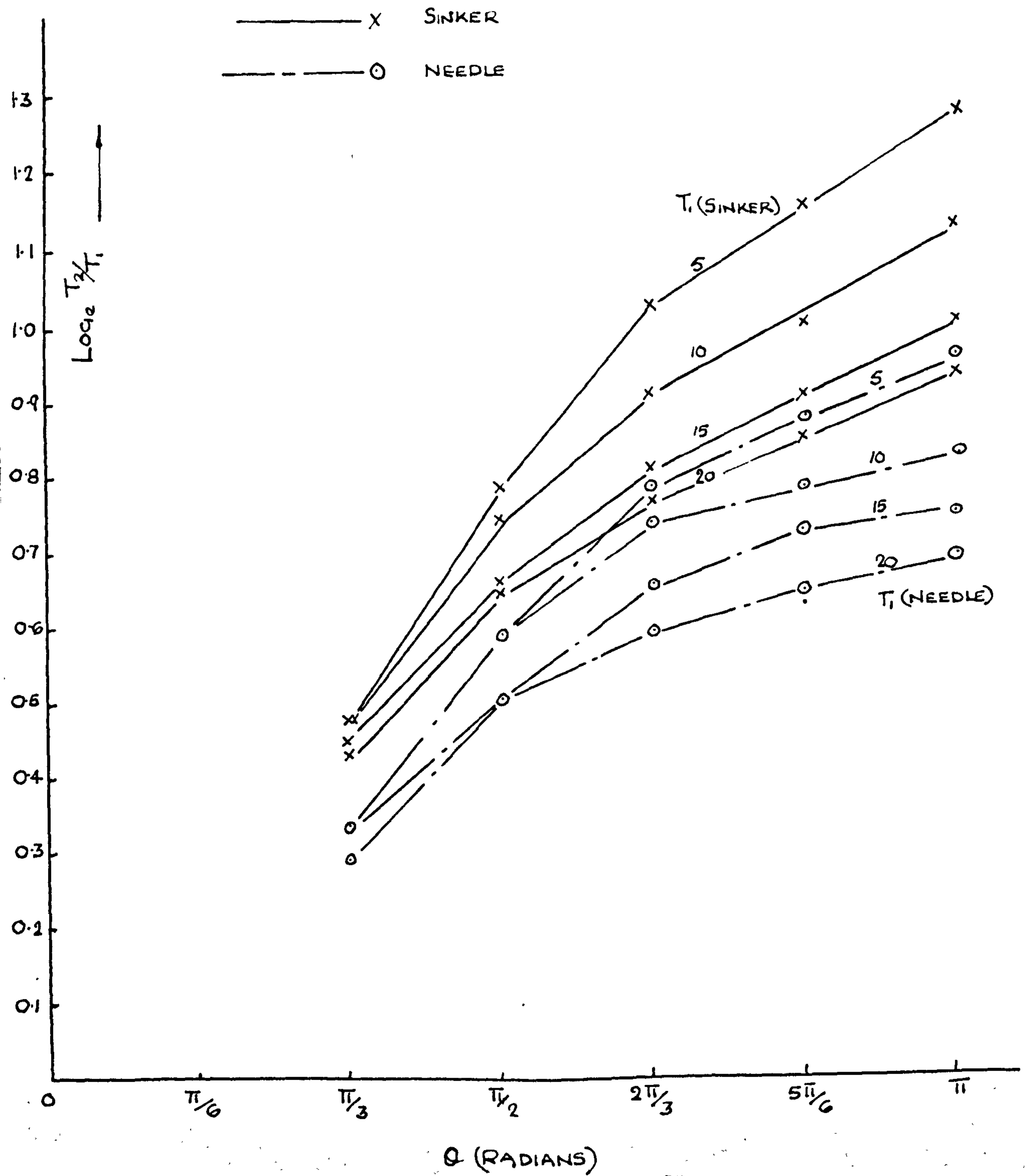
The apparatus which is shown in Fig. 46, consists of a yarn take-up drum rotating at approximately the speed of a knitting machine; two transducer heads with electronic amplifying meters to measure accurately and record the tension of the yarn before and after it passed over the metal surface, and a hysteresis brake unit to apply a constant yarn input tension.

The transducer head units were calibrated by hanging weights, in increments of 5 gms, up to a maximum of 50 gms, on the three pulley system of the unit, and the meter reading was recorded for each weight addition. A typical calibration graph for the output tension unit is shown in Fig. 47.

To determine a value for yarn friction, the yarn under test was passed around the metal surface at a known angle of lap ( $\theta^\circ$ ), the input tension was pre-set to a particular value, and the output tension measured on the electronic meter; using Amonton's Law the values of friction under these conditions could then be obtained. To simulate actual knitting conditions, a sinker and needle, taken from the machine used in this work, were used as the metal test surfaces.

It was realised that there were only two important angles of lap during loop formation, (i.e.  $60^\circ$  and  $180^\circ$ ).

Fig 48



For instance, it has been shown in Fig. 41 where the actual yarn diameter and knitting element dimensions were drawn to scale, that the yarn makes an angle of  $180^\circ$  with the knitting elements during loop formation, and this would apply to all the knitting elements in the loop-forming process, apart from the first angle of lap, (i.e. when the yarn first made contact with a knitting element). Examination of the feeding position of the yarn showed that the first angle of lap was approximately  $60^\circ$ , irrespective of whether the yarn met a needle or sinker first. However, for a preliminary analysis of the friction results, angles of lap of  $60^\circ$ ,  $90^\circ$ ,  $120^\circ$ ,  $150^\circ$  and  $180^\circ$  were chosen.

Using a  $\frac{1}{2}$  worsted yarn, the input tension ( $T_1$ ) was set to 5, 10, 15 and 20 gms. and the corresponding output tension ( $T_2$ ) was measured for both the sinker and needle, at all the different angles of lap.

If these results follow Amonton's Law, then  $\log_e \frac{T_2}{T_1}$  plotted against  $\theta$  will provide a linear relationship passing through the origin, and the slope will be equal to the value of yarn/metal friction ( $\mu$ ). These values are shown in Table 18\* and plotted in Fig. 48.

Several points of importance were noted from Fig. 48. For example, the curves were not linear since the friction value decreased with an increased angle of lap, and the curves did not pass through the origin. There were different curves for each value of input tension, due to a decrease in value of friction with an increase in input tension. It was also noted that although both the sinker and needle exhibited the same trends, the friction values of the sinker at almost all conditions were higher than those of the needle.

\* See Appendix

It was clear from the above observations that Amonton's Law could not be used for these results to give an accurate average value of yarn/metal friction.

It has been stated that the important angles of lap for loop formation were  $60^\circ$  and  $180^\circ$ ; since the  $60^\circ$  lap only applies for the first tension change in the knitting elements this change may be obtained directly from the experimental results of  $T_1$  and  $T_2$ . However, for the remaining build up in tension at  $180^\circ$  lap, it was considered that Howell's formula (given below) would be more suitable for an accurate value of yarn/metal friction, as this relationship is dependent upon input tension and radius of curvature (20).

$$\frac{T_2}{T_1} = e^{C\theta} \quad \text{where } C = k \left[ \frac{R}{T_1} \right]^{1-n} \quad \dots (V.4);$$

$R$  = radius of metal surface (m/m)

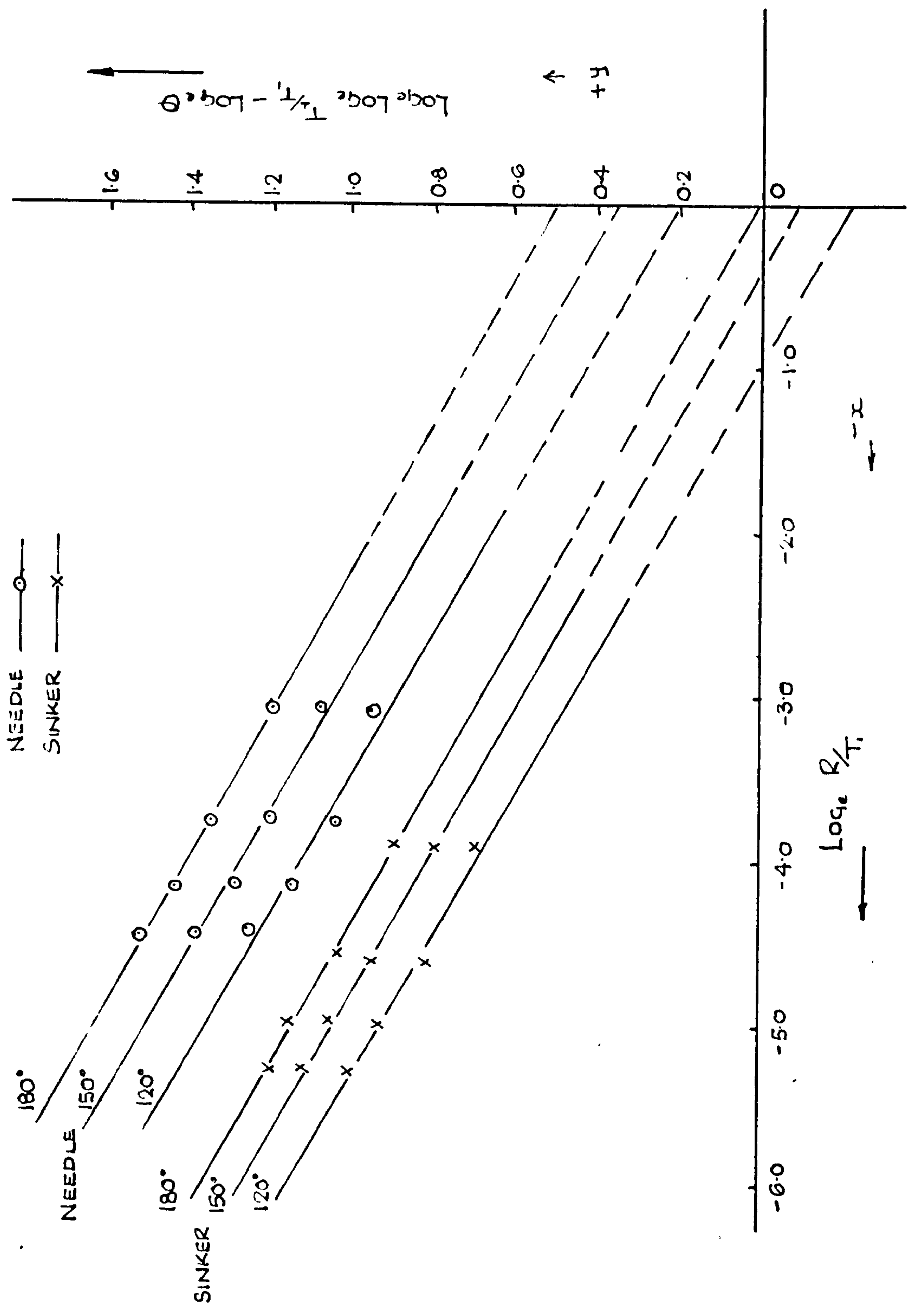
$C$  = coefficient of yarn/metal friction.

$k$  and  $n$  = frictional indices

Although this relationship would take into account the effects of input tension and radius of curvature, the changes in friction which occurred with changes in angle of lap would still be unaccounted for, i.e.  $\log_e \frac{T_2}{T_1}$  when plotted against  $\theta$  for Howell's expression should give a linear relationship as in Amonton's Law.

For a complete analysis, therefore, angles of lap of  $120^\circ$  and  $150^\circ$  were examined in addition to the value of  $180^\circ$ , and each angle of lap was examined separately when applying Howell's formula.

FIG 49





If Howell's Law is obeyed, then  $\log_e \log_e \frac{T_2}{T_1} - \log_e \theta$  plotted against  $\log_e \frac{R}{T_1}$  will provide a linear relationship, the slope will equal  $(1-n)$  and the intercept on the y axis will equal  $\log_e k$ .

The calculated values for the angles of lap  $120^\circ$ ,  $150^\circ$  and  $180^\circ$  for both sinker and needle are shown in Table 19\* and plotted in Fig. 49.

It may be observed from Fig. 49 that a linear relationship is obtained and the slopes are the same for each condition; different  $\log_e k$  values, however, are evident for the sinker and the needle and also for the different angles of lap, giving different values of friction in each case.

It has been suggested by Schoenmaker (21) that these differences in  $\log_e k$  relate to the fact that the actual angle of lap differs from the observed value, due to the extremely small and slightly differently shaped curvatures of the sinker and needle elements. However, at any given angle of lap, such as  $180^\circ$ , Howell's Law was obeyed, and therefore a value of  $(1-n)$  and  $\log_e k$  could be obtained.

The points on the graphs for the sinker and needle at  $180^\circ$  lap provided correlation factors of 0.987 and 0.983 respectively, indicating an excellent linear relationship. Calculations using the least mean square method provided the following  $n$  and  $k$  values.

<u>Needle</u>	$n = 0.7781$
	$k = 0.6060$
<u>Sinker</u>	$n = 0.777$
	$k = 0.9937$

\* See Appendix

The values of  $n$  and  $k$  may now be substituted into formula (V.4) and the yarn tension build up within the loop-forming portions may be calculated.

(c) Calculations of the values of maximum tension and the importance of their positions

(i) Calculation of maximum tension values

To obtain the tension build up from the input tension side of the loop-forming shapes, the first change in tension (i.e. when the yarn first meets a knitting element), was taken from the experimental results of the  $60^\circ$  angle of lap, (Table 18); the subsequent tensions were calculated at an angle of lap of  $180^\circ$  from Howell's formula (V.4) by substituting the appropriate  $n$  and  $k$  values, depending on whether the yarn passed around a needle or a sinker.

Before the tension build up for the reverse movement of the yarn (i.e. robbing back), could be calculated, it was necessary to establish whether the yarn was robbed back from only one needle or from two needles to the right of the knitting point. To do this, comparisons were made of the actual percentage robbing back, obtained from the stitch length results, and the maximum permissible percentage robbing back, calculated from the geometrical position of the knitting elements to the right of the knitting point. The geometrical robbing values were obtained in the following way.

The geometrical stitch length at position needle 5 (Fig. 45) for a  $45^\circ$  linear cam system at the medium cam setting, for example, may be calculated from formula (V.3),

i.e.  $l_t = 2H + 0.050''$  where  $H = \frac{G}{18} - 0.013''$  and G is measured in inches directly from Fig. 45.

The maximum possible percentage robbing back from needle 5 ( $N_5$ ) is thus obtained from

$$\frac{l_{t4} - l_{t5}}{l_{t4}} \times 100$$

where  $l_{t4}$  is the theoretical stitch length at the knitting point of the  $45^\circ$  linear cam system.

In this way the maximum geometrical robbing back values from positions  $N_5$  and sinker 6 ( $S_6$ ) for each cam system at the medium cam setting were calculated, and these values are shown below in Table 4, together with the actual percentage robbing values, which are extracted from Table 17 when knitting at 5, 10 and 20 gms input tension.

TABLE 4

Knitting Element	Calculated Stitch Length (ins) and Geometrical % Robbing Back												
	Linear Cams				Non-Linear Cams						St. & Curv		
	$45^\circ$		$55^\circ$		$52^\circ$		$57^\circ$		$62^\circ$		$55^\circ$		
	$l_t$	% RB	$l_t$	% RB	$l_t$	% RB	$l_t$	% RB	$l_t$	% RB	$l_t$	% RB	
$N_4$	0.234		0.226		0.202		0.216		0.222		0.226		
$N_5$	0.126	46.1	0.074	67.4	0.174	13.8	0.174	19.5	0.171	23.0	0.057	75.0	
$S_6$	-		-		0.152	24.8	0.146	32.4	0.126	43.2	-		
Yarn Tens (gm)	Measured Stitch Length (ins) and Actual % Robbing Back												
	$l_a$	% RB	$l_a$	% RB	$l_a$	% RB	$l_a$	% RB	$l_a$	% RB	$l_a$	% RB	
	5	0.176	24.8	0.176	22.1	0.176	12.9	0.176	18.5	0.176	20.7	0.176	22.1
	10	0.160	31.6	0.154	31.8	0.168	16.8	0.169	21.3	0.167	24.8	0.166	26.1
20	0.144	38.4	0.133	40.0	0.159	21.3	0.161	25.0	0.156	29.7	0.151	33.2	

It is evident from Table 4 that at 5 gms input tension the actual robbing back values are less than the maximum permissible robbing back values at  $N_5$ , for all the cam systems; this suggests, therefore, that yarn is only robbed back from the first needle after the knitting point, i.e.  $N_5$ .

For comparisons of maximum tension values at 5 gms input tension, the robbing back curve may be calculated using  $N_5$  as the starting position, and since no yarn movement will occur over sinker 6, the tension in the yarn between  $S_6$  and  $N_5$  may be determined by the take-down tension force on each loop - a value of 2 gms was obtained. The subsequent tension values were calculated from formula (V.4) in the same way as the input tension side.

The values of input tension build up, calculated from an initial yarn input tension of 5 gms are given in Table 5 for each cam system, together with the robbing back tension values; these tension values were plotted against the positions of the knitting elements, and the intercept of the robbing back curve and the input tension curves provides the maximum values and positions of maximum tension for each cam system, at the medium cam setting, (see Fig. 50).

FIG 50

VALUES AND POSITIONS OF MAXIMUM  
KNITTING TENSIONS FOR EACH CAM SYSTEM

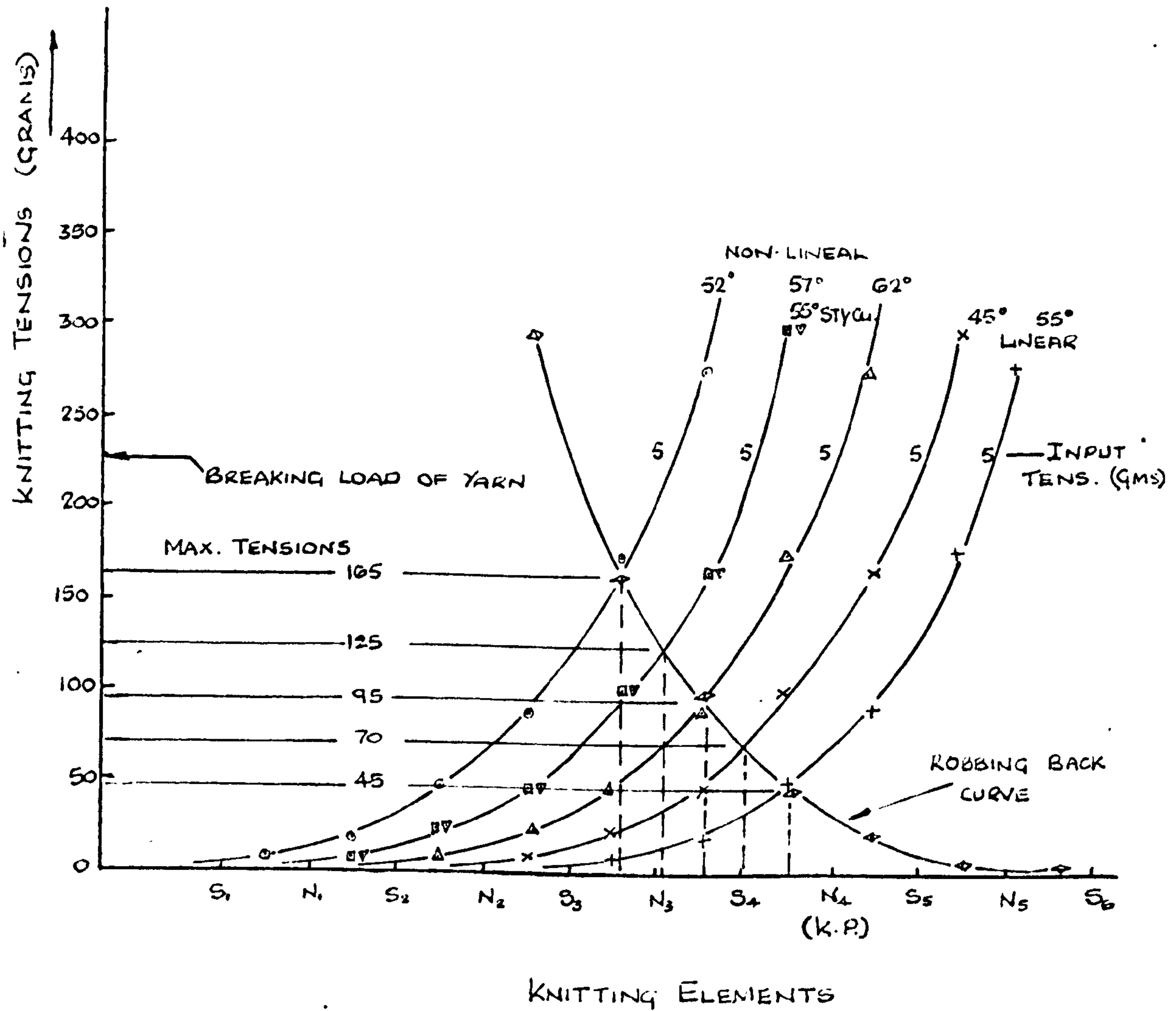


TABLE 5

Knitting Element Posit- ion	Yarn Tensions during Knitting (gms) - L.H.S.						R.B. Tensions
	Linear		Non-Linear			St. & Curv.	
	45°	55°	52°	57°	62°	55°	
(T <sub>1</sub> )	5	5	5	5	5	5	-
S <sub>1</sub>	-	-	8	-	-	-	-
N <sub>1</sub>	-	-	19.0	7	-	7	-
S <sub>2</sub>	-	-	49.8	23.2	8	23.2	-
N <sub>2</sub>	7.0	-	88.5	45.9	19.0	45.9	430
S <sub>3</sub>	23.2	8	175	101	49.8	101	291
N <sub>3</sub>	45.9	19.0	271	166	88.5	166	160
S <sub>4</sub>	101	49.8	460	301	175	301	97.3
N <sub>4</sub>	166	88.5		443	271	443	43.8
S <sub>5</sub>	301	175			460		21.9
N <sub>5</sub>	443	271					6.5
S <sub>6</sub>							2.0

It may be seen from Fig. 50 that there is a significant difference in maximum tensions between non-linear and linear cams and, as expected, the 52° non-linear gives the largest value of maximum tension (i.e. 165 gms) and the 55° linear provides the lowest value of maximum tension (i.e. 45 gms). Although the calculated value of maximum tension is high for the 52° non-linear cam, it is still below the breaking load of the yarn of 228.4 gms, and this agrees with the practical results which showed that this cam system provided satisfactory knitted fabric at yarn input tensions up to 15 gms.

It may be shown that no extra knitting elements are introduced when the cam settings of each cam system are increased from the medium to the slack setting and therefore the tension build up and maximum tension values shown in Fig. 50 apply to both medium and slack cam settings. With a decrease in cam setting from medium to tight, a reduction of one knitting element occurs, for most cam systems, and the maximum tension values are reduced. The significance of this and the effect on the horizontal forces acting on the needle, will be discussed later in part C.

(ii) Positions of maximum tension value

Knapton showed that the position of maximum tension was extremely important, as it could be related to the stitch length measured from the fabric. To illustrate this it may be seen that the position of maximum tension from the 55° linear cam system lies between S<sub>4</sub> and N<sub>4</sub> (Fig. 50). Referring back to Fig. 45 the geometrical stitch length calculated from formula (V.3) at this position should be approximately equal to the actual stitch length of 0.176".

Calculations of the geometrical stitch length from the respective positions of maximum tension for each cam system are shown in Table 6, and they illustrate clearly that there is a close relation between these calculated values and the measured values of stitch length from the fabric (i.e. 0.176").

TABLE 6

Calculated stitch length (ins)					
Linear		Non-linear			St. & Curv.
45°	55°	52°	57°	62°	55°
0.178	0.182	0.174	0.179	0.185	0.181

The positions of maximum tension for the non-linear cams were further to the left of the knitting point than those for the linear cams and yet the calculated geometrical stitch length was still virtually the same (see Fig. 50). This was easily explained, as examination of the loop-forming curves (Fig. 45) showed that the non-linear cams had a much slower rate of change of displacement than the linear cam at the knitting point, and therefore the position of maximum tension with non-linear cams must be displaced to the left in order to provide the same calculated stitch length.

An examination of the values and positions of maximum tension may be used to illustrate why the linear cam system required a lower cam setting than the non-linear cam to provide the same stitch length - this aspect was observed previously in Fig. 39.

It may be seen from Fig. 45 that after the needle reaches the position of maximum tension, its further vertical displacement is smaller in the case of the non-linear cam system than in the case of the linear cam system - i.e. less robbing back is required from the needle when using the non-linear cam. This also confirms the previous observations in Fig. 42, where it was shown that less robbing back occurred with non-linear cams than with linear cams.

(d) Effect of Increasing Yarn Input Tension on the Values and Positions of Maximum Tension

It has been shown in Table 4 that, at 5 gms input tension, the needle at the knitting point robbed yarn back from  $N_5$  only, for all cam systems. In addition, Table 4 shows that when the yarn input tension was increased to 10 and 20 gms using the linear cams, the values of actual robbing back were still less than the





maximum permissible robbing back values from the  $N_5$  position. Therefore for tension build up, calculated from input tensions of 10 and 20 gms, the robbing back curve for the linear cams will still commence at  $N_5$  with 2 gms tension.

With non-linear cams, however, it is evident from Table 4 that the actual robbing back values at 10 and 20 gms are now greater than the maximum permissible robbing back values provided from  $N_5$ , and therefore extra yarn must be drawn from the second needle after the knitting position (i.e.  $N_6$ , Fig. 45). This suggests that the robbing back curve will now be displaced to the right, when using the non-linear cams at these higher tensions, as a new starting position will be required.

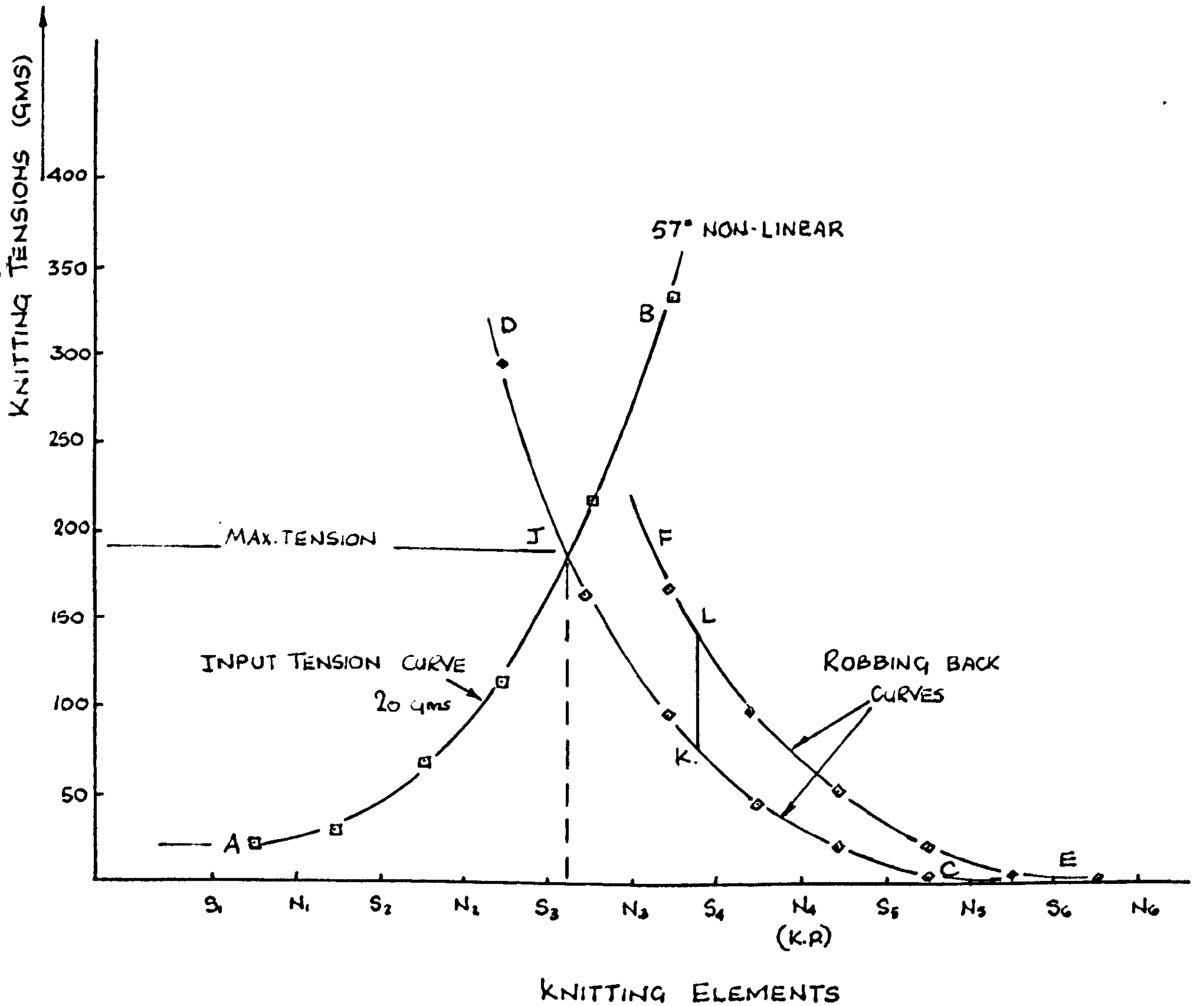
To determine this new position it was realised that since the needle butts were positively controlled in the non-linear cams, an upward movement of the needle butt would allow a certain amount of extra yarn to be drawn from between  $N_6$  and  $S_6$  (Fig. 45), before robbing back over the hook of needle 6 was necessary.

To establish whether this extra length of yarn was sufficient, calculations of the maximum permissible robbing back were made for the condition when the needle was in the  $S_6$  position. These values for each non-linear cam have been shown in Table 4, and it may be seen that they do provide a greater percentage value than the actual robbing back values for 10 and 20 gms.

This new starting position (i.e. tension = 2 gms in the  $S_6-N_6$  region), will therefore be used to calculate a new robbing back curve for conditions when yarn is drawn from two needles after the knitting point. However, to understand the tension

FIG 51

THE VALUE AND POSITION OF MAXIMUM TENSION  
WHEN ROBBING BACK FROM TWO NEEDLES.



build up in the non-linear cams at 10 or 20 gms tension, it is necessary to consider both the robbing back curves, i.e. the new robbing curve starting in the  $S_6-N_6$  region, and the previous robbing curve in the  $N_5-S_6$  region.

The calculated values of the input tension side from 20 gms for the  $57^\circ$  non-linear cam system, together with the calculated values of the new robbing back curve ( $S_6-N_6$ ), are shown in Table 20, and these values together with the robbing back values from  $N_5-S_6$ , given previously in Table 5, are plotted against the positions of the knitting elements (see Fig. 51).

From Fig. 51, AB represents the input tension curve starting at 20 gms, and EF and CD represent the two robbing back curves where EF is the new robbing back curve from the  $S_6-N_6$  region, and CD is the original robbing curve from the  $N_5-S_6$  region.

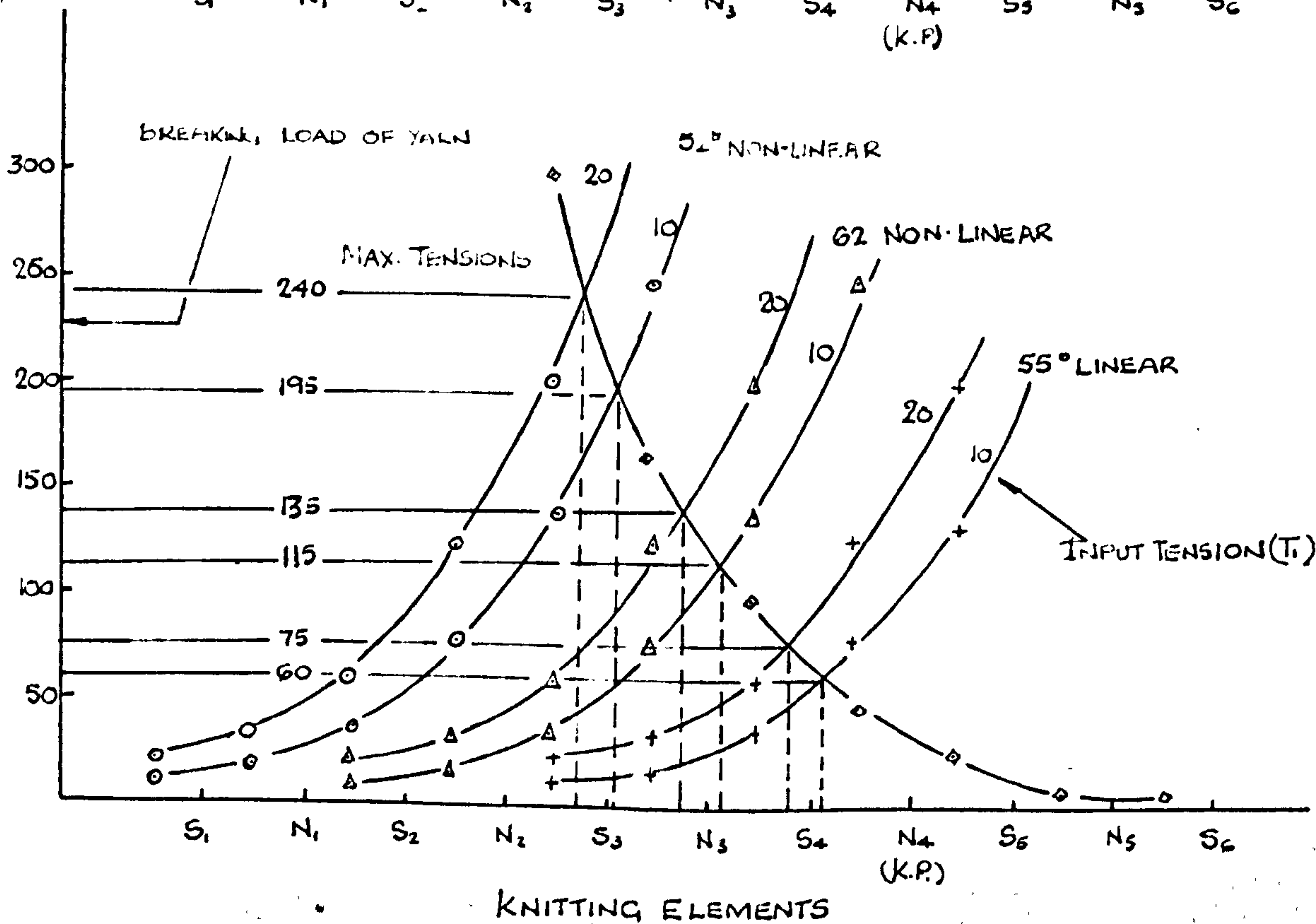
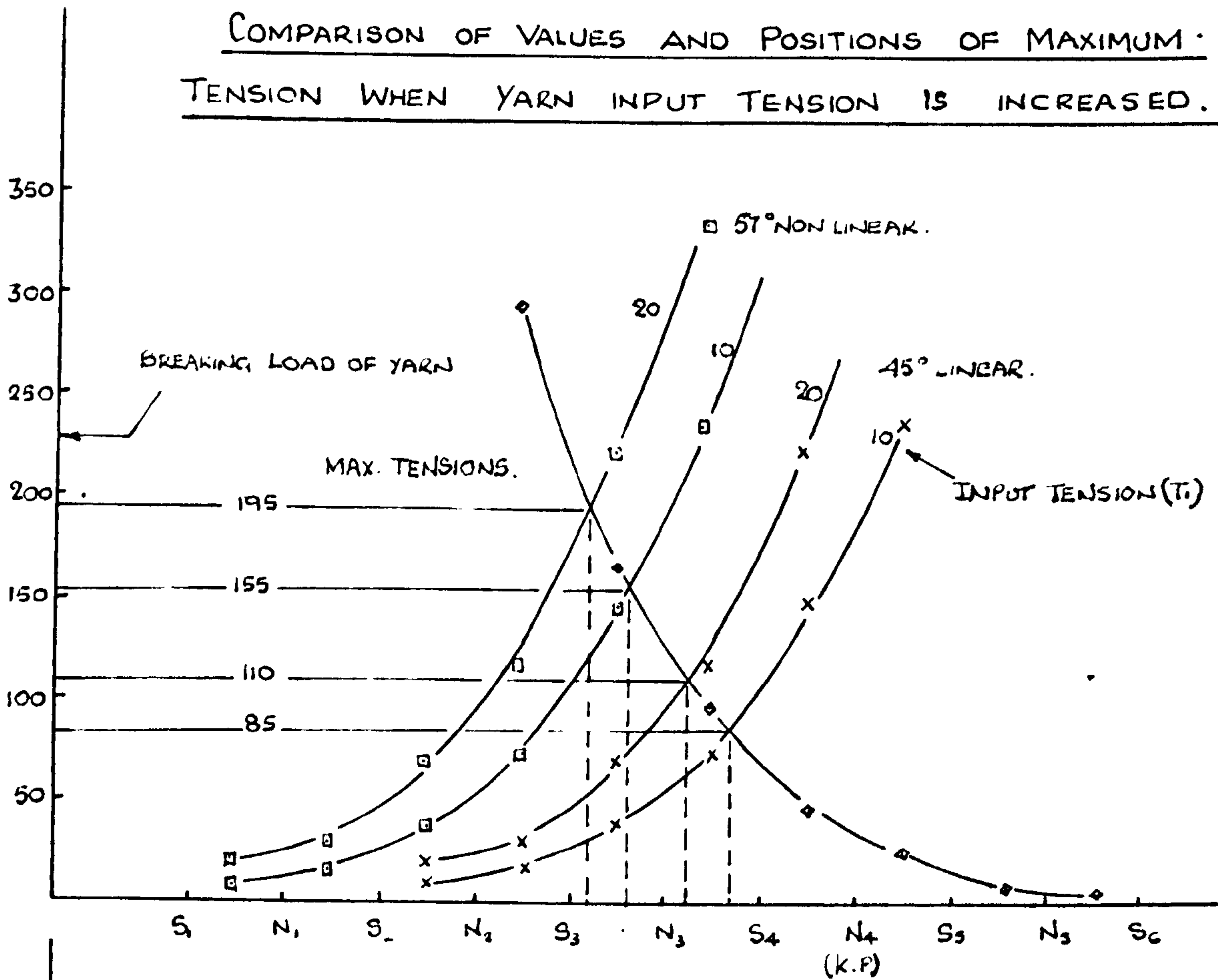
The position where the final stitch length is drawn is at position J (Fig. 51) and the geometrical stitch length at this point is 0.158", which agrees closely with the measured stitch length of 0.161" from Fig. 43. From position J robbing back will now proceed until the needle reaches the maximum vertical displacement at the knitting point ( $N_4$ ). The theoretical stitch length at the knitting point for the  $57^\circ$  non-linear cam system is 0.216" (see Table 16)\*; therefore, the total quantity of yarn to be robbed back will be  $0.216'' - 0.158'' = 0.058''$ . However, examinations of Table 4 have shown that the maximum quantity of yarn that can be supplied by  $N_5$  in the  $57^\circ$  non-linear cam system is  $0.216'' - 0.174'' = 0.042''$ . Therefore, at a new position K, (Fig. 51), where  $l_t = 0.200''$ , the descending needle will require more yarn (i.e.  $0.058'' - 0.042'' = 0.016''$ ) which must be drawn from the second needle after the knitting point.

\* See Appendix

FIG 52

COMPARISON OF VALUES AND POSITIONS OF MAXIMUM TENSION WHEN YARN INPUT TENSION IS INCREASED.

KNITTING TENSIONS (GMS)



This will result in a rise in tension at position K to a value L where the intercept with the new robbing back curve (EF) is made - it will then be possible to rob back the final yarn length of 0.016 inches.

It may be seen that there are two high tension peaks when yarn is robbed from two needles, but the important one for estimating both the stitch length and the maximum tension is the one determined by position J. This suggests that to estimate the maximum tensions and the stitch length for any of the cam systems, under all conditions of input tension, it is necessary only to consider the intersection of the input tension and the robbing back curve calculated from the first needle after the knitting point (i.e.  $N_5-S_6$  region).

The input tension build up values for all the cams at 10 and 20 gms are shown in Table 21\* and are plotted against the position of the knitting elements in Fig. 52.

Examinations of the curves for any cam system (Fig. 52) show that an increase in input tension produces an increase in maximum tension value, and the position of maximum tension is displaced further to the left of the knitting point. Values of the geometrical stitch length were calculated from these positions of maximum tension and are shown in Table 7, together with the measured stitch length values extracted from Table 17.

\* See Appendix

TABLE 7

Yarn Input Tension (gms)	Linear		Non-Linear			
	45°	55°	52°	57°	62°	
	Measured Stitch Length (ins)					
10	0.160	0.154	0.168	0.169	0.167	
20	0.144	0.133	0.159	0.161	0.156	
	Calculated Stitch Length (ins)					
	10	0.158	0.153	0.163	0.168	0.172
	20	0.140	0.130	0.156	0.158	0.161

These results give stitch lengths which are in remarkably good agreement with those measured from the resultant fabric. The values of maximum tension were also found to correspond closely with the practical results. For instance, knitting trials showed that occasional holes appeared in the fabric made with a 52° non-linear cam, at input tensions above 15 gms. Examination of Fig. 52 shows that the maximum tension value for the 52° non-linear cam at 20 gms input tension is 240 gms, which is slightly higher than the breaking load of the yarn (228 gms.) Thus, both practical and calculated evidence suggests that 15 - 20 gms is the maximum knitting input tension that can be used with this cam system to produce a satisfactory fabric using a  $\frac{1}{20}$  worsted yarn.

With the linear cams, satisfactory fabric could be knitted at yarn input tensions of up to 30 gms, and examination of the maximum tension values showed that they would still be below the breaking load of the yarn.

An examination of the values and positions of maximum tension illustrated why significant differences in measured stitch length occurred between the linear and non-linear cams, when the yarn input tension was increased - a feature that was observed previously in Fig. 43.

It may be seen from Fig. 52 that for a given change in input tension, the movement of the position of maximum tension to the left of the knitting point was approximately the same irrespective of the type of cam system. Examination of the shapes of the cam (Fig. 45) explained why this should occur - the non-linear cams have a slow change in displacement and therefore a given movement to the left of the knitting point provides only a relatively small decrease in the distance of the needle below the sinker level, and therefore in the calculated stitch length. With linear cams, however, the same movement to the left of the knitting point gives much greater changes in depth below sinker level, and therefore greater differences in calculated stitch length.

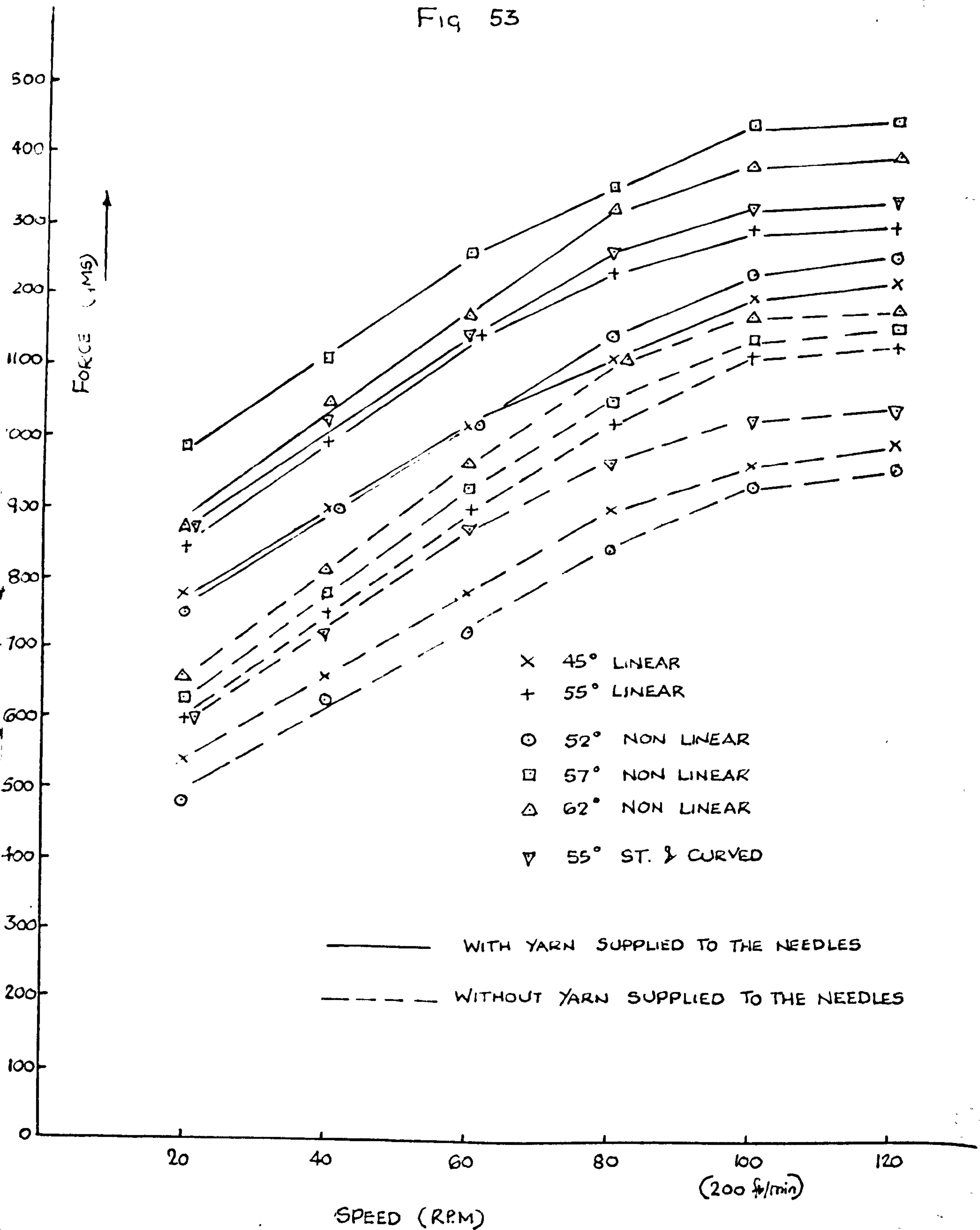
### C. Comparison of the Forces Developed within the Various Cam Systems

A description of the experimental procedure when using the torque device or the force measuring device, has been given in Chapter 4. In the following experiments most of the results for the practical analysis were obtained using the more sensitive strain gauge force measuring device. However, for certain experiments, such as when severe knitting conditions were applied, the torque device was more suitable for obtaining the results.

#### 1. Measurements of forces with and without yarn when the machine speed was increased

The force values were obtained from the force measuring device at machine speeds ranging from 20 - 120 r.p.m., when

Fig 53





using each cam, set at the slack setting. The values are given in Table 22\* and the force/speed relationships with and without yarn are shown in Fig. 53.

An examination of the forces without yarn (Fig. 53) shows that the force increases with speed for all cam systems, and the rate of force increase appears to be dependent upon the maximum angle of the cam irrespective of whether the cam is linear or non-linear. For instance there is a greater increase in force for the  $62^{\circ}$  non-linear, from 20 - 120 r.p.m. than for the  $45^{\circ}$  linear cam.

The measured increase in force with increase in speed is a large effect; for example, with the  $45^{\circ}$  linear cam the force increases from 540 gms to 990 gms with an increase in speed of from 20 - 120 r.p.m., i.e. almost 100% increase in force - similar large increases in force may be observed for the other cam systems.

When yarn is supplied to the needles (Fig. 53), the force also increases with speed for all cam systems. The difference between the forces with and without yarn for any cam at any speed gives the force component due to the insertion of yarn to the needle hooks. It may be observed from Fig. 53 that the yarn force component is reasonably constant with speed for all the cams.

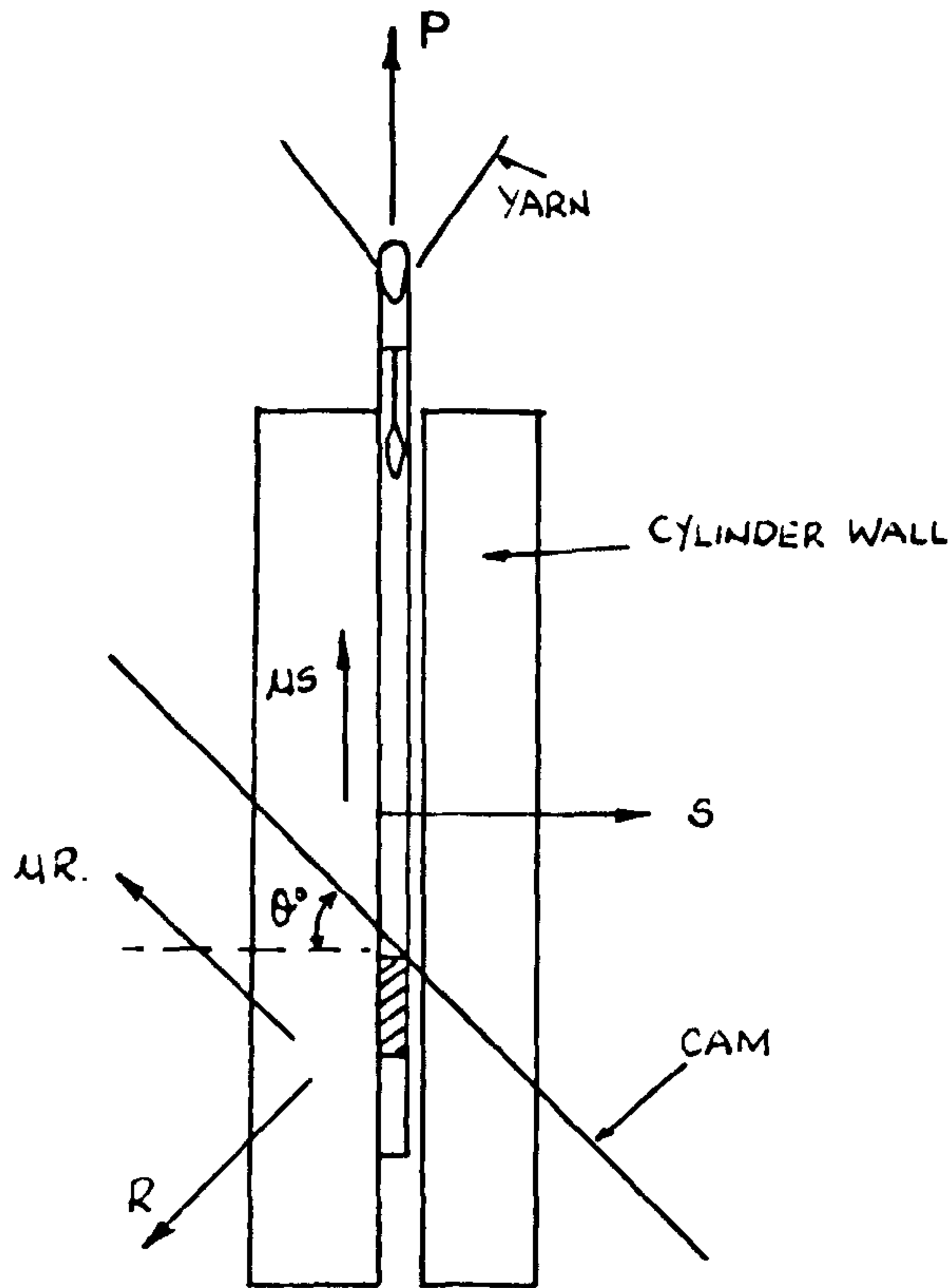
## 2. Expressions for the forces acting on the needles

Munden (7) has given the following expression for calculating the forces acting on the needle in linear cam systems; the forces are shown in Fig. 54.

\* See Appendix

Fig 54

FORCES ACTING ON THE NEEDLE.



RESOLVING VERTICALLY

$$P + \mu S + \mu R \sin \theta = R \cos \theta$$

RESOLVING HORIZONTALLY

$$\mu R \cos \theta + R \sin \theta = S$$

$$\therefore S = P \left[ \frac{\mu \cos \theta + \sin \theta}{(1 - \mu^2) \cos \theta - 2\mu \sin \theta} \right]$$

$$R = \frac{S}{\mu \cos \theta + \sin \theta}$$

WHERE :

S = FORCE REQUIRED TO MOVE NEEDLE DOWN THE CAM TRACK.

R = FORCE ACTING BETWEEN THE CAM FACE AND THE NEEDLE BUTT

$$S = \frac{P(\mu \cos \theta + \sin \theta)}{\{(1 - \mu^2) \cos \theta - 2\mu \sin \theta\}}$$

where S = horizontal force required to move the needle up or down the cam track (gms);

$\theta$  = cam angle (degrees);

$\mu$  = coefficient of friction between the needle butt and cam track, and between the needle shank and the trick wall;

P = force resisting the vertical movement of the needle (gms).

Knapton suggested that with the introduction of yarn to the needles the same form of equation would apply, but the value of P would now be given by

$$P = P_F + P_\theta$$

where  $P_F$  is the force resisting the vertical movement of the needles, and  $P_\theta$  is the force on the needle hook due to the tensions in the yarn,

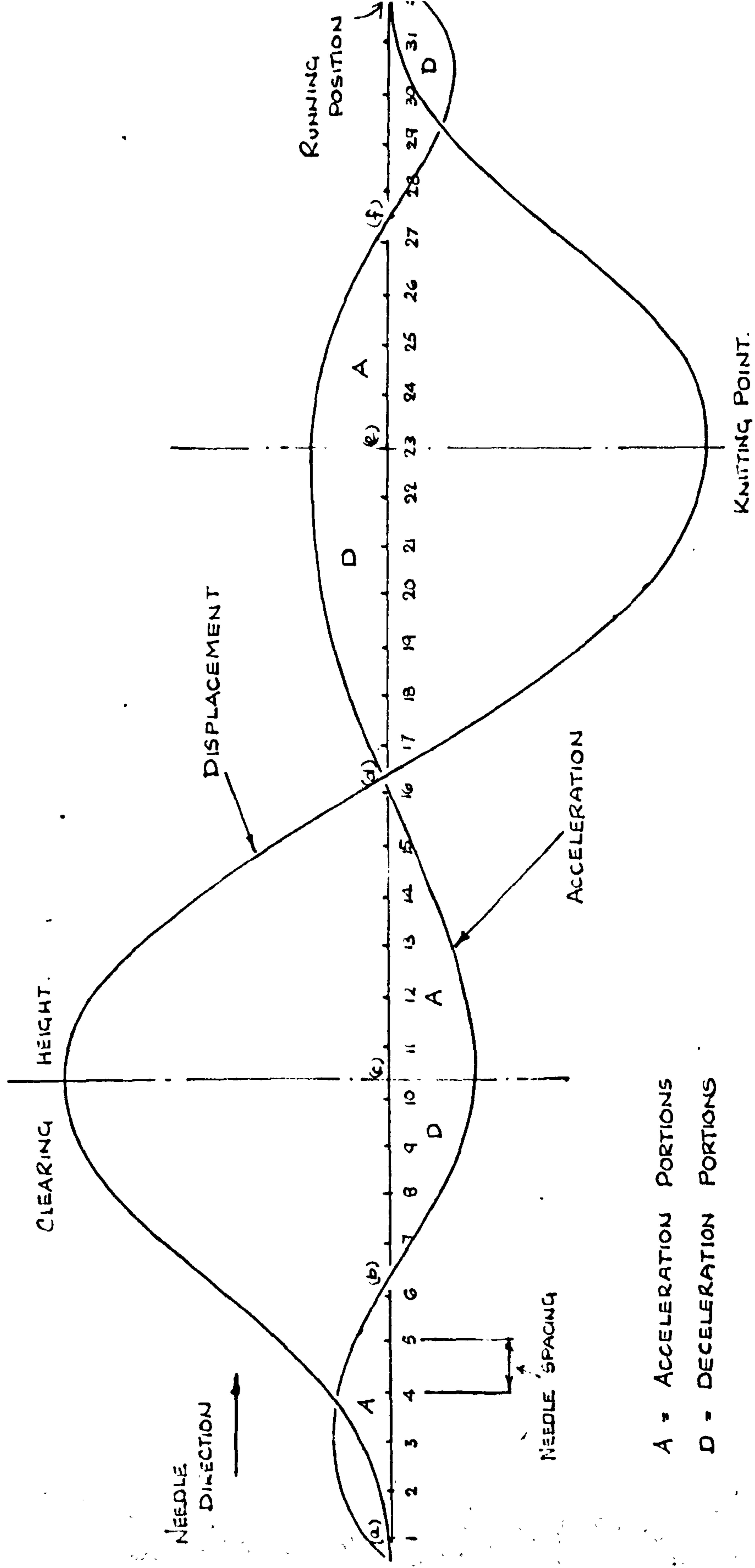
i.e. 
$$S = \frac{(P_F + P_\theta) (\mu \cos \theta + \sin \theta)}{\{(1 - \mu^2) \cos \theta - 2\mu \sin \theta\}}$$

or 
$$S = \frac{P_F (\mu \cos \theta + \sin \theta)}{\{(1 - \mu^2) \cos \theta - 2\mu \sin \theta\}} + \frac{P_\theta (\mu \cos \theta + \sin \theta)}{\{(1 - \mu^2) \cos \theta - 2\mu \sin \theta\}}$$

Therefore, the horizontal force required to move the needle up or down the cam track without yarn is given by

$$S_1 = \frac{P_F (\mu \cos \theta + \sin \theta)}{\{(1 - \mu^2) \cos \theta - 2\mu \sin \theta\}} \quad \dots (V.5)$$

# DISPLACEMENT AND ACCELERATION CURVES FOR THE 57° NON-LINEAR CAM



NEEDLE DIRECTION →

RUNNING POSITION

CLEARING HEIGHT

DISPLACEMENT

ACCELERATION

KNITTING POINT

(f)

A

D

(d)

(c)

D

(b)

A

(a)

NEEDLE SPACING

A = ACCELERATION PORTIONS

D = DECELERATION PORTIONS

SCALE :

DISPLACEMENT 5:7:1

ACCELERATION  $15 \text{ ins}^{-1}$

=  $5.9 \text{ cm}^{-1}$

and the additional component of horizontal force due to the insertion of yarn to the needle hooks is given by

$$S_2 = \frac{P_\theta (\mu \cos \theta + \sin \theta)}{\{(1 - \mu^2) \cos \theta - 2\mu \sin \theta\}} \dots (V.6)$$

3. The forces acting on the needles without yarn supplied to the needle hooks

(a) Calculation of the forces

Equation (V.5) was used to calculate the forces on the needles for each of the designed cam systems. For these calculations a value of  $\mu = 0.1$  was assumed a reasonable value for well lubricated hardened steel components, and an average value of  $P_F = 10$  gms was obtained by finding the average weight necessary to move the needle vertically in its trick - this was measured by hanging weights on the needle butts.

For the  $45^\circ$  and  $55^\circ$  linear cam systems, the calculation of the force  $S_1$  was straight forward, as  $\theta^\circ$  was equal to  $45^\circ$  and  $55^\circ$  respectively for all the needles acting in the cam systems; the total force (i.e.  $\Sigma S_1$ ) was obtained from  $n \times S_1$  where  $n =$  number of needles acting in the cam system.

With non-linear cam systems and the  $55^\circ$  straight and curved cam, the angle  $\theta$  varied throughout the cam profile, and to obtain a value of force  $\Sigma S_1$  it was necessary first to measure the value of  $\theta$  for each needle position. For example, Fig. 55 shows the displacement curve for the leading edges of the  $57^\circ$  non-linear cam, and, the needles acting on this cam system are shown numbered 1 - 32. At any needle position the angle of  $\theta$  may be measured quite

accurately (or calculated as shown in section C, Chapter 2) and the value of  $S_1$  for each needle may be calculated from formula (V.5). The value of horizontal force  $\Sigma S_1$  is obtained, therefore, from the sum of the  $S_1$  values calculated from all the needles acting on this cam system.

These calculated values for the non-linear cam systems and those obtained for the linear cam systems are shown below in Table 8.

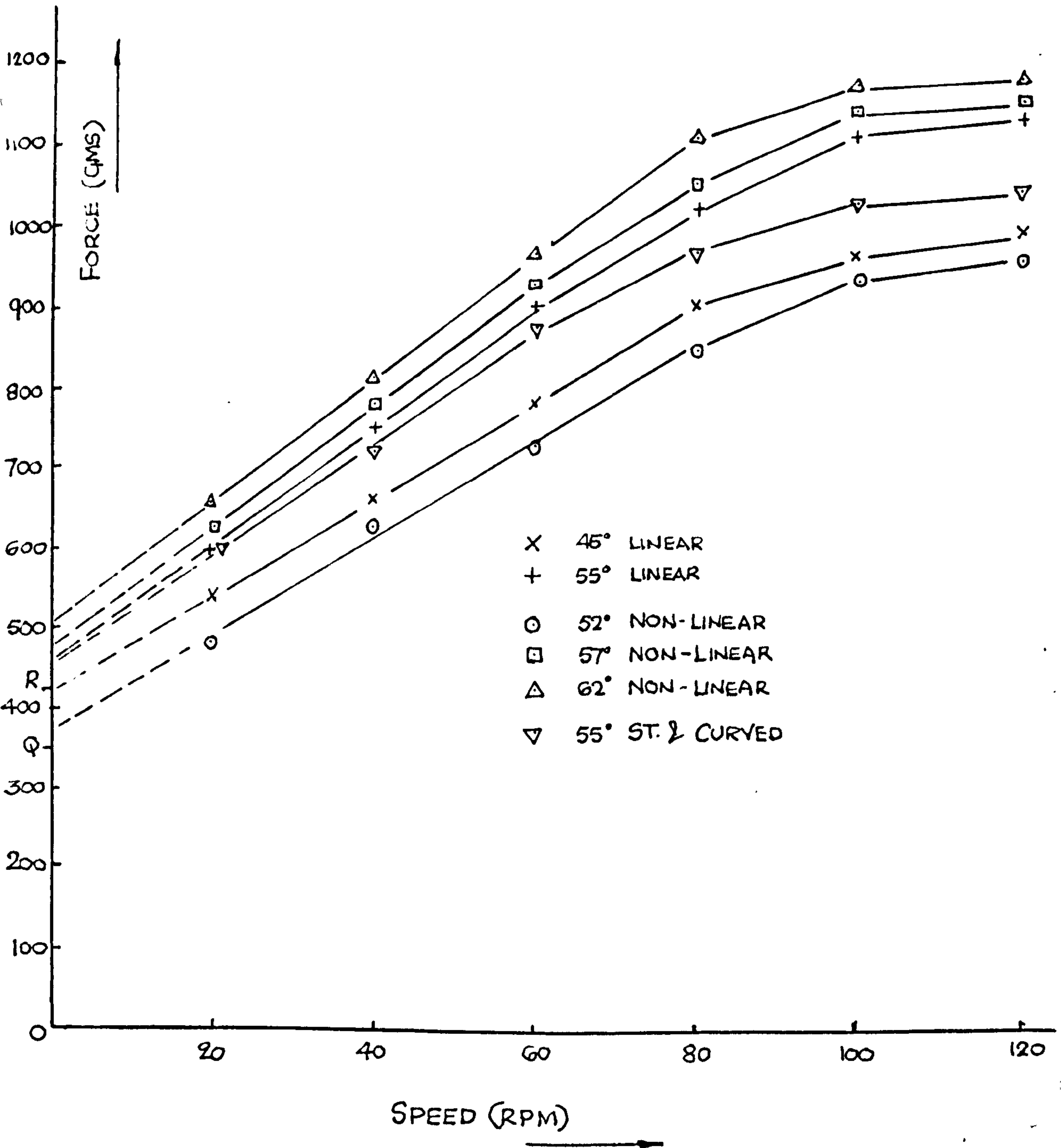
TABLE 8

Cam System	Nos. of Needles	$\Sigma S_1$ (gms)
45 Linear	25	348
55 Linear	18	394
52 Non-Linear	35	308
57 Non-Linear	32	328
62 Non-Linear	28	335
55 Str. and Curv.	41	353

✓ It is noted from Table 8 that although the non-linear cam systems have more needles acting on the cam track than the linear cams, the calculated horizontal force  $\Sigma S_1$  for non-linear cams are slightly smaller. This is due to the fact that a large proportion of the needles in the non-linear cams act on relatively low cam angles, with only a few acting at the maximum cam angle, whereas all the needles in the linear cams act at the maximum cam angle.

FIG 56.

FORCE / SPEED — WITHOUT YARN



(b) Comparison of the calculated and measured forces

The measured values of force/speed without yarn were extracted from Fig. 53 and are shown again in Fig. 56. The calculated force values  $IS_1$  given in Table 8 are seen to be smaller than the measured values, even at the lowest machine speed of 20 r.p.m. However, extrapolation of the measured values to zero speed in Fig. 56 provides reasonably close agreement with the calculated values. For example, from Table 8 the calculated force for the  $45^\circ$  linear cam = 348 gms and this value (shown at position Q in Fig. 56) may be compared with the extrapolated measured value for the same cam which equals 420 gms (shown at position R, Fig. 56).

Since the calculated force  $S_1$  is a relationship in which the mass of the needle butt is considered equal to zero and, therefore, impact forces at the ends of the linear cam are not taken into account, it seems reasonable for the calculated and measured force values to agree at zero speed.

As there is a difference in the values Q and R, however, the estimated value of friction ( $\mu = 0.1$ ) seems to be a slightly inaccurate assessment of the true conditions of friction in the knitting machine. For example, with the  $45^\circ$  linear cam a value of friction of  $\mu = 0.15$  is necessary to give a value of Q equal to the extrapolated value of force at zero speed (R). Different values of friction ( $\mu$ ) would be necessary for the other cams to give complete agreement between the calculated force values and extrapolated values of measured force at zero speed; since this extrapolation is a linear extension



of a non-linear relationship between force and speed, the values of friction ( $\mu$ ) thus obtained are not likely to be very accurate. In view of these considerations, the value of  $\mu = 0.1$  was used for all subsequent calculations of the forces.

(c) Attempts to explain the measured force increases which occur when the machine speed is increased

(i) Calculations of forces acting on the needle under dynamic conditions

In order to explain why there should be such significant increases in force with increase in machine speed, the forces needed to accelerate and decelerate the needle butts through the cam system were calculated. The needle butts will resist the acceleration with an inertia force

$$F_I = m\ddot{y} \quad \dots (V.7)$$

where  $F_I$  = inertia force (gms);

$\ddot{y}$  = acceleration and deceleration of the needle butt (cms/sec<sup>2</sup>);

$m = W/g$  where  $W$  is the weight of the needle (0.564 gms).

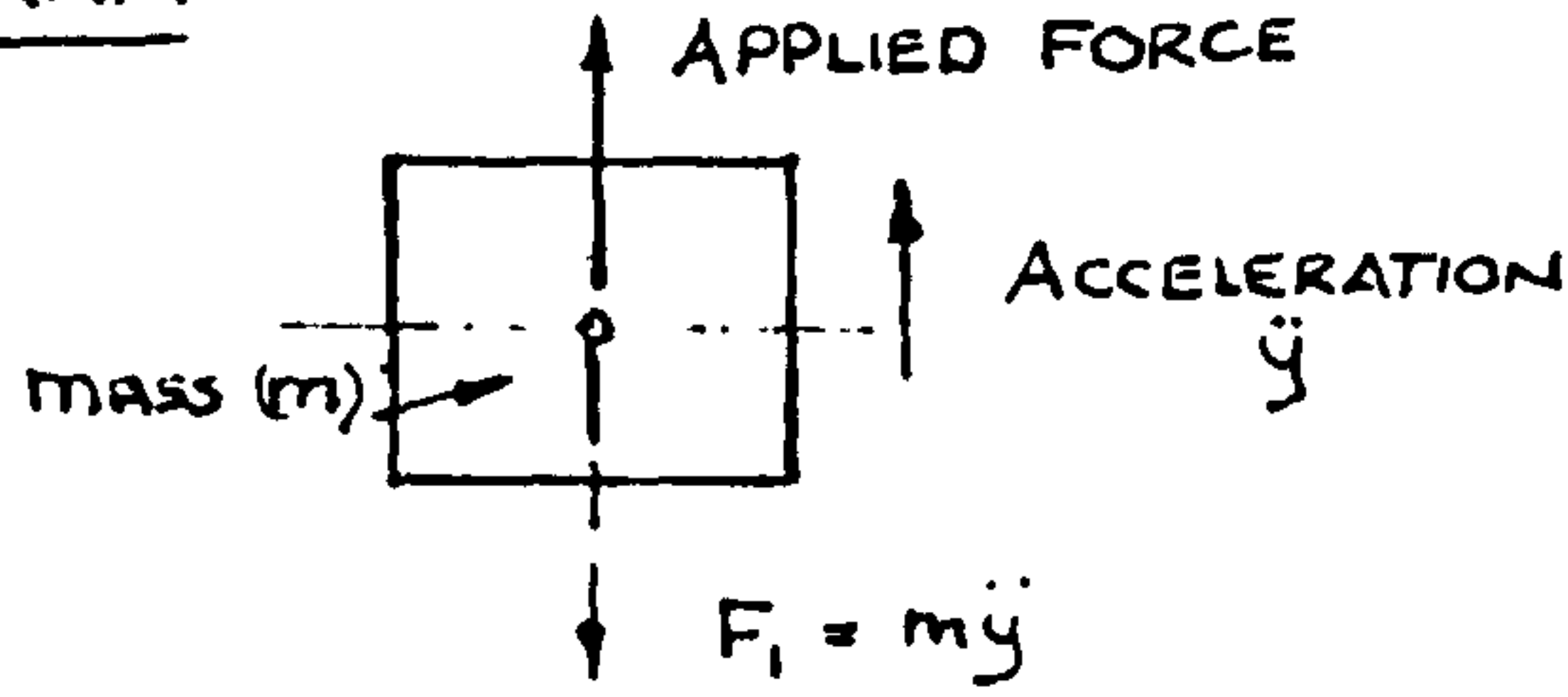
The inertia force will have a direction opposite to that of acceleration and, using D'Alembert's principle, a free body diagram of all forces may be made and the dynamic condition may be analysed as a static problem (Fig. 57.A).

With the linear cam systems it is known that the needle butt provides instantaneous changes in acceleration at the three points of impact within the cam system and,

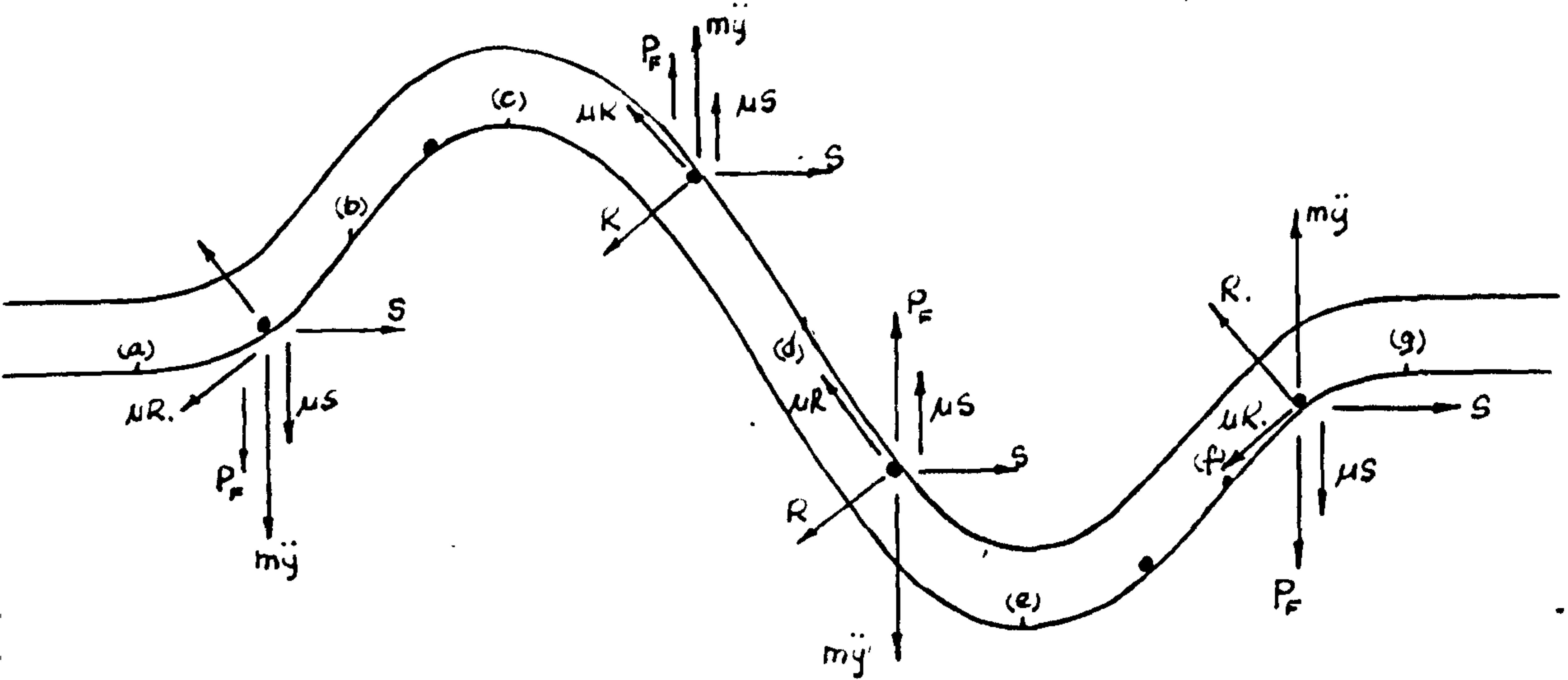
FIG 57

FORCES ACTING ON THE NEEDLES IN A NON-LINEAR CAM

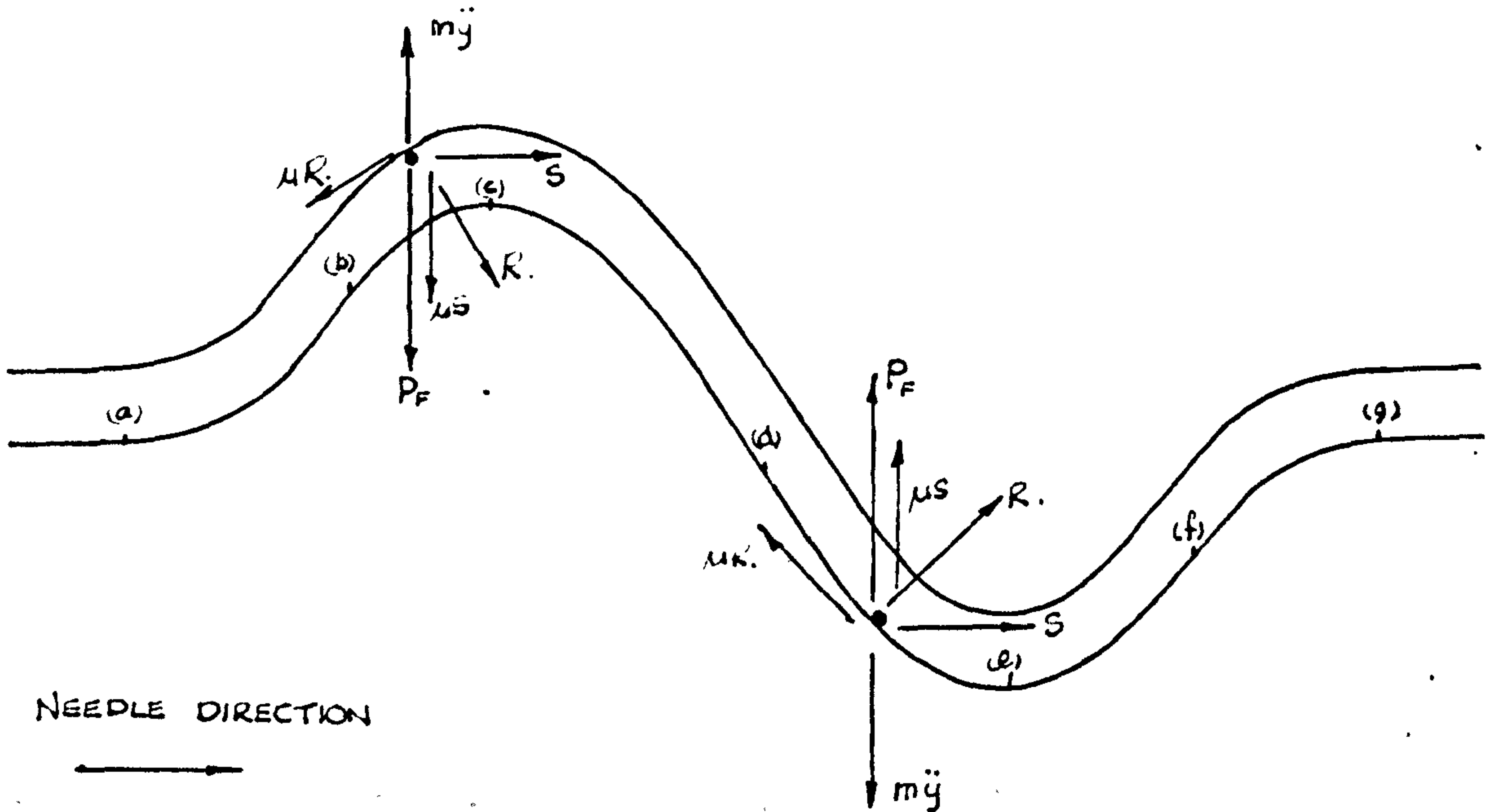
A FREE-BODY DIAGRAM



B FORCES - WHEN NEEDLE BUTTS ARE ON THE CAMS LEADING EDGES



C FORCES - WHEN NEEDLE BUTTS CROSS OVER TO NON LEADING EDGES



therefore, a value of inertia force cannot easily be evaluated for these types of cam system. However, it has been shown in Chapter 2 that non-linear cams have been designed to provide continuous velocity and acceleration characteristics and that a real value of acceleration ( $\ddot{y}$ ) at any position on the cam profile may be obtained from

$$\ddot{y} = v^2 y'' = v^2 \frac{d^2 y}{dx^2}$$

where the derivative  $d^2y/dx^2$  gives the vertical displacement on the theoretical acceleration curve.

The acceleration curve for the  $57^\circ$  non-linear cam is shown in Fig. 55 and at needle 3, for example, the vertical acceleration  $d^2y/dx^2$  is  $5.25 \text{ ins.}^{-1}$ . Therefore, at a horizontal needle velocity of 200 ft/min, (i.e. 100 r.p.m.) the acceleration  $\ddot{y}$  is  $8,400 \text{ ins/sec}^2$  or  $21,330 \text{ cms/sec}^2$ . Thus from formula (V.7), the inertia force on needle 3 =  $\frac{21,330 \times 0.564}{981} = 12.27 \text{ gms.}$

Similarly, the inertia force  $F_I$  may be obtained for each needle acting in the  $57^\circ$  non-linear cam system.

Since values of inertia force may be calculated, the forces acting on the needle may now be resolved again for each portion of the  $57^\circ$  non-linear cam system (Fig. 57.B). Resolving forces horizontally and vertically on the curve portions ab, cd and ef gives the following relationship,

$$S_3 = \frac{(P_F + m\ddot{y})(\sin \theta + \mu \cos \theta)}{(1 - \mu^2) \cos \theta - 2\mu \sin \theta} \dots (V.8)$$

This may be written

$$S_3 = (P_F + m\ddot{y}) \cdot f(\mu, \theta).$$

Resolving forces horizontally and vertically on the curve portions bc, de and fg, gives

$$S_4 = \frac{(P_F - m\ddot{y})(\sin \theta + \mu \cos \theta)}{(1 - \mu^2) \cos \theta - 2\mu \sin \theta} \quad \dots (V.9)$$

This may be written

$$S_4 = (P_F - m\ddot{y}) \cdot f(\mu, \theta).$$

It is noted that  $m\ddot{y}$  is added to  $P_F$  in formula (V.8) as the needle butts are accelerating in curve portions ab, cd and ef, but  $m\ddot{y}$  is subtracted from  $P_F$  in formula (V.9) as the needle butts are decelerating in curve portions bc, de and fg.

The respective acceleration and deceleration portions are given as A and D on the acceleration curve shown in Fig. 55 and referring back to needle 3 (Fig. 55) it may be noted that the needle butt is accelerating. Therefore, from formula (V.8)

$$S_3 = (10 + 12.27)(f(\mu, \theta))$$

where  $f(\mu, \theta)$  is 0.393 when calculated from the cam angle at needle 3, thus providing a value of  $S_3 = 8.754$  gms.

Calculations for a horizontal needle velocity of 200 ft/min at each needle position on the  $57^\circ$  non-linear cam system are shown in Table 9 where values of  $\theta^\circ$ ,  $f(\mu, \theta)$ ,  $P_F$ ,  $m\ddot{y}$ ,  $S_3$  and  $S_4$  are given.

TABLE 9

Needles in 57° Non-Linear Cam (Fig. 55)	$\theta^\circ$	$f(\mu, \theta)$	$P_F$ (gms)	$m\ddot{y}$ 100 r.p.m. (gms)	$S_3$ or $S_4$ (gms)	$g(\mu, \theta)$	$S_5$ (gms)
(a)	1	0	0.10	10	5.18	1.52	
	2	5	0.19	10	10.35	3.87	
	3	15	0.39	10	12.27	8.75	
A	4	30	0.78	10	10.35	15.79	
	5	40	1.14	10	6.77	19.13	
	6	45	1.39	10	1.72	16.29	
(b)	7	50	1.72	10	5.18	8.27	
	8	40	1.14	10	11.40	-	0.73 - 1.03
D	9	30	0.78	10	16.60	-	0.47 - 3.10
	10	10	0.29	10	19.0	-	0.08 - 0.65
(c)	11	10	0.29	10	17.6	8.0	
	12	20	0.51	10	15.2	12.75	
	13	35	0.94	10	13.6	22.20	
A	14	45	1.39	10	10.35	28.30	
	15	55	2.18	10	6.20	35.30	
	16	57	2.40	10	1.72	28.13	
(d)	17	55	2.18	10	3.45	14.28	
	18	55	2.18	10	8.62	3.0	
	19	50	1.72	10	9.80	0.30	
D	20	45	1.39	10	13.8	-	0.89 - 3.4
	21	40	1.14	10	14.5	-	0.73 - 3.3
	22	25	0.63	10	16.25	-	0.36 - 2.27
(e)	K.P. 23	0	0.10	10	16.6	2.69	
	24	10	0.29	10	14.2	7.01	
A	25	30	0.78	10	12.1	17.11	
	26	40	1.14	10	7.26	19.69	
	27	50	1.72	10	2.41	21.34	
(f)							

Continued ...

TABLE 9 (Continued)

Needles in 57° Non-Lin. Cam (Fig. 55)	$\theta^\circ$	$f(\mu, \theta)$	$P_F$ (gms)	$m\ddot{y}$ 100 r.p.m. (gms)	$S_3$ or $S_4$ (gms)	$g(\mu, \theta)$	$S_5$ (gms)
28	45	1.39	10	3.45	9.1	...	
29	40	1.14	10	9.30	0.8		
30	30	0.78	10	13.8	-	0.47	- 1.8
D 31	15	0.39	10	13.8	-	0.17	- 0.63
32 (g)	5	0.19	10	6.9	0.60		

A = Acceleration

D = Deceleration

$\Sigma (S_3 + S_4)$

= 324

$\Sigma S_5$

= 16.2

It may be observed that for certain needles in the decelerating portion (i.e. in curves bc, de and fg), the inertia force  $m\ddot{y}$  is larger than the value of  $P_F$  and this suggests that these needles (i.e. needles 8, 9, 10, 20, 21, 22, 30 and 31) will cross over to the non-leading edges of the cam system and in doing so, the needles will move from one side of the needle trick to the other and start to drive the machine.

For these special needles a new relationship of horizontal force was found (Fig. 57.C)

$$S_5 = \frac{(P_F - m\ddot{y})(\sin \theta - \mu \cos \theta)}{(1 + \mu^2) \cos \theta} \quad \dots (V.10)$$

This may be written as

$$S_5 = (P_F - m\ddot{y}) \cdot g(\mu, \theta)$$

The values of  $g(\mu, \theta)$  and  $S_5$  were calculated for the special cross-over needles and are shown also in Table 9.

The total horizontal force in the direction of needle movement was therefore obtained from  $\Sigma(S_3 + S_4) + \Sigma S_5$  and at the horizontal velocity of 200 ft/min the values for the  $57^\circ$  non-linear were  $324 - 16 = 308$  gms.

This value may be compared with the horizontal force at zero speed, (i.e.  $\Sigma S_1 = 328$  gms), for the  $57^\circ$  non-linear cam (Table 8).

These values show that only a small difference in calculated horizontal force has resulted from increasing the machine speed from 0 to 100 r.p.m. In fact, there is a slight decrease in force, mainly due to the few needles which cross over to the opposite side of the trick and start to drive the machine.

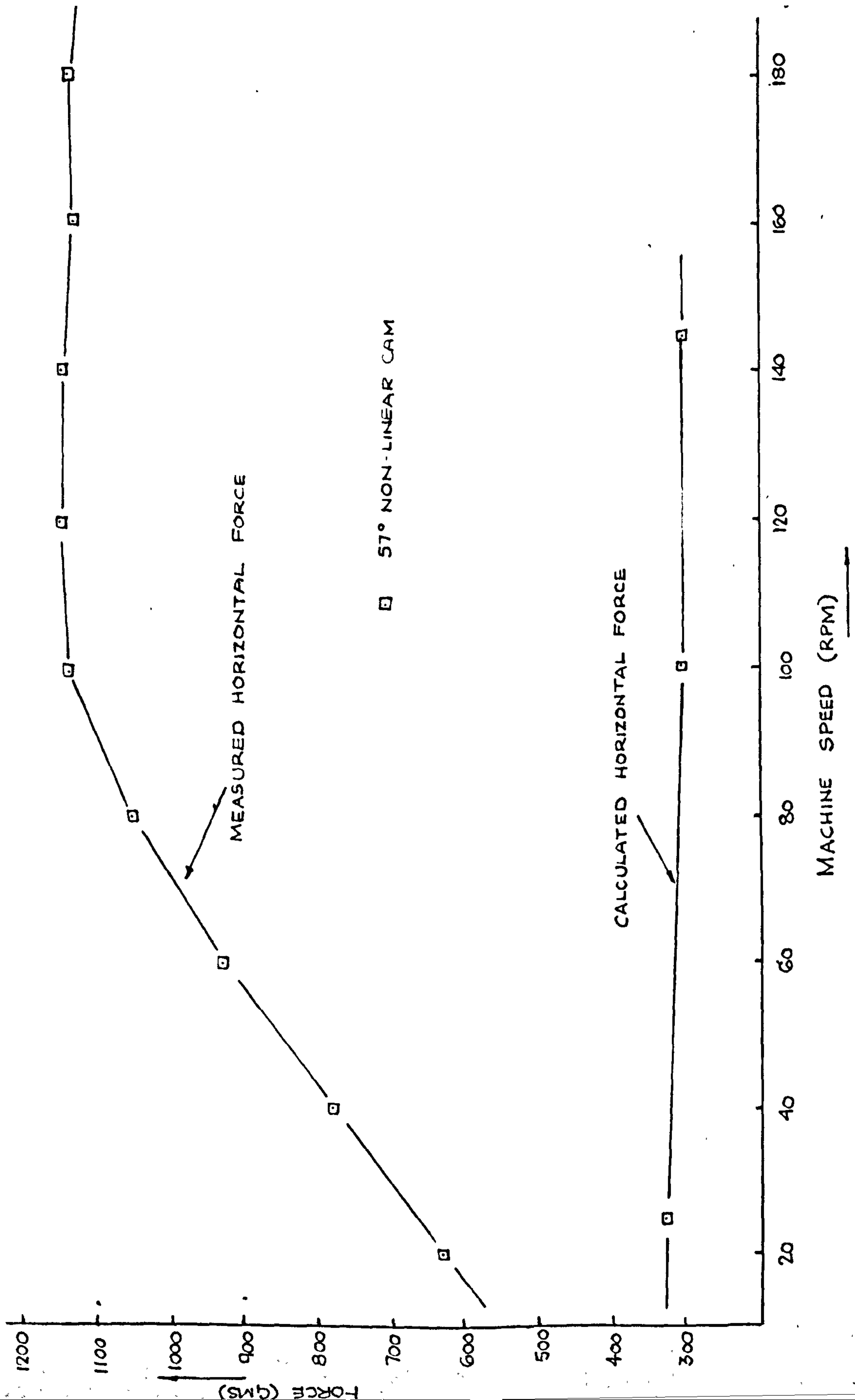
Further calculations of the horizontal forces  $S_3$ ,  $S_4$  and  $S_5$  for the  $57^\circ$  non-linear cam, at different horizontal velocities (V) of 50 ft/min (i.e. 25 r.p.m.) and 300 ft/min (i.e. 145 r.p.m.), also show that there is only a small difference in the  $\Sigma(S_3 + S_4 + S_5)$  compared with the  $\Sigma S_1$  values at zero speed - these values of  $S_3$ ,  $S_4$  and  $S_5$  are given in Table 23\*.

It is interesting to note from the results in Table 23\* that at 50 ft/min no needle butts cross over to the non-leading edges, whereas at 300 ft/min, the cross over needles are more numerous than at 200 ft/min.

Further measurements of force with the  $57^\circ$  non-linear cam for a speed range of 20 - 200 r.p.m. are given in Table 24\* and plotted in Fig. 58. To compare the calculated horizontal forces with these measured values, the calculated

\* See Appendix

7.14 00





values of  $\Sigma(S_3 + S_4 + S_5)$  for the machine speeds of 25;100 and 145 r.p.m. (Table 9 and Table 23) are also plotted in Fig. 58.

Fig. 58 shows that the measured force increases until a machine speed of 120 r.p.m. is attained, (as shown previously in Fig. 56), but remains substantially constant up to the maximum machine speed of 200 r.p.m. The calculated forces  $\Sigma(S_3 + S_4 + S_5)$ , on the other hand, are substantially constant for the calculated speed range from 25 - 145 r.p.m.

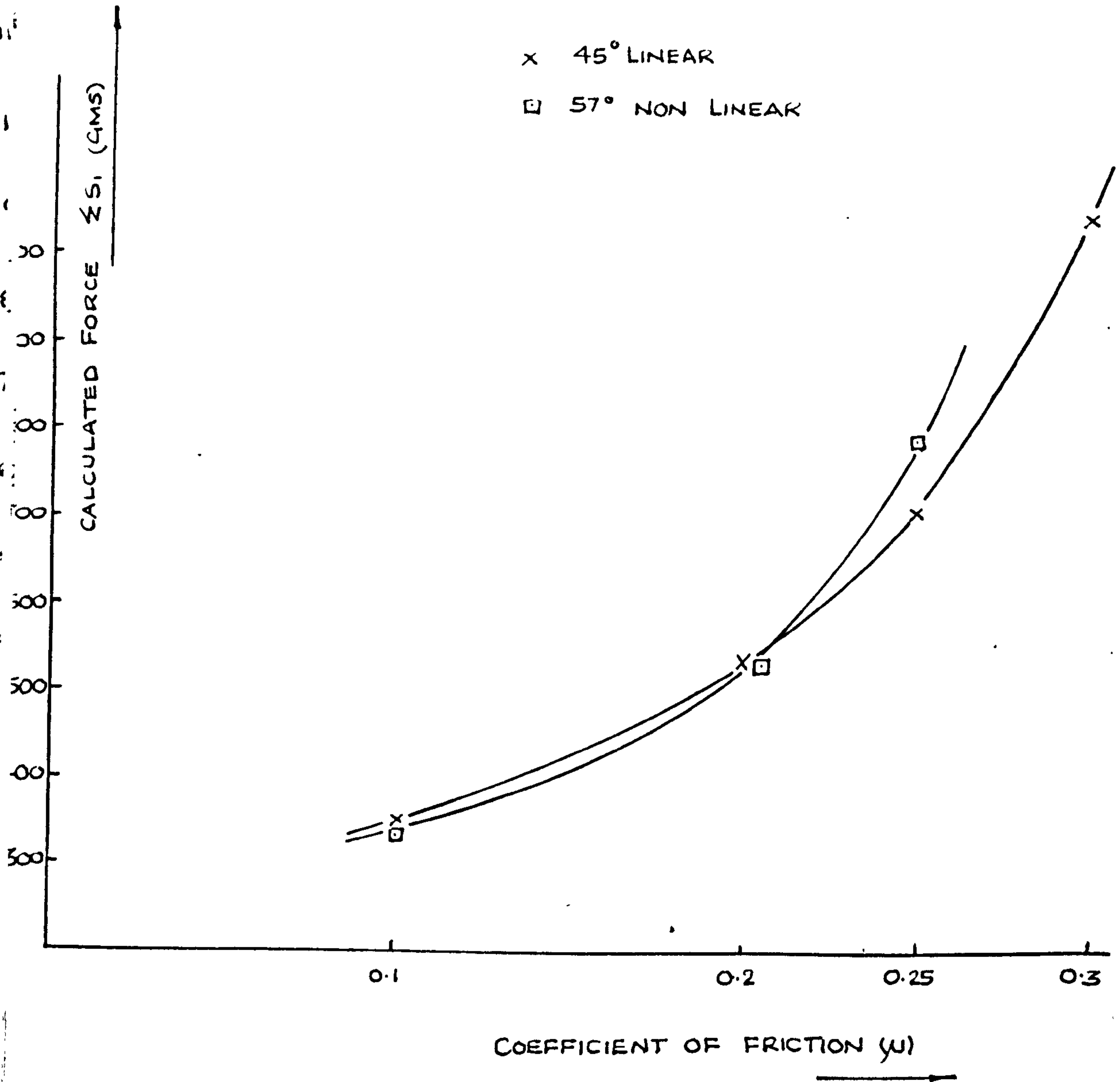
This analysis provides interesting information about the acceleration forces of the needles, and this will be used later to estimate the maximum knitting speeds that might be expected from non-linear cams. However, the analysis indicates clearly that the needle acceleration forces do not contribute to an increase in measured force when the speed is increased.

#### (ii) Friction and lubrication effects

The previous analysis with the non-linear cam has shown that the needle acceleration forces, at low and high speeds, provide virtually no increase in the sum of the calculated horizontal forces and therefore no explanation has been given for the marked increase in measured force, which occurs when the machine speed is increased.

It was suggested, therefore, that the observed force increases with increase in machine speed may be due to either (a) an increase in value of friction ( $\mu$ ), i.e. the friction between the needle butt and cam track and between the needle shank and the trick wall, or (b) an increase in

Fig 59



value of  $P_F$ , i.e. the force required to move the needle vertically in its trick.

(a) Effects of increase in value of friction ( $\mu$ )

At first sight it would appear that an increase in friction with increase in machine speed would offer an explanation of the observed effects since, using equation (V.5) it can be shown that the value of  $\Sigma S_1$  increases with increase in friction ( $\mu$ ) for both linear and non-linear cam systems - see Fig. 59. The value of  $S_1$  calculated for each needle in the  $57^\circ$  non-linear cam, using four different values of  $\mu$ , is given in Table 25.\*

However, a more careful examination of Fig. 53 reveals that the measured force increases cannot easily be related to an increase in friction ( $\mu$ ). The effect of introducing yarn to the needles is to increase the force required to rotate the machine by a constant amount, which does not change appreciably with increase in machine speed for any of the cam systems. It has been stated previously that the additional force, due to the inclusion of yarn in the knitting system, is given by formula (V.6); from this formula an increase in value of friction ( $\mu$ ) with an increase in machine speed would imply an increase in the calculated yarn force ( $\Sigma S_2$ ). It would appear, therefore, that since the measured force component of the yarn does not increase with speed, the increases in measured force cannot simply be explained by an increase in friction ( $\mu$ ).

(b) Effect of increase in value of  $P_F$

Another possible explanation, as suggested earlier, for the force increases with increase in machine speed, is that a change in  $P_F$  occurs as the speed increases. So far it has been assumed that  $P_F$  has a constant value of 10 gms.

FRICITION AND LUBRICATION EFFECTS

FIG 60

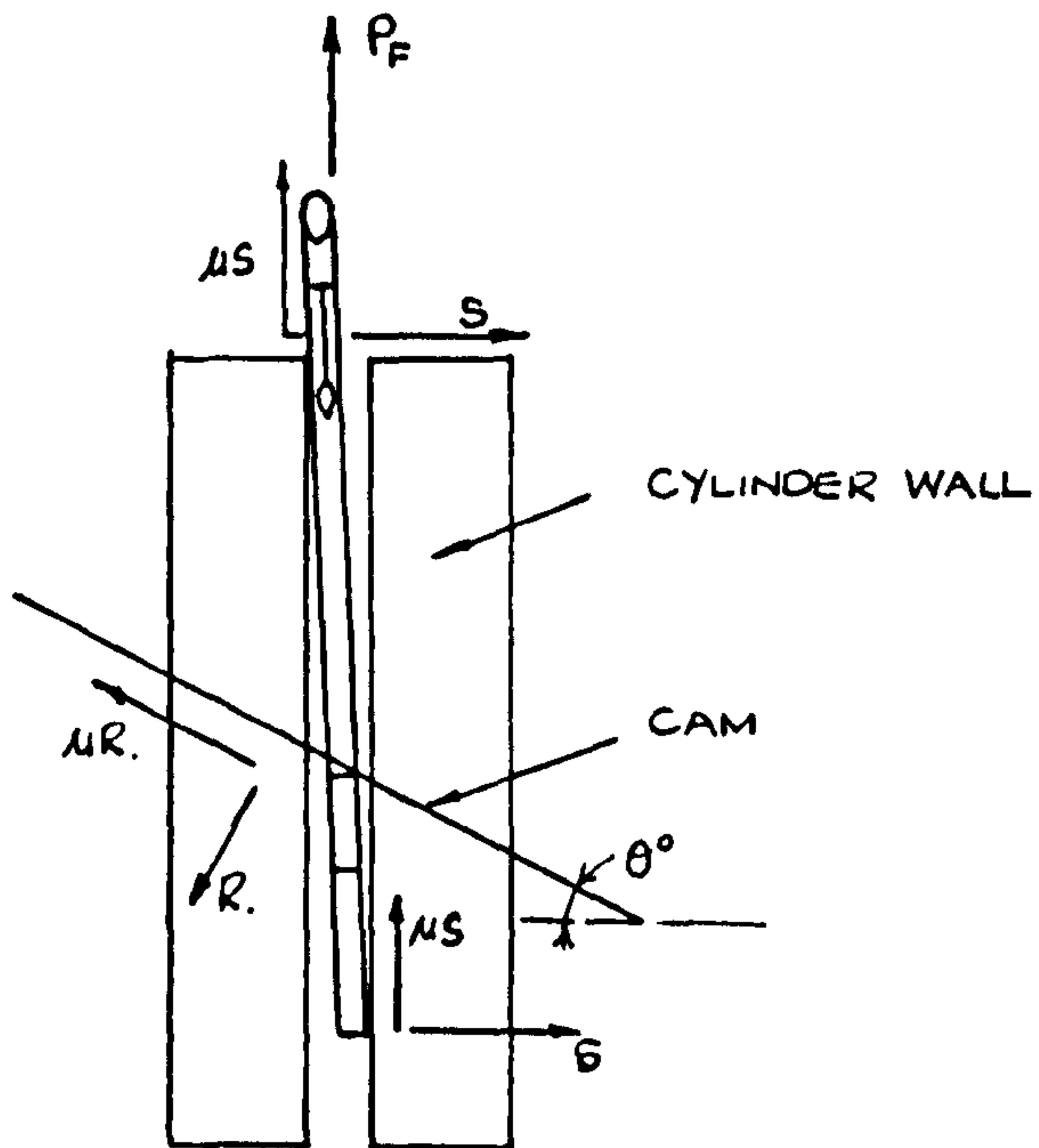
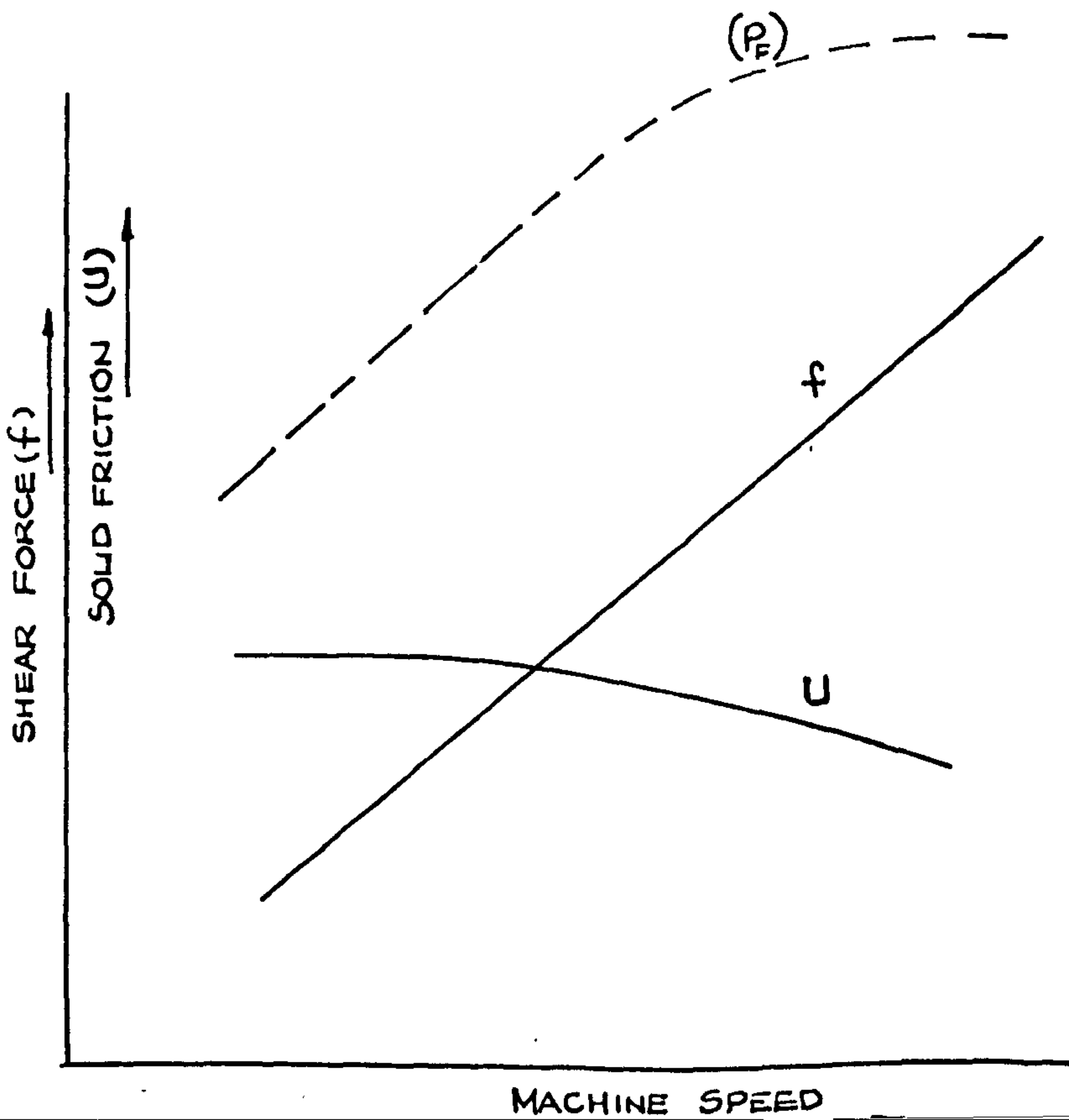


FIG 61



A further examination of the forces acting on the needle (see Fig. 60), shows that the friction forces  $\mu S$  and  $\mu R$  may be considered as resulting from point contact between the needle and the trick, and the needle and the cam face (22). However,  $P_F$  may be thought of as resulting from a contact between two surfaces, namely, the surface of the side of the needle shank and that of the trick. These surfaces are well lubricated by means of a film of oil, and it was decided to consider the effect of speed on this lubrication and therefore on  $P_F$ . It is known that for a Newtonian fluid, the shear force ( $f$ ) needed to overcome the viscous drag, when a mass of fluid is bounded by two parallel planes at a distance 'a', is given by

$$f = \frac{\mu_1 v_1}{a} \quad (V.11)$$

where  $\mu_1$  is the viscosity of the oil;

$v_1$  is the velocity of the oil on the moving surface.

Experiments by Colomb and Poiseuille (23) showed that the shear force was proportional to the velocity and, therefore, the viscosity of the oil was constant with increase in speed. In addition to this, early investigators, such as Towers (24) and Reynolds (25), have suggested that there is no actual metal contact between well lubricated surfaces, since they are separated by a film of liquid lubricant which exerts a pressure which is zero at the boundaries of the area of contact and a maximum at some point in the area. If the surfaces are not parallel, then as the velocity of sliding increases the pressure exerted by the lubricant increases and tends to push the sliding component further away from the fixed portion. Since in practice the metal surfaces

may be considered to be comprised of a series of small bumps and hollows, then this pressure increase, exerted by the lubricant, will reduce the number of contacts of the bumps and give a reduction in the sliding metal/metal friction force ( $U$ ).

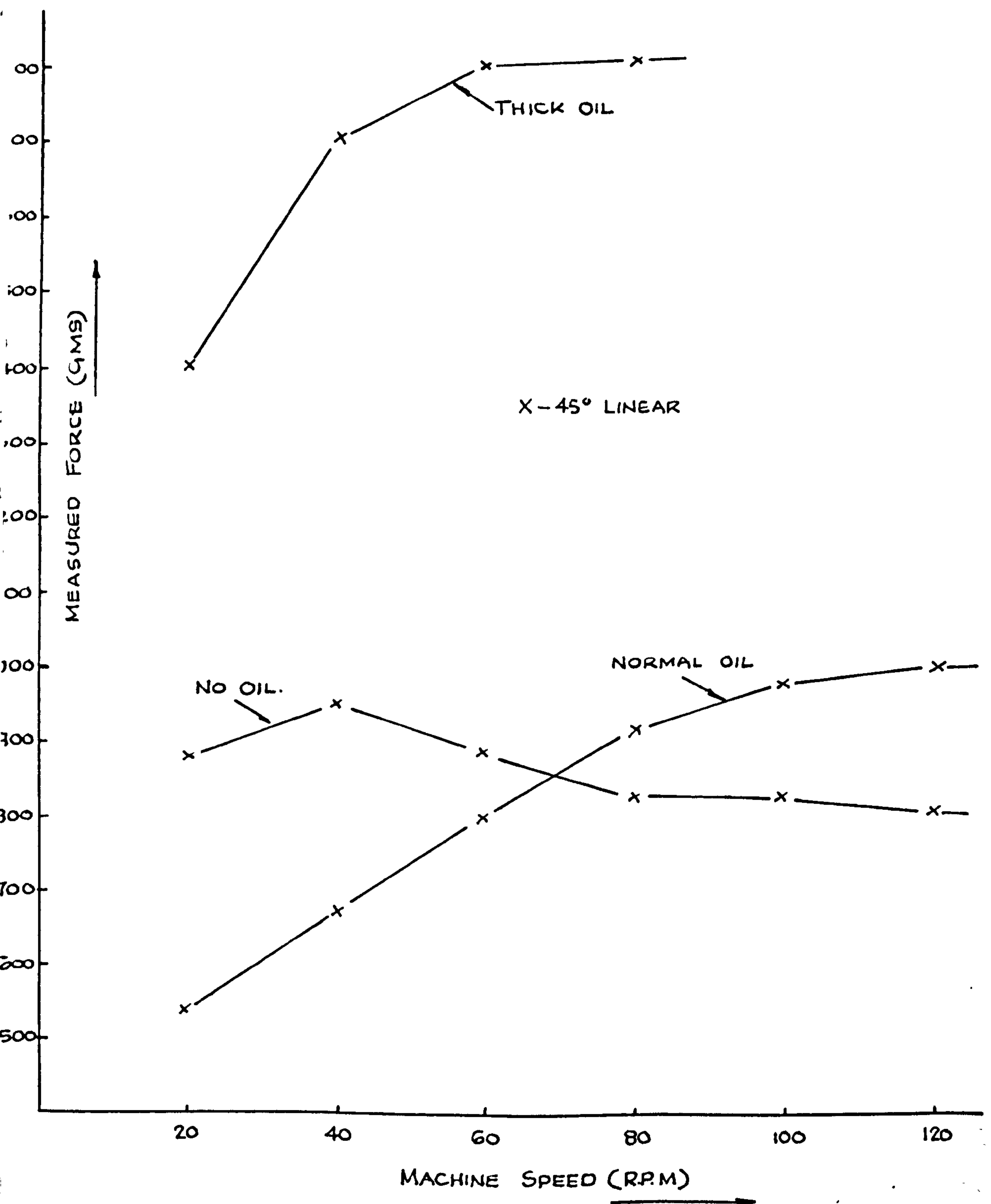
The theories described above, often referred to as boundary hydrodynamic lubrication, may possibly explain the measured force increase with speed, which was observed in this work. For example, Fig. 61 shows a sketch of the possible relationships of the shear force ( $f$ ) against speed needed to overcome viscous drag and the solid friction force ( $U$ ) against speed. It may be seen from Fig. 61 that by adding these two curves, the relationship of resultant force is similar to the measured force values with speed (see Fig. 56).

To establish more precisely whether a condition of boundary hydrodynamic lubrication was responsible for the measured force increases, further experiments were carried out using the force measuring apparatus and the  $45^\circ$  linear cam. It was considered that if a lubrication effect did cause the measured force increases then an experiment without oil between the needle and track should give relatively no change in force, with increase in speed. Also, if a thick oil were used (i.e. an oil with a greater viscosity than normally used) then the force gradient would be steeper than that obtained with the normal oil.

The force values were measured at 20 r.p.m. increments for a speed range of 20 to 120 r.p.m. for three conditions of lubrication.

FIG 62

LUBRICATION EFFECTS



- (1) ordinary lubricant (i.e. Vickers, Trinivol-M) as used for all previous experiments;
- (2) no lubricant at all;
- (3) thick oil (Vickers, Vymox-H).

The force values are given in Table 26\* and plotted in Fig. 62, and it is seen from Fig. 62 that the force values behave as predicted above. It is suggested, therefore, that the addition of the shear force ( $f$ ) needed to overcome the viscous drag, and the metal/metal friction force ( $U$ ) provides an increase in the force  $P_F$  as the machine speed is increased; at speeds above 120 r.p.m. the force  $P_F$  remains substantially constant, probably due to the reduction in metal/metal friction ( $U$ ) as shown in Fig. 61.

It may be concluded, therefore, that the increases in measured force with increase in machine speed, as shown in Fig. 56, are due to an increase in the value of force  $P_F$ , and a calculation from formula (V.5) showed that a value of  $P_F = 30$  gms. gives an agreement with the measured force results for the  $45^\circ$  linear cam, at machine speeds above 120 r.p.m.

4. The horizontal forces acting on the needles with yarn supplied to the needle hooks

(a) Examination of the measured forces

The measured force developed within a cam system when yarn is supplied to the needles comprises :

- (i) the horizontal force to move the needle up or down the cam track without yarn, and
- (ii) the additional component of horizontal force due to the insertion of yarn to the needle hooks.

\* See Appendix



The first of these forces has been examined in detail in the previous section.

To obtain a value of horizontal force due to the insertion of yarn, it was necessary to measure the total force with yarn and subtract this from the force values without yarn.

It has been observed from Fig. 53 that the effect of introducing yarn to the needle hooks increases the force required to rotate the machine by an amount that does not change appreciably with an increase in machine speed, for any cam system.

A value of the additional yarn force component may be obtained, therefore, from any speed. For example, the force values with and without yarn at 60 r.p.m. obtained from each cam system at the slack cam setting, are extracted from Table 22 and are shown again in Table 10.

TABLE 10

Column	Measured Force (gms). 60 r.p.m. Slack setting	Linear		Non-Linear			St. & Curved
		45°	55°	52°	57°	62°	55°
A	With Yarn	1020	1140	1020	1260	1170	1140
B	Without Yarn	780	900	720	930	960	870
C	Force due to Yarn	240	240	300	330	210	270
	Calculated Values (gms) Slack setting						
D	Max. Tensions	70	45	165	125	95	125
E	$\Sigma P_{\theta}$	155	101	-	-	-	-
F	Force due to Yarn $\Sigma S_2$	216	220	222	230	145	230

Subtracting column B from column A in Table 10 gives the additional component of the yarn force (see column C).

The values in column C show that there are only relatively small differences in yarn force between different cam systems. This was surprising, as the calculated maximum yarn tension, shown in Fig. 50 and given in column D of Table 10, are much higher for non-linear cams than for linear cams; a significant difference in the force results might therefore have been expected. This aspect is investigated below.

(b) Calculations of the additional component of horizontal force due to insertion of yarn to the needle hooks

An expression for the additional component of horizontal force, due to the insertion of yarn to the needle hooks has already been given in formula (V.6),

$$\text{i.e. } S_2 = \frac{P_\theta (\sin \theta + \mu \cos \theta)}{(1 - \mu^2) \cos \theta - 2\mu \sin \theta}$$

Knapton (8) suggested, and Wignall (10) showed, that  $P_\theta$  may be obtained from the sum of the calculated tension values acting vertically on the needle hooks during loop-formation. These yarn tension values have been calculated in Section B, (Chapter 5), and the values of  $P_\theta$  for the various cams may be obtained from this analysis (e.g. Table 5 and Fig. 50).

For the linear cam, the total horizontal yarn force is given by

$$\Sigma S_2 = \frac{n(\Sigma P_\theta)(\sin \theta + \mu \cos \theta)}{(1 - \mu^2) \cos \theta - 2\mu \sin \theta}$$

where  $n$  is the number of needles acting on the cam track,

(Table 8), and  $\Sigma P_{\theta}$  is the total yarn tension value acting on the needle hooks. The values of  $\Sigma P_{\theta}$  for the  $45^{\circ}$  and  $55^{\circ}$  linear cam are given in column E, Table 10.

With non-linear cams and the straight and curved cam, where different angles of  $\theta$  are measured for each needle position, it was necessary to obtain a value of  $P_{\theta}$  and hence force  $S_2$ , for each needle in the loop-forming portion of the cam system. For an example, the values of  $P_{\theta}$  from Fig. 50 and the calculated value of  $S_2$  for the individual needles in the  $57^{\circ}$  non-linear cam are given in Table 27\*. The sum of the values of  $S_2$  provide a value for the total horizontal force due to the inserted yarn, i.e.  $\Sigma S_2$ .

The calculated values of horizontal force  $\Sigma S_2$  for all the cam systems are shown in column F (Table 10), and it may be noted that there is no significant difference in force between the various cam systems - these calculated results are therefore in agreement with the practical force measurements (column C, Table 10). However, a comparison of the maximum tension values for the  $45^{\circ}$  and  $55^{\circ}$  linear cams (given in column D, Table 10) show that the  $55^{\circ}$  linear cam has a 65% lower value of maximum tension, and yet the calculated force  $\Sigma S_2$  is virtually the same for both of these cams. Although this suggests that steeper angles with linear cams may be employed without any increase in horizontal force or machine torque, it must be appreciated that the forces on the individual needle butts would be higher under these conditions, and therefore, the steeper angled cam would have greater restrictions in maximum machine speed.

Comparisons of the maximum tensions for the  $55^{\circ}$  linear and  $52^{\circ}$  non-linear cam are surprising, as the  $52^{\circ}$  non-linear cam provides an increase in maximum tension of almost 400%,

\* See Appendix

and yet there is no difference in calculated forces between the cam systems. This is explained by an examination of the loop-forming shapes (Fig. 45), where it is seen that the angles of  $\theta$ , for non-linear cams, at all the needle positions are much lower than the angle of  $\theta$  for the linear cams; the maximum angle in the loop-forming portion of the  $52^\circ$  non-linear cam is  $30^\circ$ , whereas all the needles in the linear cam function at  $55^\circ$ .

This suggests that the much higher maximum tensions developed in non-linear cams do not produce a serious effect on machine performance, as they do not result in much higher total horizontal forces than the forces in linear cams.

It may be noted that the difference between the calculated and measured force is greater for the non-linear cams than for the linear cams (Table 10). This can only be explained by the presence of an additional force of yarn against yarn, since the effect would be greater for non-linear cams due to more knitting elements acting in the loop-forming portion.

Examination of the forces of non-linear cams (Table 10), showed that an increase in cam angle from  $52^\circ$  to  $57^\circ$  provided an increase in both the calculated and measured yarn forces, whereas an increase in cam angle from  $57^\circ$  to  $62^\circ$  provided a decrease in calculated and measured forces. This suggests that the increase in the cam angle  $\theta$  from  $52^\circ$  to  $57^\circ$  had a greater effect on yarn force than the reduction in force due to fewer knitting elements acting in the loop-forming portions. When the cam angle is increased from  $57^\circ$  to  $62^\circ$  the opposite effect occurs.

(c) Calculations of the maximum knitting speeds with non-linear cams

Using the previous analysis of the forces acting on the needle with and without yarn, it was now possible to make an assessment of the maximum knitting speeds obtainable with non-linear cams.

A simple experiment was made to determine the breaking load of a needle butt when the needle was supported in the trick wall. The knitting cylinder was clamped in a horizontal plane and weights were suspended from the needle butt until breakage occurred. It was found that for twelve tests the static load required to break the butt varied from 30 to 45 lbs.

The horizontal force acting upon each of the needles in the stitch portion of the  $57^{\circ}$  non-linear cam may be calculated for machine speeds, for example, of 100 and 3,000 r.p.m., without yarn being supplied to the needles. Using a value of  $P_F$  equal to 30 gms, the values of horizontal force were calculated from equations (V.8), (V.9) and (V.10), and are given in Table 11.

The calculated values for the additional component of horizontal force due to the yarn (i.e.  $S_2$ ) were taken from (Table 27) and are shown also in Table 11. The yarn forces for needles 11 to 19 were determined from the fabric take-down tension (i.e. 2 gms/needle) since these needles were not in the loop-forming zone.

TABLE 11

57° Non-Linear Cam System			
Needle Pos. (Fig. 55)	Calculated Horizontal Force Without Yarn		Calculated Force Due to Yarn $S_2$
	100 r.p.m.	3,000 r.p.m.	
11	13.8	4,880	0.58
12	23.0	7,400	1.02
13	41.0	11,750	1.88
14	56.0	13,750	2.78
15	78.9	13,000	4.36
16	76.0	4,020	4.80
17	57.6	- 4,300	4.36
18	46.6	- 7,300	4.36
19	34.8	- 11,200	3.44
20	22.5	- 11,700	16.70
21	17.7	- 10,100	78.60
22	8.65	- 5,550	129.00
23 (KP)	1.34	1,583	6.6
24	4.58	3,920	2.0

For any needle, the calculated total horizontal force, when yarn is supplied to the needles, is obtained from the sum of the horizontal force due to yarn ( $S_2$ ) and the corresponding horizontal force without yarn.

Examination of Table 11 shows that at 100 r.p.m. which is slightly above the commercial speed for a 8" diameter machine, the maximum horizontal force without yarn occurs at

needle 15, (i.e. the approximate position of the maximum cam angle), whereas the maximum total force with yarn occurs in the loop-forming zone (i.e. needle 22), and has a value of 137.6 gms.

At 3,000 r.p.m. (approximately 30 times commercial speed), the maximum calculated horizontal force with or without yarn occurs at needle 14, and is equal to 13,750 gms. At this very high speed the component of force due to the yarn is negligible compared with the force without yarn, which is dependent upon the machine speed.

This value of calculated maximum horizontal force may be compared with the measured breaking load of the needle butt of 30 lbs. (i.e. 13,620 gms) and the analysis suggests that speeds as high as thirty times commercial speed might be achieved with a non-linear cam before needle butt breakage would occur.

It was realised, however, that imperfections in the manufacture of a non-linear cam would make this speed unobtainable in practice, but the value may be used as a guide to the maximum possible speeds, as far as butt breakage is concerned.

(d) Effect of cam setting on the force values

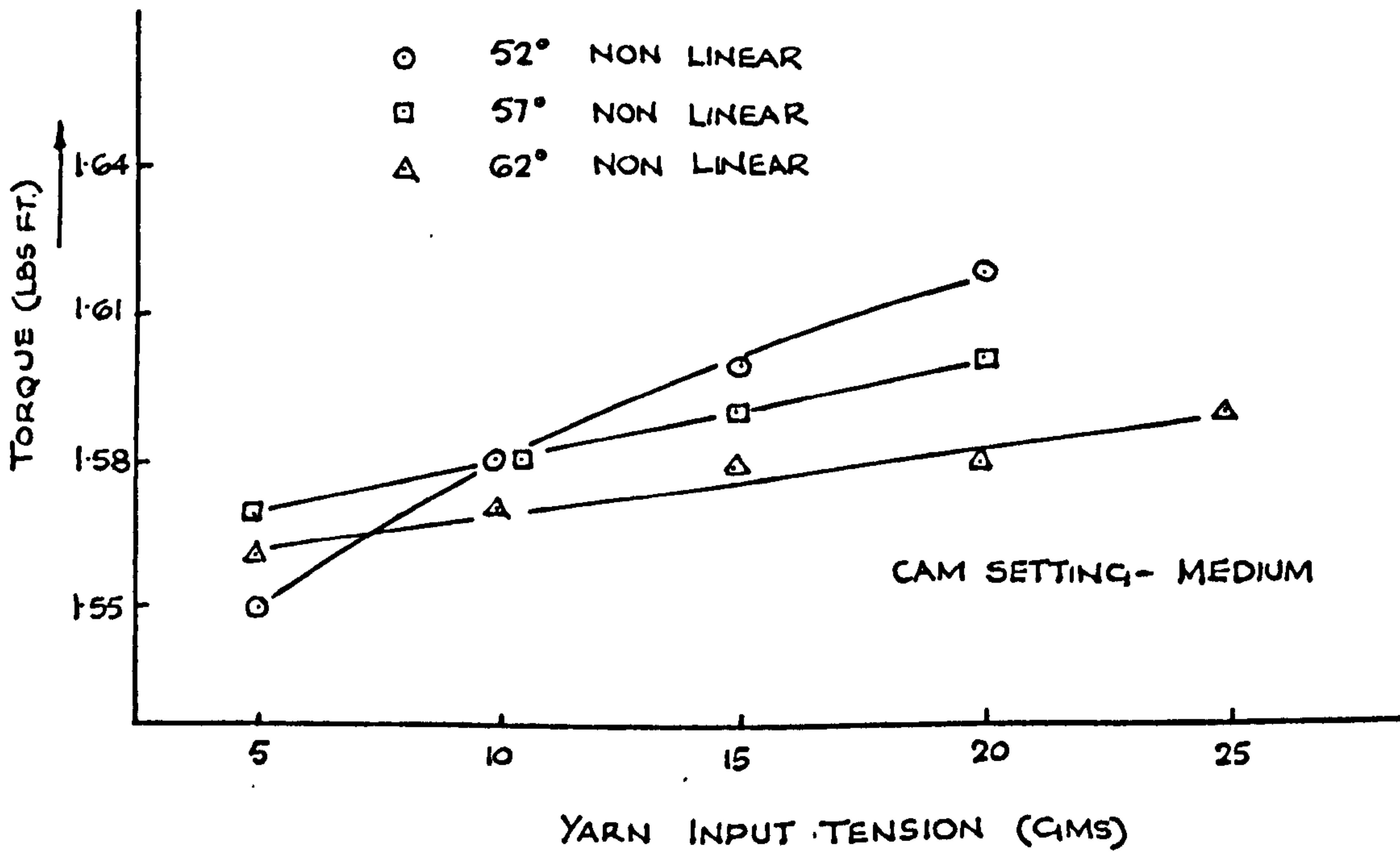
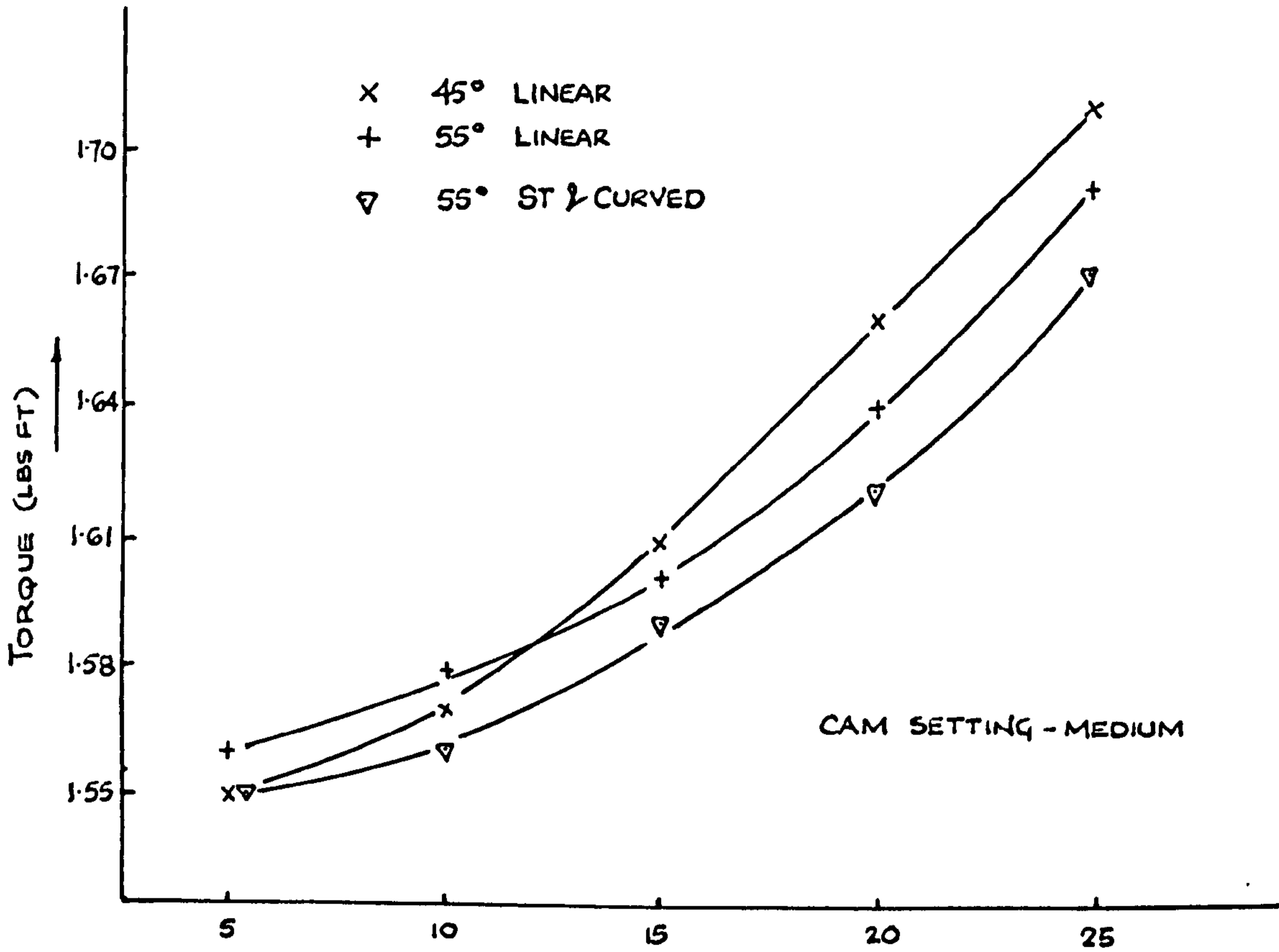
The values of force at the medium and tight cam settings were measured, with yarn supplied to the needles, at a machine speed of 60 r.p.m. These values are shown in column G, Table 12, together with the force values obtained previously for the slack cam setting. The values of force obtained without yarn are shown in Column H, and, by subtracting these values from the values given in column G, the additional component of yarn force is obtained (see column J, Table 12).

TABLE 12

Column	Measured Force gms; 60 r.p.m.	Cam Setting	Cam Systems					
			Linear		Non-Linear			S. & Cu.
			45°	55°	52°	57°	62°	55°
G	With Yarn	Slack	1,020	1,140	1,020	1,260	1,170	1,140
		Medium	1,080	1,190	1,080	1,360	1,250	1,190
		Tight	1,190	1,300	1,190	1,480	1,360	1,300
H	Without Yarn	All Settings	780	900	720	930	960	870
J	Force due to Yarn.	Slack	240	240	300	330	210	270
		Medium	300	290	360	430	290	320
		Tight	410	400	470	550	400	430
	Calculated Values (gms)							
K	Max. Tensions	Slack	70	45	170	130	95	130
		Medium	70	45	170	130	95	130
		Tight	45	30	130	95	95	95
L	Force due to Yarn. $\Sigma S_2$	Slack	216	220	222	230	145	230
		Medium	216	220	222	233	145	233
		Tight	138	146	135	130	145	130



FIG 63



were measured at a machine speed of 60 r.p.m. when the yarn input tension was increased from 5 to 25 gms. The values obtained from the medium setting are shown in Table 28\*, and plotted in Fig. 63.

It was observed from Fig. 63 that the torque values for the linear cams increased significantly when the tension was increased to above 10 gms, and yet only marginal increases in torque were observed with non-linear cams.

The horizontal component of force due to the yarn ( $\Sigma S_2$ ) was calculated from formula (V.6), for yarn input tensions of 5, 10 and 20 gms for the  $57^\circ$  non-linear and  $45^\circ$  linear cams. These values are given in Table 13 and show that the force  $\Sigma S_2$  increases with increase in input tension due to the increase in yarn tension ( $P_0$ ) for both cam systems, but does not account for the higher measured values of torque obtained with the linear cams.

TABLE 13

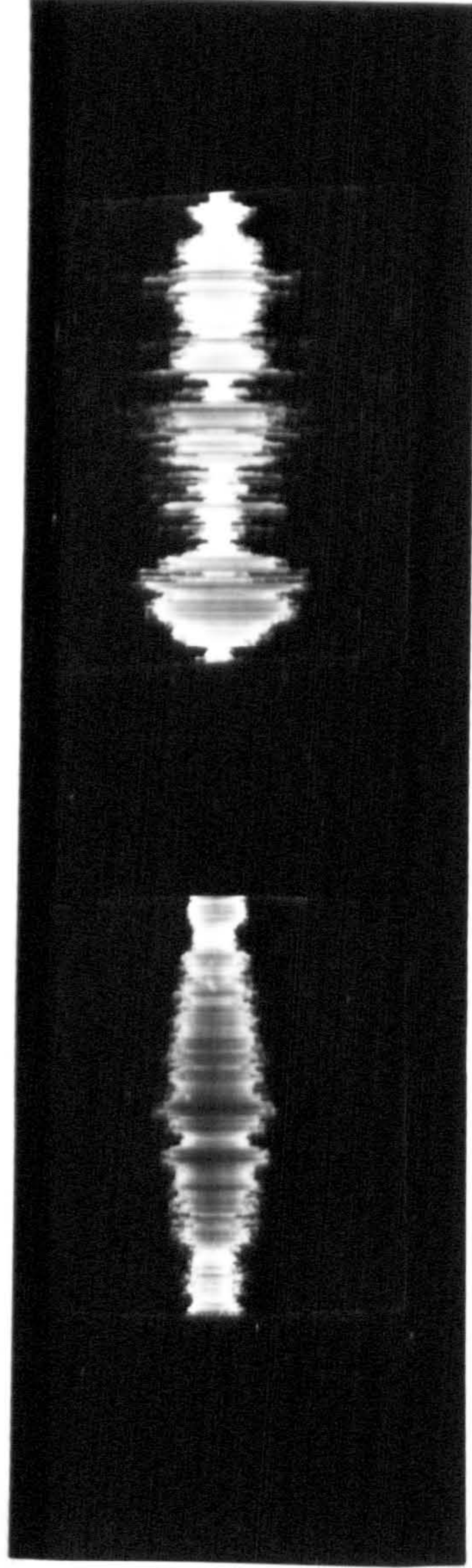
Yarn Input Tension	$\Sigma S_2$ (gms)	
	$57^\circ$ Non-Linear	$45^\circ$ Linear
5	230	216
10	324	300
20	447	399

\* See Appendix

FIG 64

A

B



However, from previous investigations into the effect of yarn tension/stitch length (Fig. 43), it was shown that linear cams provided larger reductions in stitch length as the tension increased, whereas only small changes in stitch length occurred with non-linear cams. It follows, therefore, that it was the more tightly knitted loops that caused the high torque values with the linear cams.

5. Comparison of the Force Results when the Profiles of the non-linear Cam Systems were set with both Correct and Incorrect Needle Butt Clearance

It has been shown in Chapter 2 that an important condition for obtaining continuous acceleration in non-linear cams was that the needle butt must maintain contact with the calculated leading edge of the cam profile throughout the knitting motion. To achieve this it was important that the non-leading edges of the cam profile allowed only a minimum needle butt clearance. It became evident, however, that it was equally important to avoid setting the profile with insufficient clearance.

Fig. 64 shows a film trace taken at a camera film speed of 5 ins/sec of the needle forces acting in the 52° non-linear cam, at a machine speed of 60 r.p.m. Trace A shows the forces when the cam profiles are set with correct clearance, and Trace B when the profiles are set with insufficient clearance.

In some parts of trace B extremely high forces are apparent and a closer examination of the cam profile showed that at one or two positions the needle butts tended to jam, due to the tightness of the cam. Some of the needles -

FIG G5 A

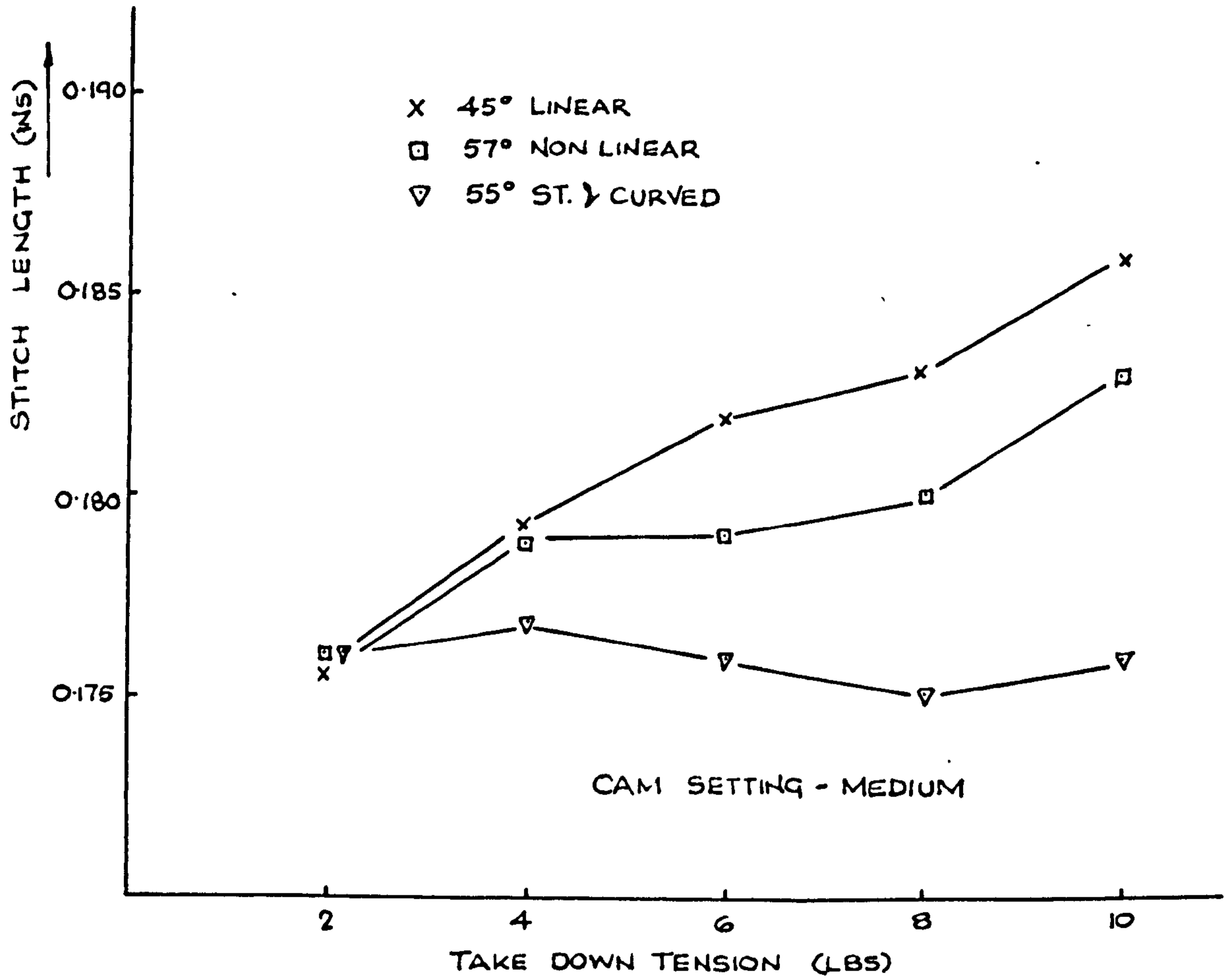
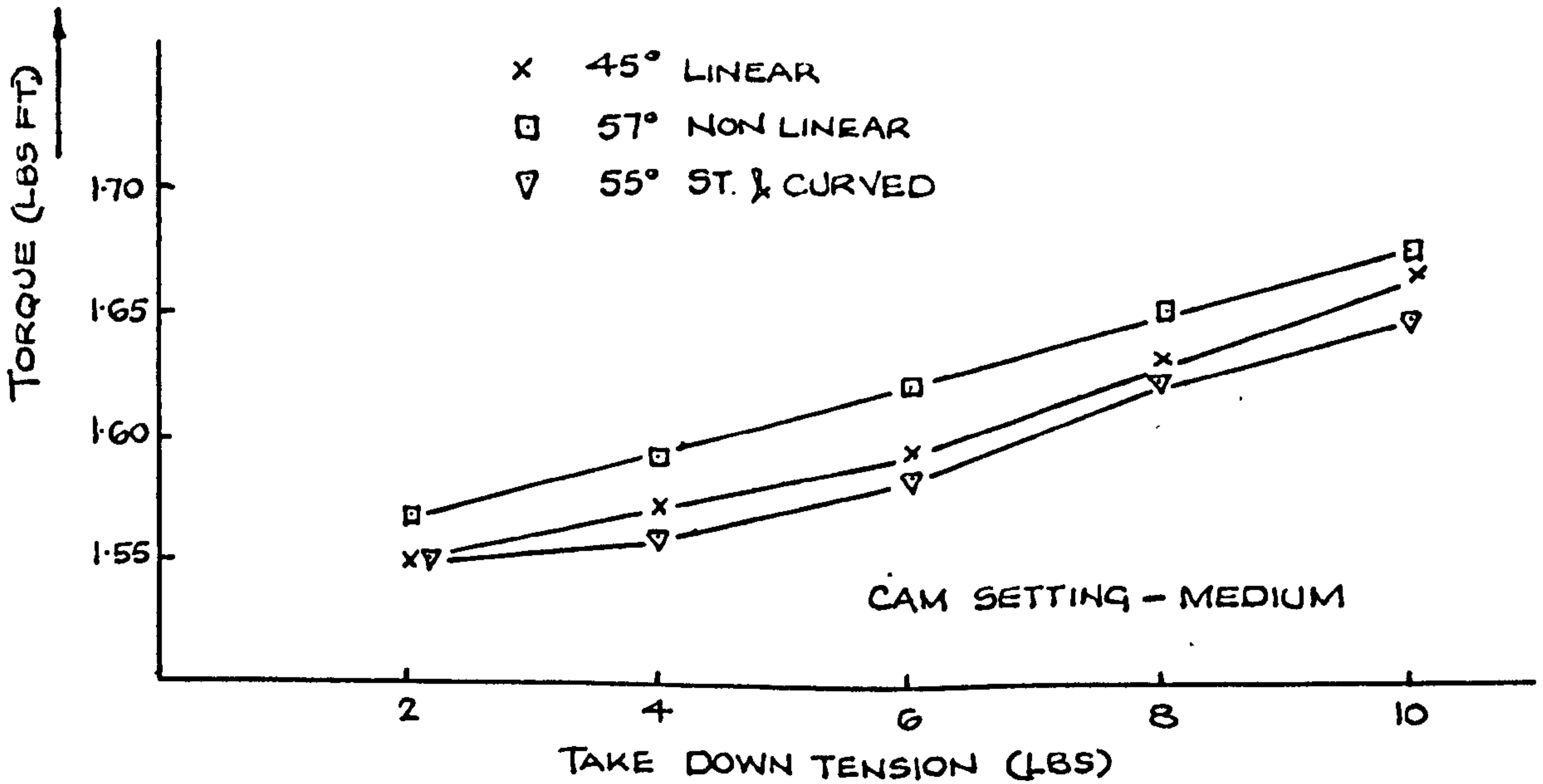


FIG G5 B



presumably those with slightly thicker butts - were subjected to three or four times the force of the other needles when passing through the tight positions; this high force would eventually lead to butt breakage.

It is interesting to note that no difference in fabric appearance occurred, and no needle butts actually broke during a series of experiments at this cam setting; without the strain gauge apparatus this high force effect would, therefore, have gone undetected. This experiment illustrated how useful this measurement of cam force would be for machine builders to test the cam systems on new machinery.

D. Comparison of the Knitting Performance of the Various Cam Systems

1. Effect of Take-Down Tension on Stitch Length and Torque

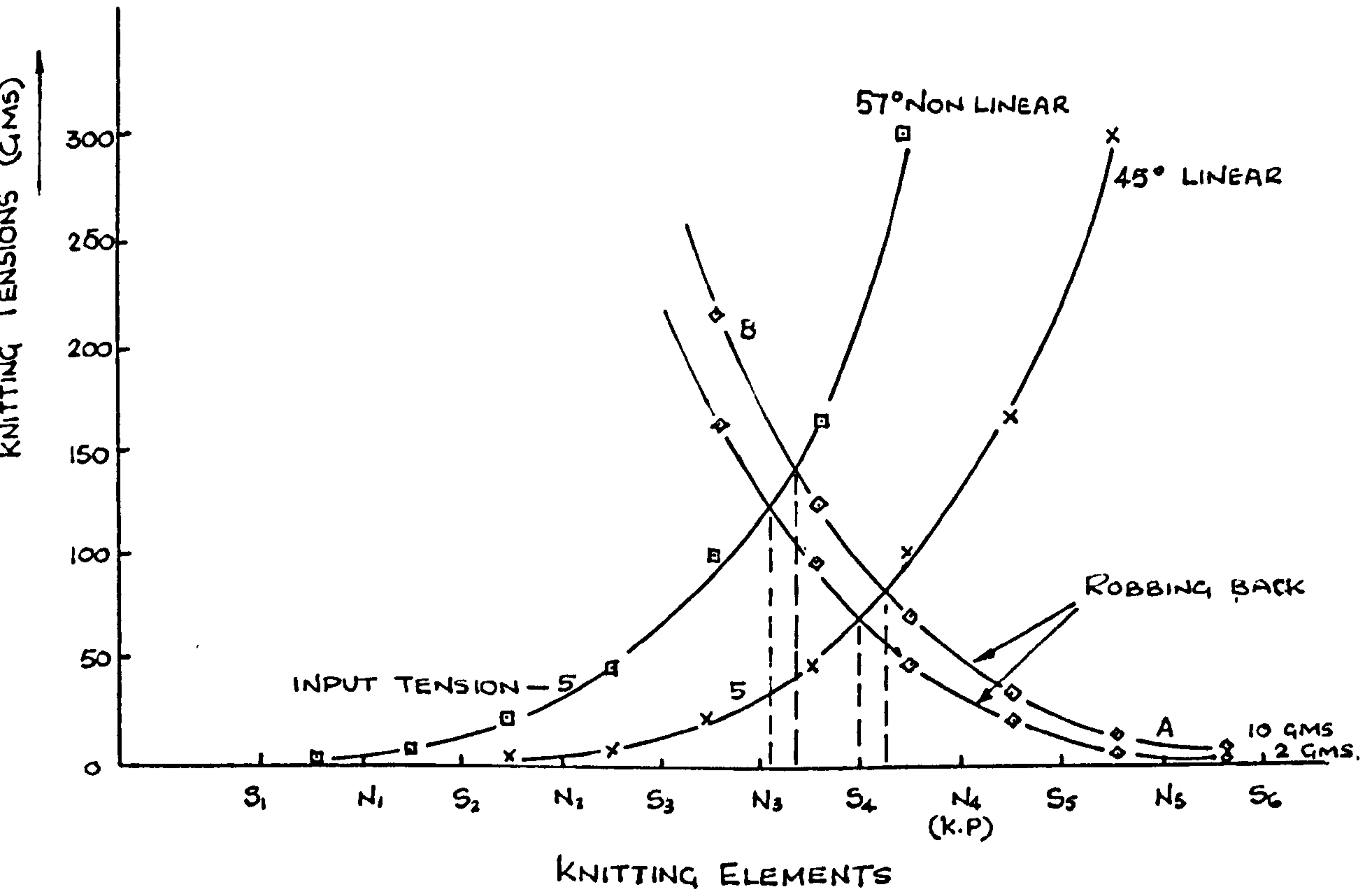
Each cam system was set to provide slack, medium and tight stitch lengths, and the take-down tension was varied from 2 - 10 lbs in increments of 2 lbs. The values of stitch length and torque were recorded for each test, and the results for the medium setting are shown in Table 29;\* the results from the other cam settings are not shown as they gave similar results. The relationship of stitch length/take-down tension for the 45° linear, 57° non-linear and the 55° straight and curved cam, is shown in Fig. 65A.

An increase in stitch length of approximately 5% occurs for an increase in take-down tension of 8 lbs. for the 45° linear and 57° non-linear cam, whereas no increase in stitch length is observed for the 55° straight and curved cam (see Fig. 65A).

\* See Appendix

FIG 66

EFFECT OF INCREASING TAKE DOWN TENSION ON THE  
POSITIONS OF MAXIMUM TENSION



✓ The increases in stitch length are explained easily, as an increase in take-down tension will increase the loop tensions on the right-hand side of the knitting point, and this will restrict the robbing back movement of the yarn. Therefore a needle descending to the knitting point will find it easier to draw extra yarn from the package, and a larger stitch length will occur. This explanation may be related to the diagrams of tension building up and positions of maximum tension (see Fig. 66). An increase in take-down tension of 8 lbs will provide higher values of tension in the robbing back curve (see curve AB), and therefore the position of maximum tension will be displaced nearer to the knitting point - thus the calculated stitch length will increase and may be related to the increases in measured stitch length, as shown in Table 14.

TABLE 14

Cam System	Take-Down Tension (lbs)	Measured Stitch Length (ins)	Calculated Stitch Length (ins)
45° Linear	2	0.176	0.178
	10	0.186	0.189
57° Non-Linear	2	0.176	0.179
	10	0.184	0.188

The values of calculated stitch length when the take-down tension is 10 lbs are 0.189 and 0.188 for the 45° linear and 57° non-linear cam respectively, and these values are in close agreement with the measured values at 10 lbs. For both the



calculated and measured stitch lengths, the  $57^{\circ}$  non-linear provided less increase in stitch length for a given increase in take-down than the  $45^{\circ}$  linear cam. This is explained by the difference in shape of the linear and non-linear cams.

Since the  $55^{\circ}$  straight and curved cam has an unrestricted area for free movement after the knitting point, it was realised that an increase in take-down tension would tend to pull the upthrow needle vertically, until a point was reached where the tension in the loop of this needle was less than the downward frictional forces of the needle in its track. Therefore an increase in take-down tension would give greater vertical displacement of the needle, but would not affect robbing-back, since the initial values of tension for this curve would always remain at the same low value.

The results of machine torque/take-down tension obtained from the torque meter are shown in Fig. 65B and it is noted that approximately 6% increase in torque occurs for an increase in take-down of 8 lbs. Further measurements using the strain gauge force device, which measures only the needle forces, showed virtually no increase in force with an increase in take-down tension; this suggests that most of the increase in torque is due to the loading of the bearing system by the additional take-down weight.

## 2. Effect of Yarn Count on Stitch Length and Torque

Yarn counts ranging from 1/32 to 2/24 worsted were used for knitting fabrics with all cam systems at slack, medium and tight settings. Torque measurements were recorded, and stitch length values were measured from each sample fabric; only the values from the slack setting are shown, as this setting allowed the largest range of yarn counts to be used (see Table 30)\*

\* See Appendix

FIG 67 A

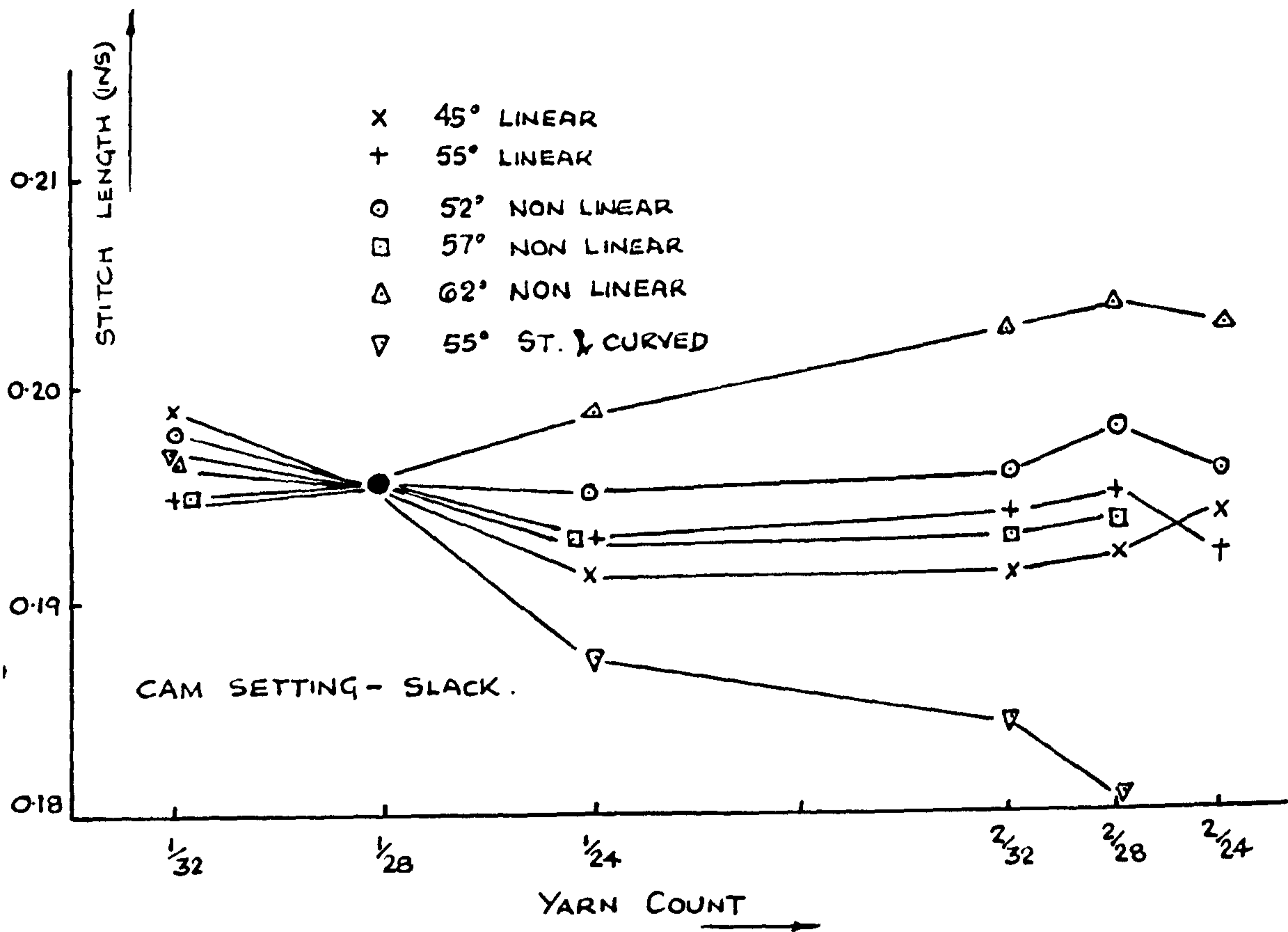
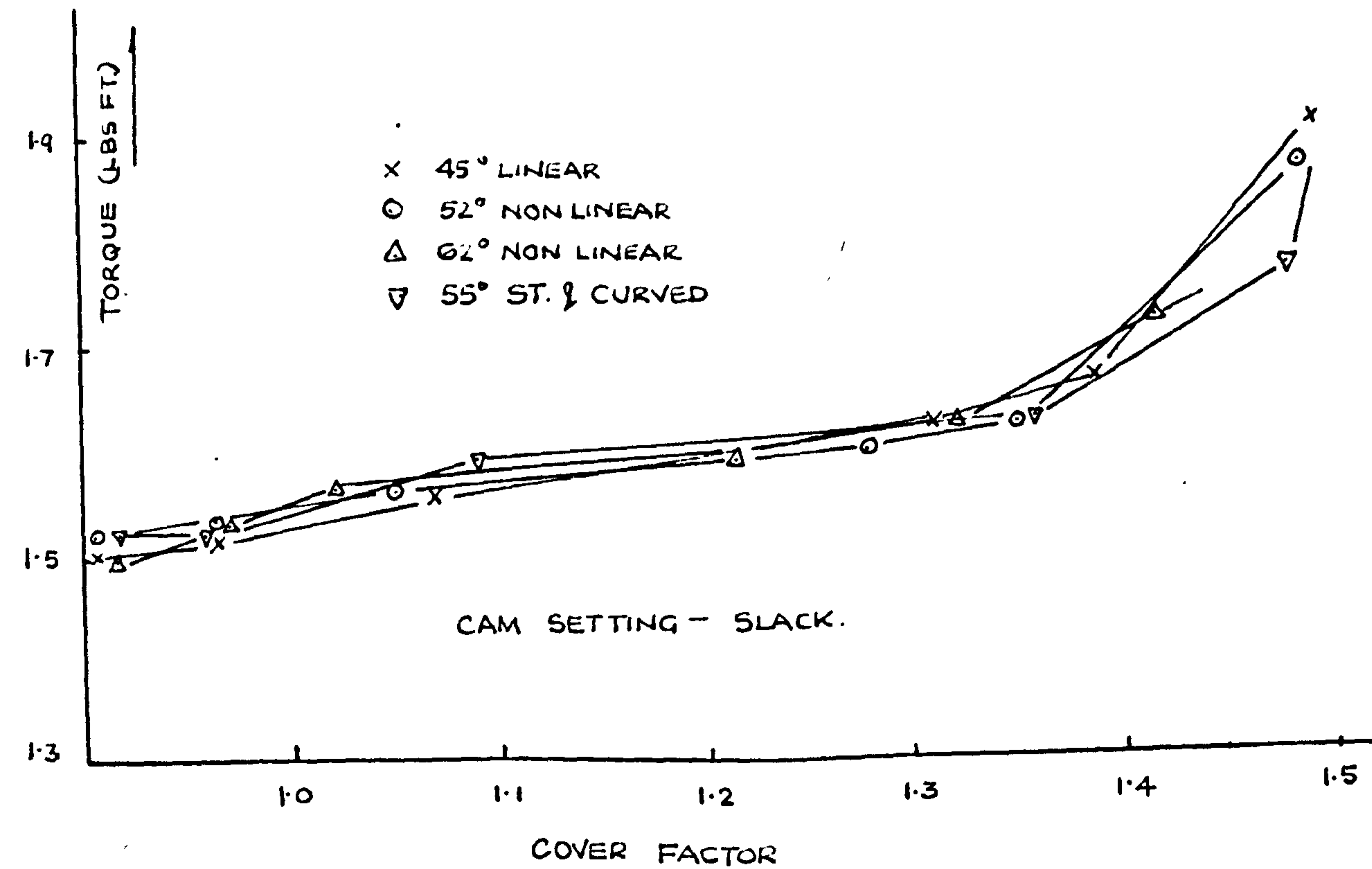


FIG 67 B



From the relationship of stitch length/yarn count (Fig. 67A), it was observed that with an increase in yarn count from 1/32 to 2/24, the stitch length decreased by 9% for the 55° straight and curved, but remained reasonably constant for the other cams, although the 62° non-linear provided slight increases in stitch length. These results may be explained as follows.

It is to be expected that an increase in yarn count would result in increases in tension build-up, and therefore higher maximum tensions, due to the greater angles of yarn/metal contact and yarn/yarn contact. With most of the cam systems the increases in tensions would be similar for both the input tension side and robbing back side, and therefore no changes in robbing-back, and hence stitch length, would occur. With the 55° straight and curved cam, however, the input tension side would produce a greater increase in tension build-up than the robbing-back side, since the needles in this cam are not controlled by the cam system after the knitting point, and are free to rise to the required height to allow immediate robbing-back. With this cam, therefore, an increase in yarn diameter would result in more robbing which would give shorter knitted stitch length values.

Fabric knitted from a 2/28 worsted yarn with the 52° and 57° non-linear cams, exhibited many small holes, and the yarn was extremely weak and therefore difficult to unrove. This illustrated clearly that the values of maximum tension were in excess of the breaking load of the yarn, and it suggested that the range of yarn counts that may be knitted with linear cams would be reduced when using non-linear cams.

For examinations of torque and yarn count, it is more useful to plot torque against cover factor (c.f.) - see Table 30\* and Fig. 67B -

$$\text{where c.f.} = \frac{1}{l\sqrt{N_w}}$$

$l$  = measured stitch length (ins)

$N_w$  = worsted count.

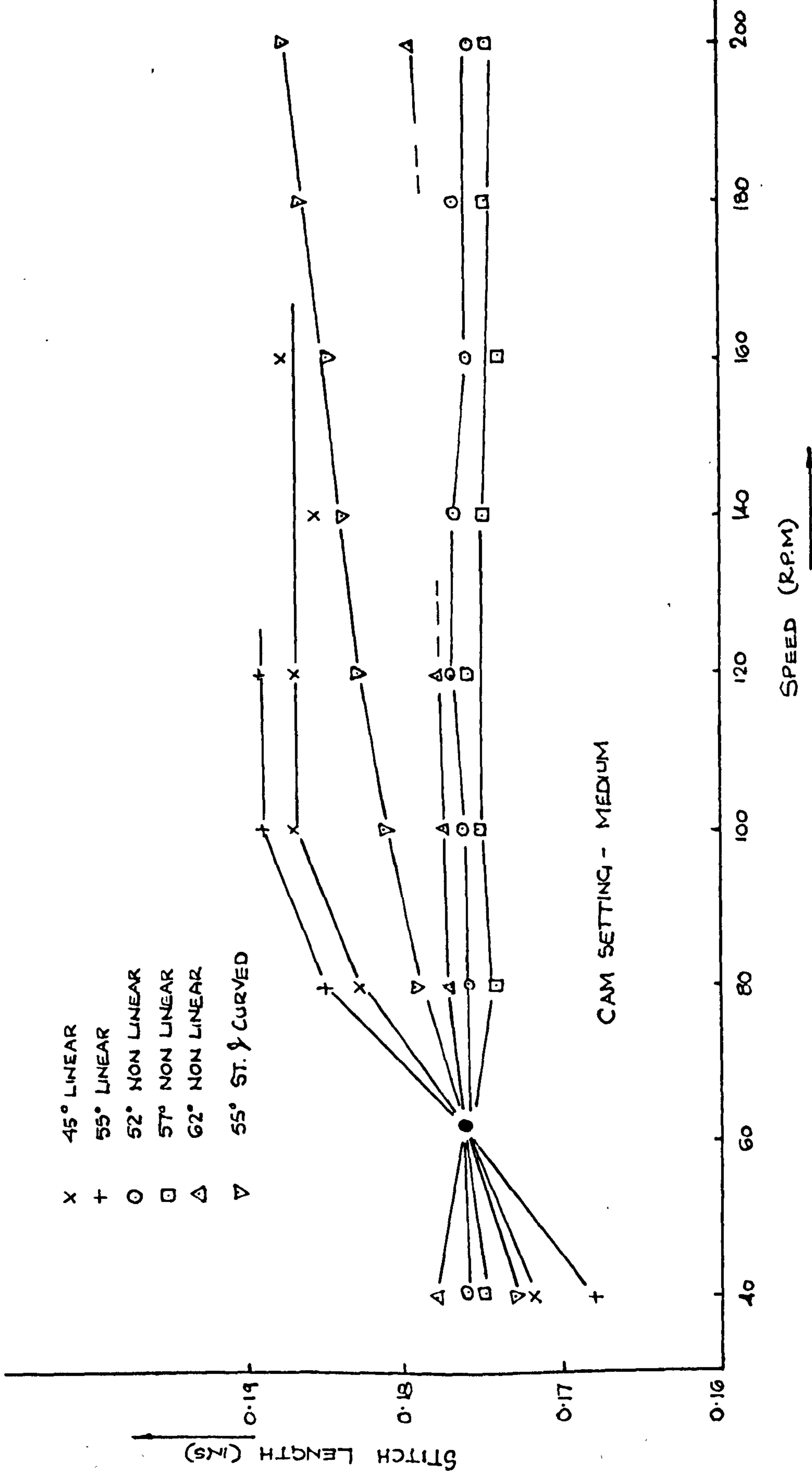
To avoid confusion only the values of torque and cover factor for four cams are shown in Fig. 67B. From Fig. 67B, an increase of only 7% machine torque was observed for an increase in cover factor of from 0.90 to 1.35; however, a substantial increase in torque occurred for all cam systems when the cover factor increased from 1.35 to 1.50. This relationship suggests that the limiting cover factor, for reasonable machine performance, is 1.35 as above this, extremely high torque values occur and irregular breakage of needle butts occurred. The results of torque and cover factor are in close agreement with the observations made by Wignall (10).

### 3. Effect of Machine Speed on Knitting Performance

Each cam system was set to the medium cam setting, and fabric was knitted at machine speeds ranging from 40 to 200 r.p.m. A machine speed of 200 r.p.m. which is approximately three times the commercial speed for this 8" diameter machine, was the maximum speed that could be obtained on this modified commercial knitting machine. For these experiments the machine speed was increased in 20 r.p.m. increments, and each speed change was maintained for the maximum time permitted by the gravity take-down unit.

\* See Appendix

FIG 60



- X 45° LINEAR
- + 55° LINEAR
- O 52° NON LINEAR
- 57° NON LINEAR
- △ 62° NON LINEAR
- ▽ 55° ST. & CURVED

The following information was obtained in order to assess the performance of the various cam systems :

- (a) A comparison of stitch length value at different machine speeds.
- (b) A comparison of fabric appearance at speeds above commercial speeds.
- (c) The maximum knitting speeds prior to needle butt breakage.
- (d) A comparison of the circumferential lengths of the six cam systems.

(a) Comparison of stitch length values

The stitch length values are shown in Table 31\*, and stitch length/speed is plotted in Fig. 68.

It may be seen from Fig. 68 that the effect of speed on stitch length is completely different for linear and non-linear cam systems. With linear cams the stitch length increases up to a speed of 100 r.p.m. and remains reasonably constant until the maximum speed is reached. The 55° straight and curved cam provides a slower rate of increase in stitch length, and the stitch length continues to increase up to the maximum speed of 200 r.p.m. With non-linear cams, however, it is interesting to note that virtually no change in stitch length occurs between 40 and 200 r.p.m.

Examination of the knitting point areas of the linear cam systems showed that there was sufficient clearance to allow the needle butt to move freely in either upwards, downwards or horizontal directions, for a distance

\* See Appendix

equivalent to the needle spacing, whereas with the non-linear cam systems the needle butts were positively controlled throughout their movement.

With linear cams functioning at speeds between 40 and 100 r.p.m. it was considered that the forces of the yarn were sufficient to overcome the frictional forces and the acceleration forces of the needle butts at the knitting point; this consequently caused the needle to rise before it reached the upthrow cam, thus allowing robbing-back to occur immediately - this was confirmed by observing the needle butts through a cut-away portion of the cam section using a travelling microscope. As the speed increased up to a 100 r.p.m. the acceleration forces increased and the needle butt did not rise as much, therefore less yarn was available for robbing-back and the stitch length increased.

At speeds between 100 to 200 r.p.m. it was considered that the acceleration force of the needle was sufficient to exceed the yarn forces, and the needle butt moved horizontally (and even downwards at the highest speeds), until it reached the upthrow cam. The upthrow cam lifted the needle and provided the same robbing-back values, and therefore a constant stitch length at the higher speed range.

It is evident from the above analysis why a non-linear cam system provides no change in stitch length when the machine speed is increased - the movement of the needle butt is predicted by the cam system and will therefore always rise to the same height after the knitting point, irrespective of needle butt speed, and hence provide the same percentage robbing-back.

(b) Comparison of fabric appearance produced at high speeds

Examination of the fabric produced at high speeds showed that linear cam systems provided a distorted fabric with irregular loop sizes and occasional holes from speeds above 140 r.p.m.; non-linear cams and the 55° straight and curved cam, however, gave a good quality fabric at the maximum speed of 200 r.p.m. (see Fig. 69). This difference in fabric appearance must be attributed to different needle butt control, and it was considered that with linear cams the needle fling from the upthrow cam would be quite large at higher speeds, and vary from needle to needle depending on the stiffness of the needle in the trick walls. More needle fling will occur with slack needles, and therefore variations in robbing-back and hence loop size, will occur from one needle to the next.

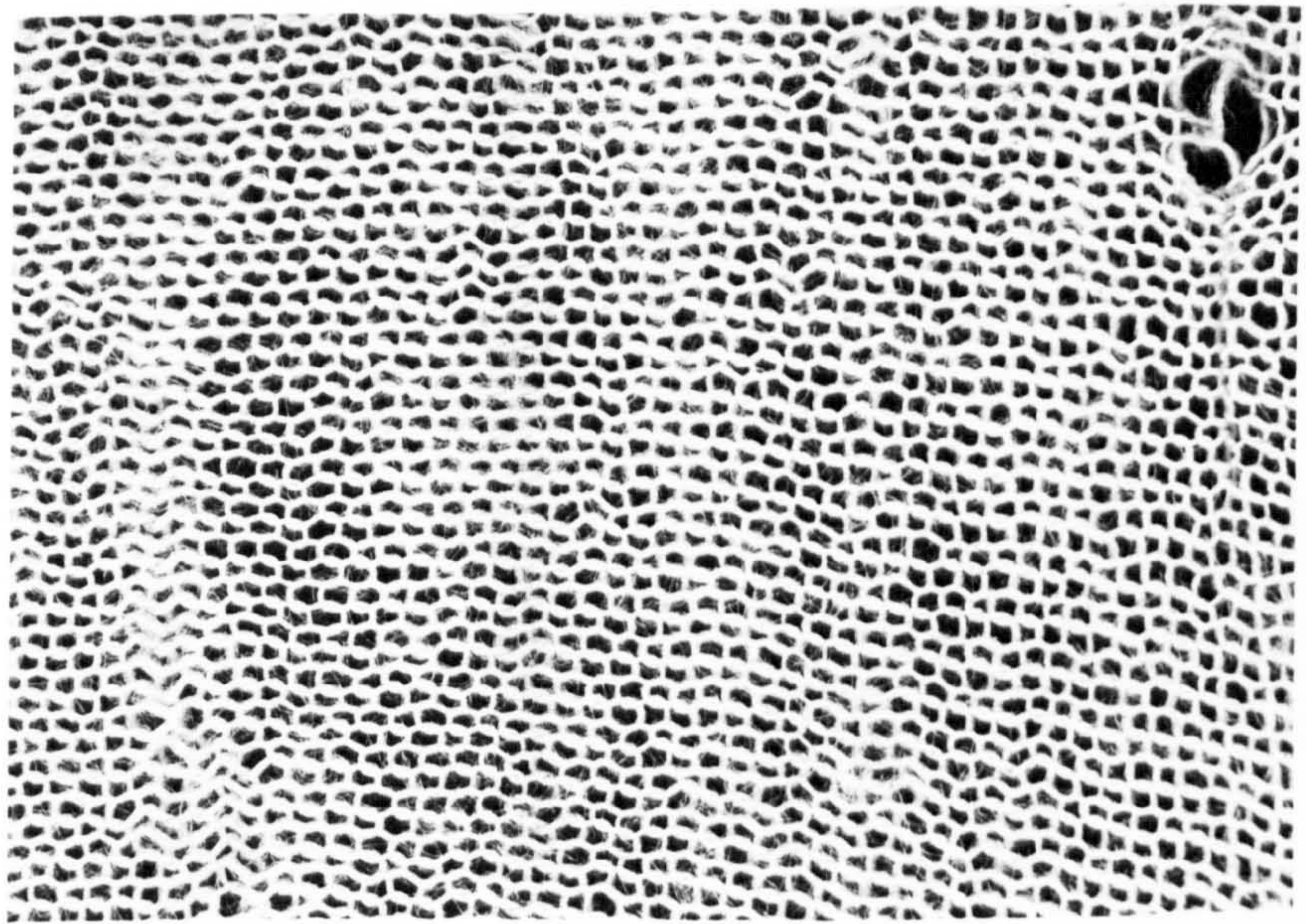
From the examination of fabric appearance at high speeds, it would appear that non-linear cams were far more suitable for high speed knitting than linear cams, and the limitations of maximum speed with linear cams may be determined by the appearance of the fabric.

(c) Comparisons of maximum knitting speeds

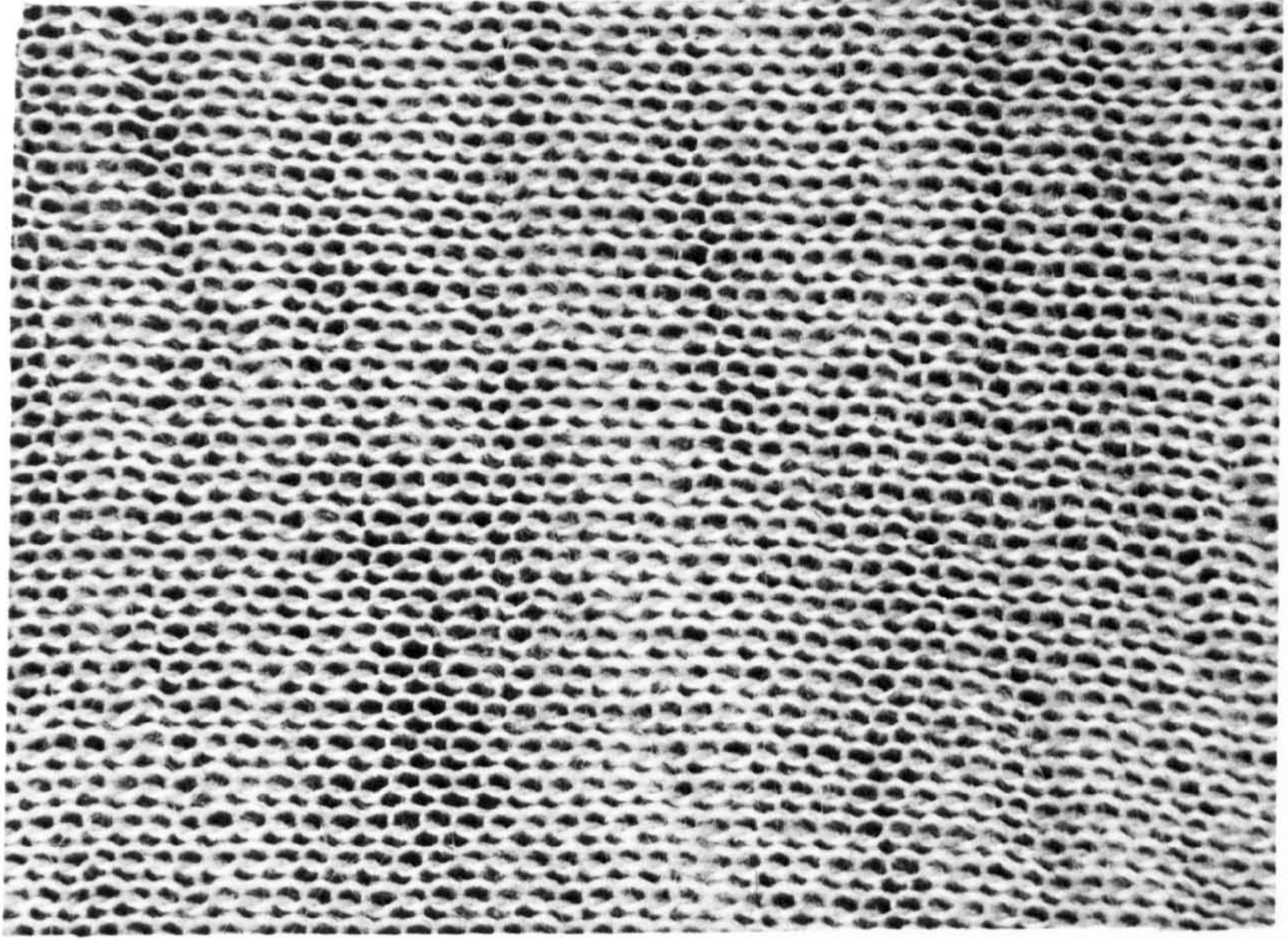
The maximum knitting speeds prior to needle butt breakage were noted during the speed trials and the following information was obtained :

- (1) The needle butts in the 45° and 55° linear cams gave satisfactory performance up to speeds of 160 and 120 r.p.m. respectively, but above these speeds an occasional needle butt broke.





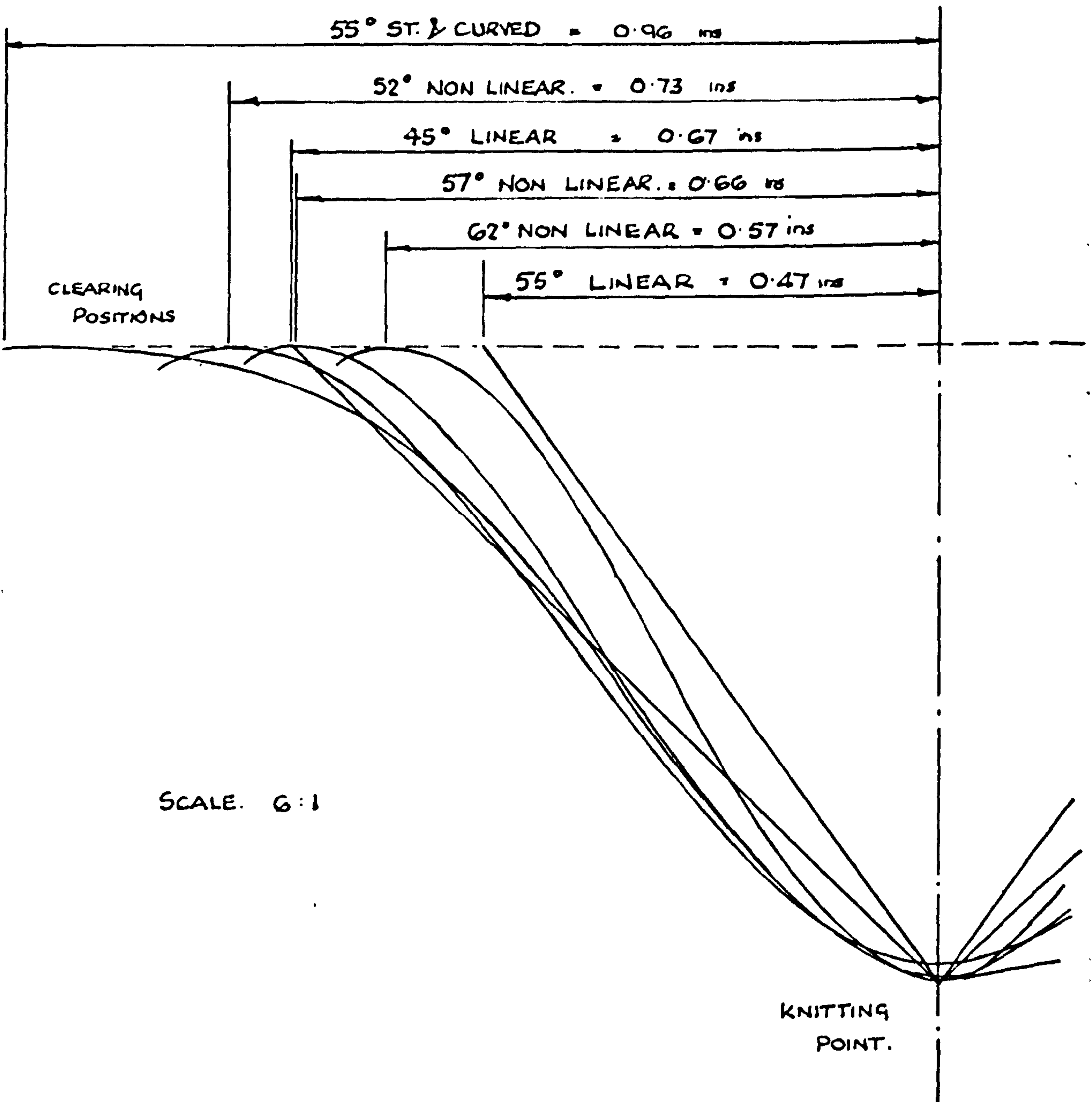
FABRIC KNITTED AT 140 R.P.M.  
WITH THE 55° LINEAR CAM



FABRIC KNITTED AT 200 R.P.M.  
WITH THE 57° NON-LINEAR CAM

FIG 70

DIAGRAM OF THE STITCH CAM PORTIONS ILLUSTRATING  
HALF THE CIRCUMFERENTIAL LENGTH



SCALE. 6:1

- (ii) The needle butts in the  $62^{\circ}$  non-linear cam provided satisfactory performance when continuously running at a speed of 200 r.p.m. but an occasional needle butt broke when starting and stopping the machine. This suggested that the maximum angle of a knitting cam system is ultimately determined by the high forces introduced when starting and stopping the needles, rather than the forces involved in continuous running at high speeds.
- (iii) With the  $52^{\circ}$  and  $57^{\circ}$  non-linear cam and the  $55^{\circ}$  straight and curved cam, satisfactory needle butt performance was achieved at the maximum speed at which it was possible to rotate the machine (i.e. a machine speed of 200 r.p.m. which is three times commercial speed) and no butt breakage occurred when starting and stopping the machine.

(d) Comparisons of Circumferential Lengths

To assess the overall rates of fabric production for each cam system, it was necessary to compare the circumferential lengths.

To illustrate the circumferential lengths, the stitch cam portions of each cam system were superimposed on the same diagram (see Fig. 70). The circumferential length was assumed to be twice the horizontal distance between the knitting point and the clearing height - this usually being the case with multifeed plain knitting machines.

It may be seen from Fig. 70 that a  $55^{\circ}$  linear cam would provide more feeders than any other cam, and a  $55^{\circ}$  straight and curved cam would provide the least number of

feeders. It is interesting to note that a  $57^{\circ}$  non-linear cam system provides approximately the same number of feeders as does the  $45^{\circ}$  linear cam system.

#### 4. General Conclusions and Suggestions for Future Work

The maximum machine speed values may be compared with the corresponding values of circumferential length to determine the rates of fabric production.

It has been shown, for example, that a  $57^{\circ}$  non-linear cam and a  $45^{\circ}$  linear cam have approximately the same circumferential length, and yet practical speed tests have shown that the maximum speed was 160 r.p.m. for the linear cam (i.e. approximately twice the commercial speed); and greater than 200 r.p.m. for the non-linear cam, (i.e. more than three times commercial speed). It is clear, therefore, that a minimum increase in fabric production of 30% may be achieved by using the non-linear cam.

In addition, calculations of the maximum horizontal force acting on the needle in the  $57^{\circ}$  non-linear cam, have shown that machine speeds as high as 30 times commercial speed might be achieved with the non-linear cam. It was pointed out that imperfections in the manufacture of a non-linear cam would make this speed unobtainable in practice, although the calculations provide a guide to the high speeds that might be expected from non-linear cams, and suggest that much greater speeds than those obtained with the linear cam may be achieved.

It was considered that the  $55^{\circ}$  straight and curved cam system, which was designed with controlled acceleration characteristics, might also provide extremely high maximum knitting speeds compared with the linear cams.

It is suggested that a non-linear cam with a maximum cam angle of  $55^{\circ}$  could be used on commercial machines to provide at least three times commercial speed, or, if for immediate purposes it was thought to be desirable to maintain a separate adjustable stitch cam, then a cam system such as the  $55^{\circ}$  straight and curved might be used to provide approximately twice the present rate of fabric production.

Since the theoretical evidence suggests that speeds much greater than three times commercial speed may be achieved with non-linear cams, it is considered that, for future work, the knitting machine should be completely redesigned, to allow high speeds up to approximately 30 times commercial speed to be obtained. Modifications to the machine, necessary to permit continuous running at these high speeds, as discussed previously in Chapter 3, would include a complete roller bearing system for the main crown wheel assembly, and a balanced fabric rolling mechanism.

With such a machine it would be possible to assess the maximum practical knitting speeds for both the non-linear and straight and curved cams, and an assessment could be made as to whether the limiting speed were caused by poor fabric appearance, needle butt breakage or latch and needle hook failure.

CHAPTER 6

INCREASING THE RATES OF FABRIC PRODUCTION ON  
CIRCULAR WEFT KNITTING MACHINES BY REDUCING THE  
AMPLITUDE OF THE VERTICAL NEEDLE MOVEMENT

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INCREASING THE RATES OF FABRIC PRODUCTION ON  
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AMPLITUDE OF THE VERTICAL NEEDLE MOVEMENT

Preamble

It has been stated in Chapter 1 that the rate of fabric production in circular weft knitting machines is determined by :

- (a) the number of feeders,
- (b) the circumferential speed of the knitting cylinder.

The number of feeders that may be fitted to a machine of given diameter is determined by the circumferential length of each feeder, and this in turn is dependent upon :

- (i) the cam angle,
- (ii) the amplitude of vertical needle movement; (i.e. the vertical distance between the needles at the clearing position and the knitting point).

In Chapter 5 it has been shown that large increases in fabric production may be achieved from a combination of steep cam angles and higher machine speeds with non-linear cam systems. The purpose of the work in this chapter is to consider the alternative method of increasing rates of fabric production, i.e. reductions in amplitude of vertical needle movement.

This feature of machine construction was investigated as follows :

- A. By an examination of new types of needle to give reductions in latch motion,
- B. By the application of a compound tubular needle to a circular weft knitting machine,
- C. By the application of a lifting sinker motion to a circular weft knitting machine.

A. Considering New Types of Needle to give Reductions in Latch Motion

From an examination of the latch needle during loop formation (see Fig. 3, Chapter 1), it is evident that the vertical distance between the running position and the clearing position (i.e. the distance necessary to clear the old loop from the end of the open latch) represents almost 50% of the total vertical needle movement. It is considered that if any substantial reductions in vertical amplitude are to be achieved then the needle must be redesigned to function without this latch clearing motion.

A few provisional designs of such needles are shown in Fig. 71. Diagrams A, B and C and D consider latch components which are designed to open either upwards against the back of the needle hook or outwards away from the needle stem - both types of latch motion reduce the vertical amplitude by avoiding the downwards latch motion employed with the conventional latch needle.

Diagram A considers a latch component that opens upwards against the back of the needle hook, and is pivoted in the same way as the conventional latch needle. Diagrams B, C and D consider latch components that are free to move outwards through the slot in the back of the needle stem, where B and D are

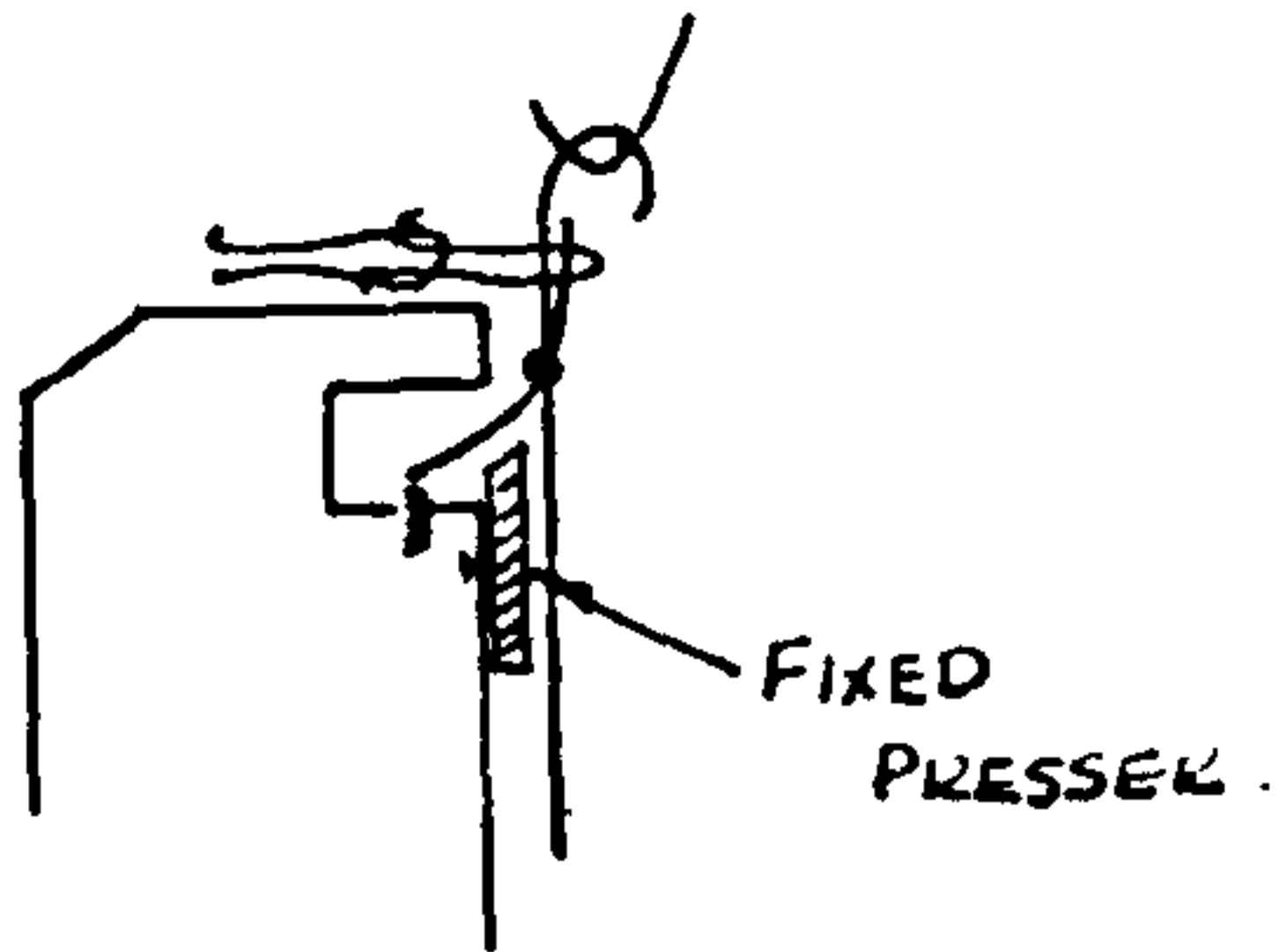


NEEDLE DESIGNS

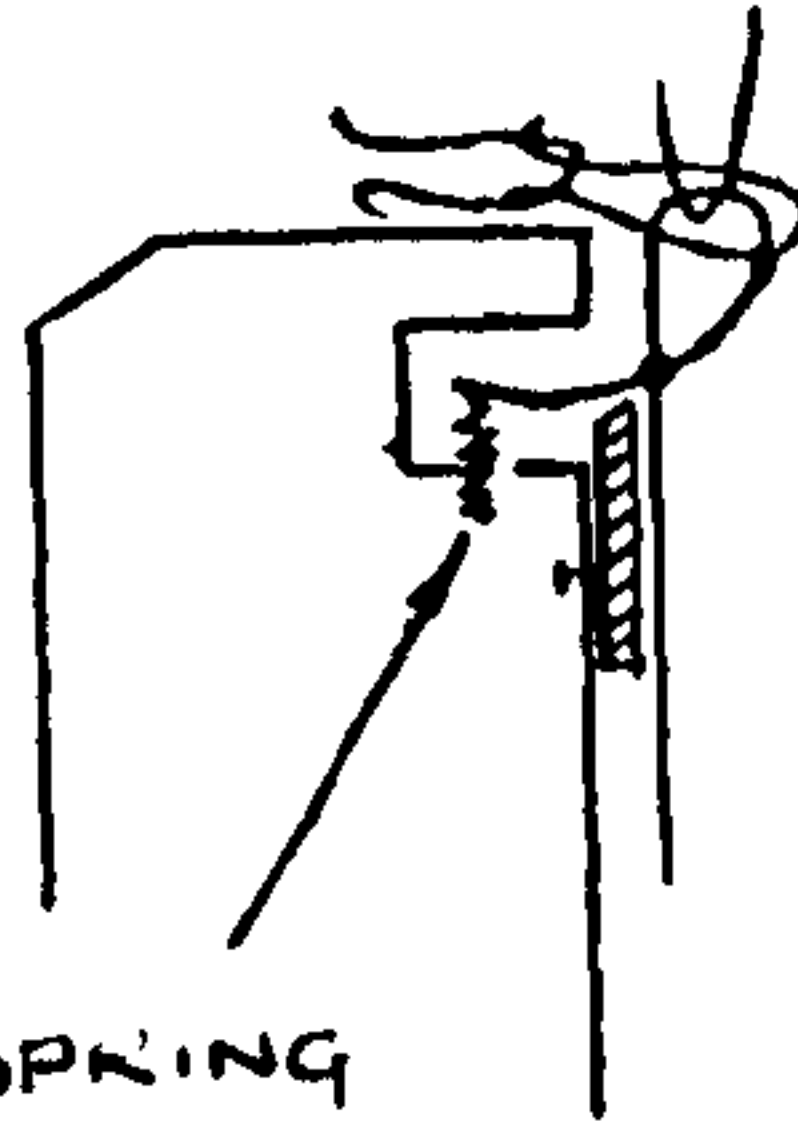
CLEARING POSITION

KNOCK-OVER POSITION

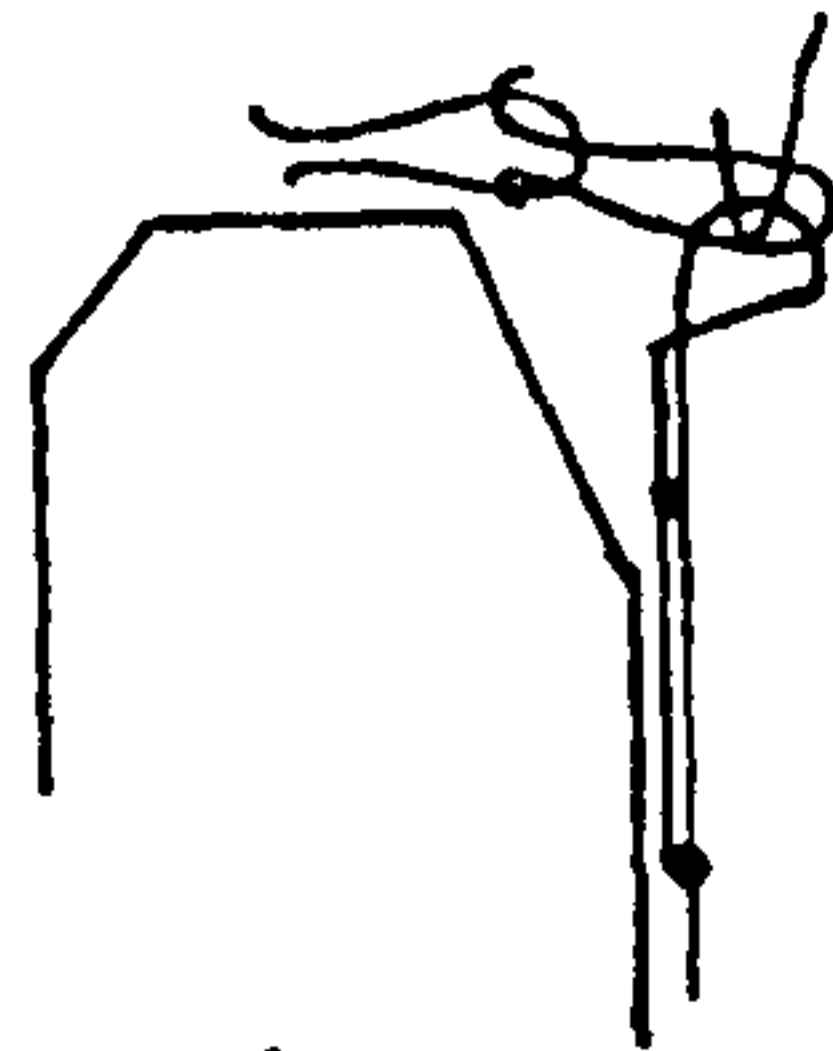
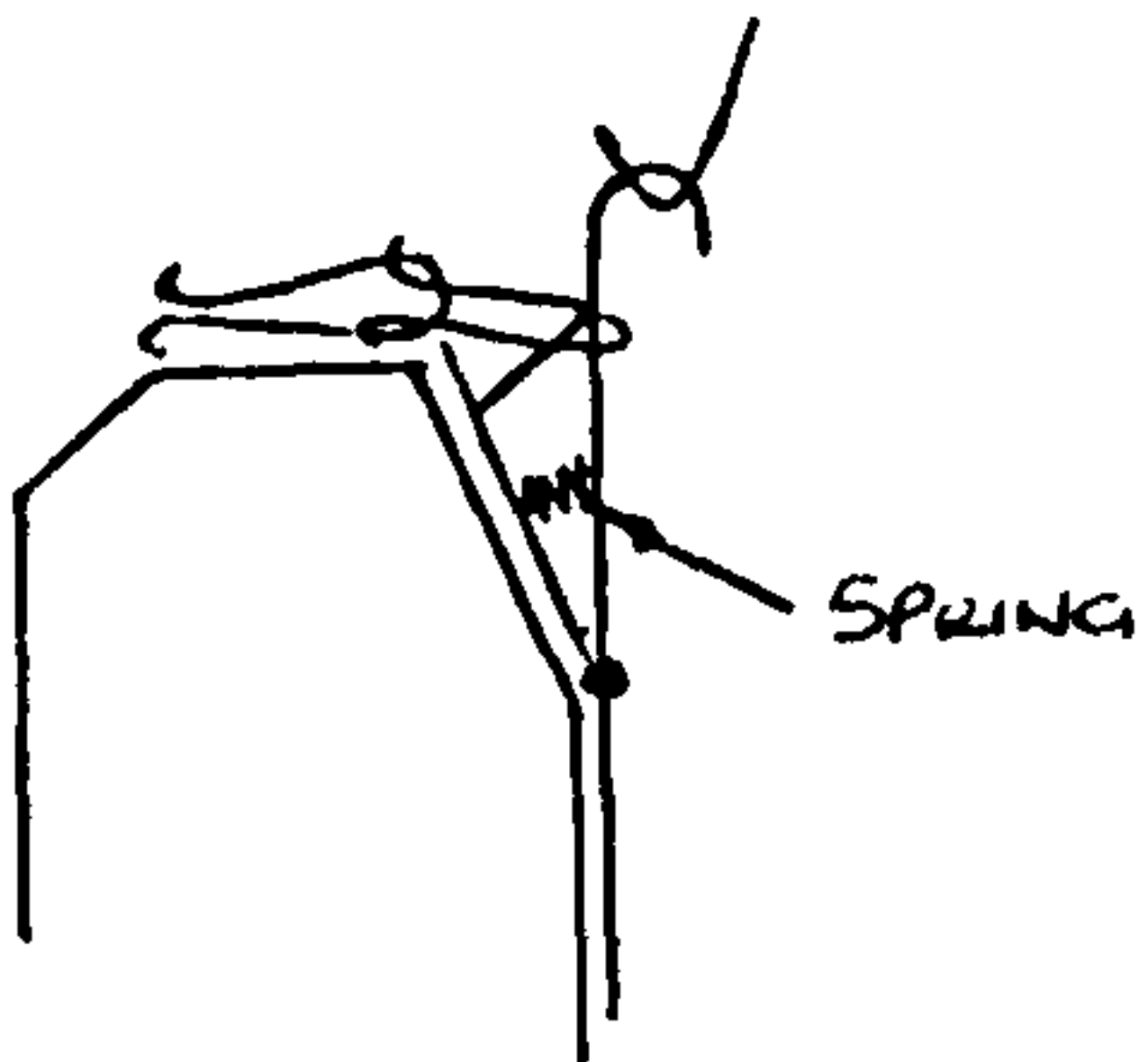
A



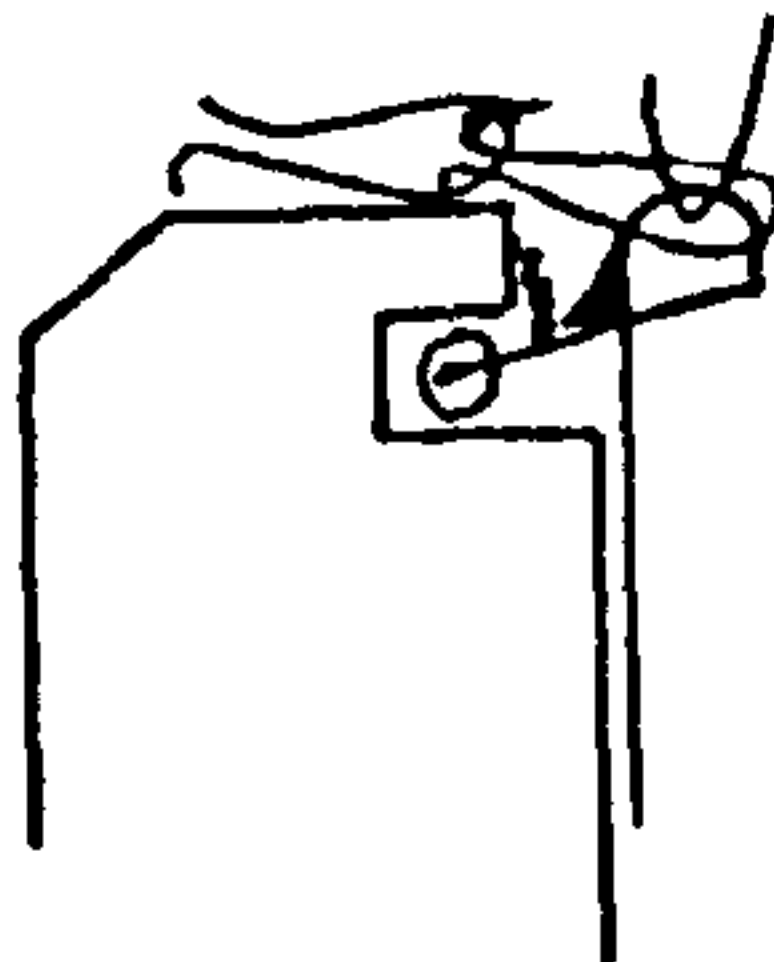
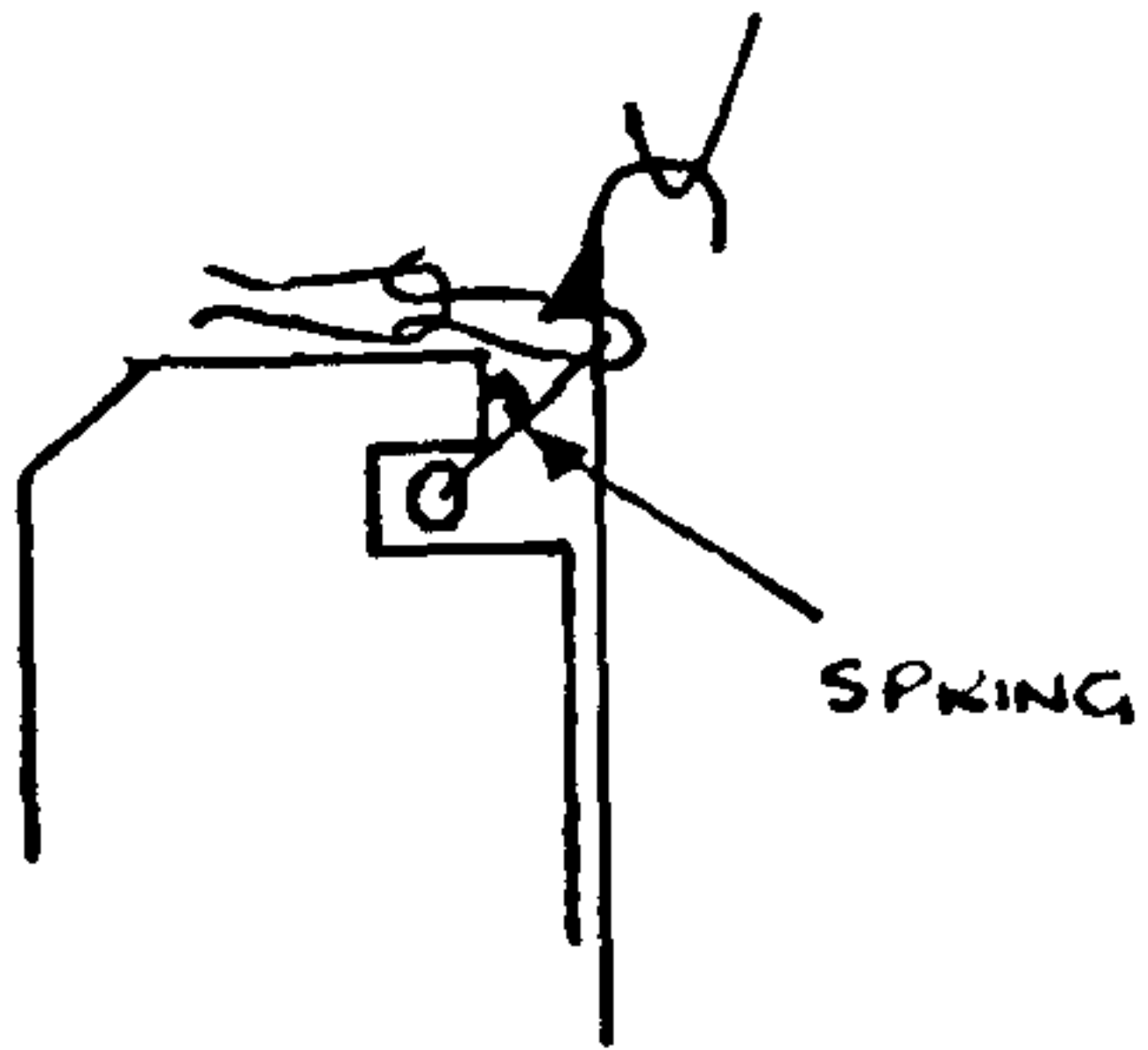
SPRING



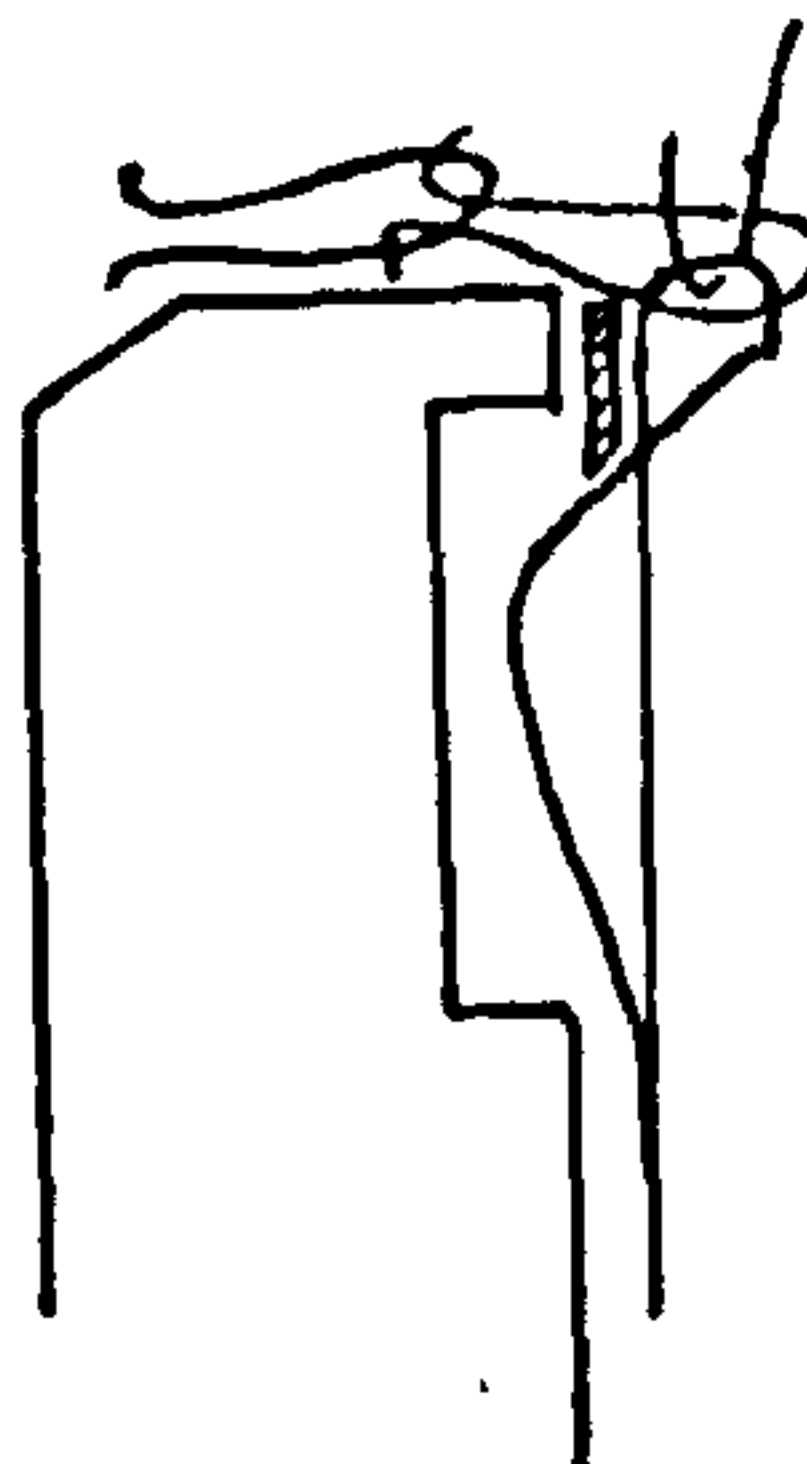
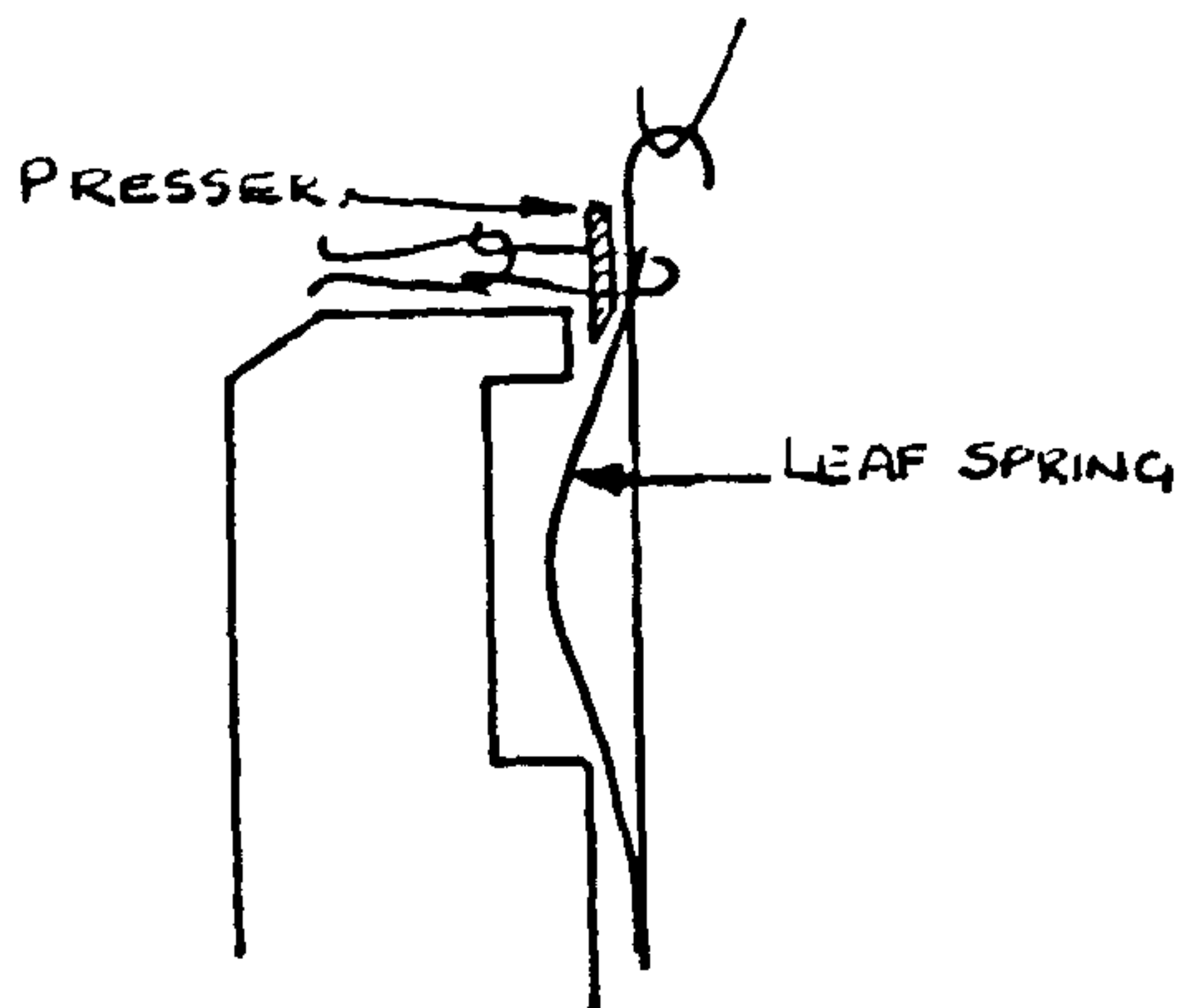
B



C



D



pivoted at a lower position on the needle stem and C is pivoted from the cylinder wall.

For all the designs A to D it was considered that a satisfactory mating of latch and hook could be achieved by a spoon shape at the end of the latch. This is similar to the conventional latch needle but with these special designs the spoon is inverted and acts on the inside of the needle hook.

The conventional latch needle is often termed a self-acting needle, as the knitted loops open and close the latch during loop formation. With the special designs, however, it was apparent that a mechanical system must be used to open and close the latch.

Some provisional mechanical systems for opening and closing the latch component may be seen from Fig. 71.

For design A a fixed presser is mounted on the trick wall and this would close the latch on to the hook when the needle descended; the latch would be opened by means of a spring when the needle was lifted by the upthrow cam.

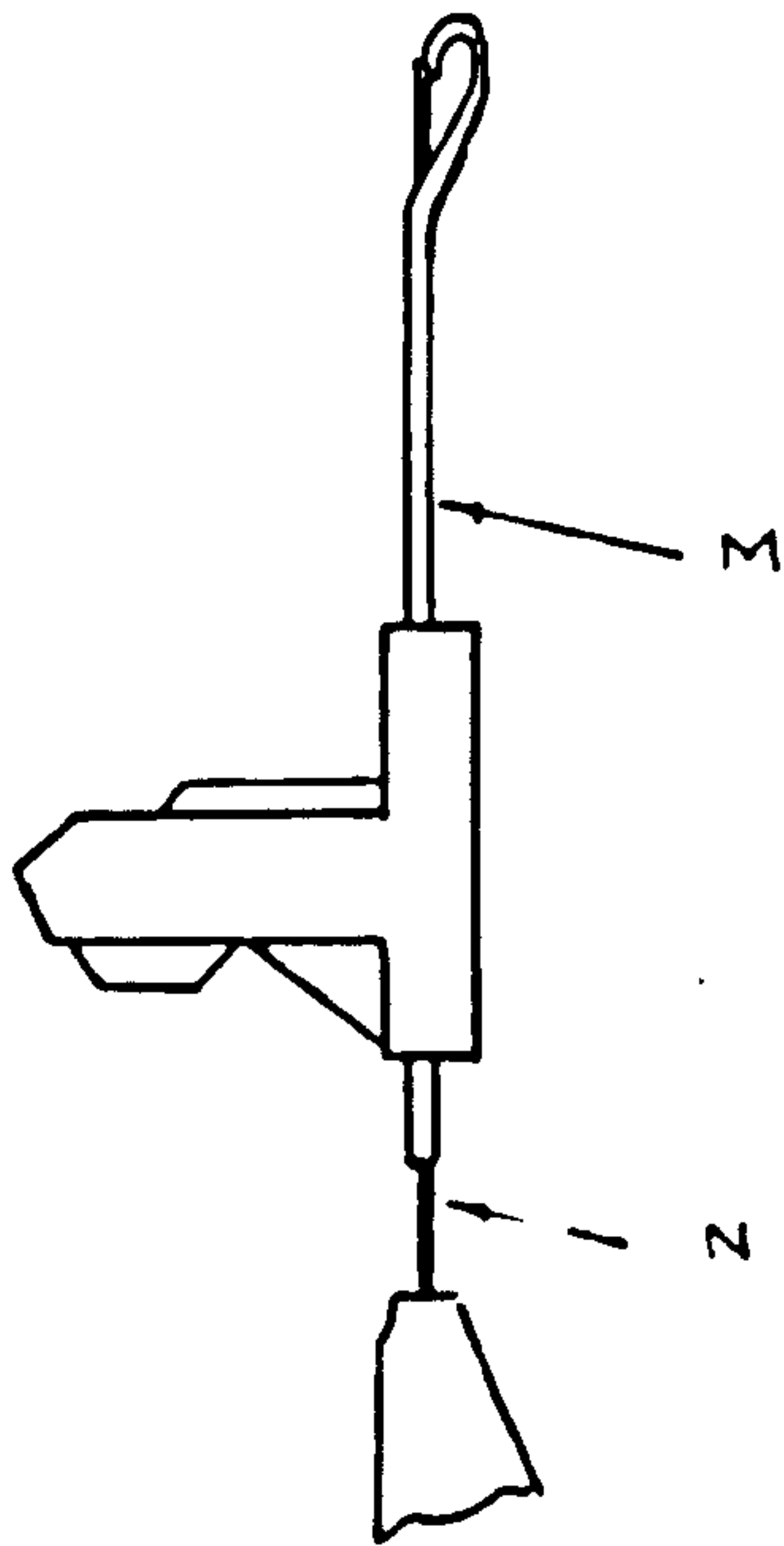
For design B the latch component would close on to the needle hook due to the shape of the trick wall, as the needle descended, and could be opened by means of a compression spring between the latch and needle stem.

For design C a projection on the back of the needle hook would close the latch when the needle descended, and a tension spring would open the latch when the needle was raised.

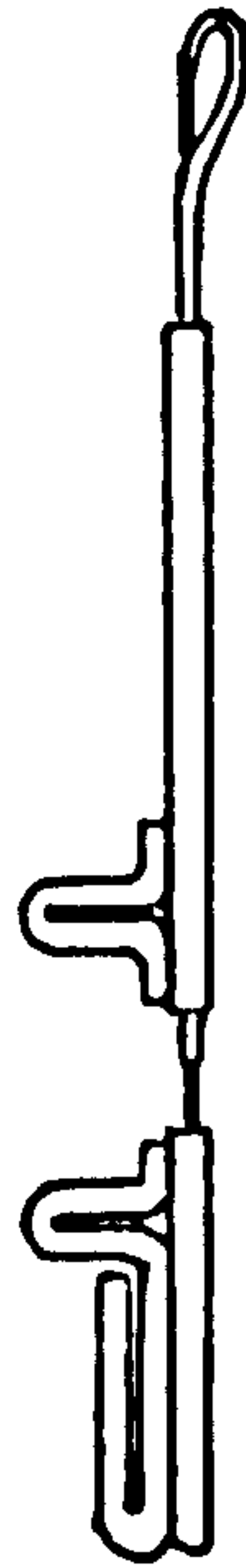
For design D an external presser arrangement would operate to close the leaf spring to the hook, and the latch would open when the presser was released.

FIG 72

A WARP KNOTTING  
COMPOUND NEEDLE



B. MODIFIED COMPOUND  
NEEDLE FOR WEFT KNOTTING



It was appreciated that although a mechanical system for opening and closing the latch might be made to work on these lines, the resultant needle would inevitably be complicated, and modification to the knitting cylinder would be difficult.

It was considered that a more appropriate needle to suit the necessary requirements would be a modification of the existing compound needle, which is used on some warp knitting machines. The existing warp knitting needle is shown in A, Fig. 72, and it comprises a hook portion (M) made from metal tubing, and a tongue portion (N) which slides freely in the tube. The hook and tongue portions are mounted in separate horizontal bars, and the relative motion necessary to bring the tongue and hook together is provided by a cam shaft arrangement which is attached to the horizontal bars.

If such a needle were to work on circular weft knitting machines, where needles would be knitting one after the other instead of together, both the hook and tongue portions would require separate needle butts and separate cam systems. The proposed design of the compound needle to be used, for this work, on a weft knitting machine is shown in B, Fig. 72.

## B. Application of a Compound Needle to a Circular Weft Knitting Machine

### 1. Construction of the needle

The compound needle was chosen for use on circular weft knitting machines, as this needle not only provides reductions in vertical amplitude, but the mechanical control of a cam system for opening and closing the tongue and hook is much more precise than the mechanical systems shown for the other modified needle designs.

The first compound needle to be made for this work was constructed from the tongue and hook parts of an existing fine gauge (i.e. 28 n.p.i.) warp knitting needle. To construct the needle, the aluminium coating was removed and replaced by suitable gauge needle butts which were soldered to the shanks of the hook and tongue. Although this needle was considered to be suitable for use on circular weft knitting machinery, it was decided to construct a coarser gauge needle for the practical application, as this would reduce the number of needles to be made.

The largest warp knitting compound needle that could be obtained was designed for a 14 gauge machine. A number of these needles were obtained and the aluminium casting was removed, the tongue and hook shanks were fitted into, and soldered to, thin hyperdermic tubes to give an overall gauge of 7 n.p.i. Needle butts were ground from 7 gauge latch needles and soldered to the tongue and hook portions. A number of modified compound needles suitable for circular weft knitting machinery were constructed. Both the fine gauge and coarse gauge compound needles constructed in this work may be seen in Fig. 75.

## 2. Modifications to a Circular Weft Knitting Machine

The circular machine obtained for this work was a rotating cam box type with a small cylinder diameter (i.e. 1 inch diameter). It had previously been used as a two feeder machine (i.e. the maximum number of feeders that may be fitted to a 1 inch diameter machine), for making pyjama cord fabric.

This small diameter machine was considered entirely suitable for the investigation, as any part of the knitting head could be modified easily or remade with normal engineering machine tools; also with such a small diameter cylinder only a relatively few modified needles were required.

The modifications made to the machine to suit the requirements of the needle included :

- (i) A new 1 inch diameter cylinder cut with 20 needle slots which were chamfered at the top of the cylinder to suit the shape of the back of the needle hook.
- (ii) A cam section with a window so that the needle butt performance could be observed.
- (iii) Special cam systems for the hook and tongue portions (discussed in detail in the following part).
- (iv) A yarn feeding system which guided the yarn through tubes to the needle hooks.

The modified machine is shown in Fig. 73.

### 3. Cam design to suit the compound needle

To illustrate the requirements for the cam design, the loop-forming action of the compound needle is shown in A, Fig. 74. At position J (clearing position) yarn is introduced to the hook of the needle and the old loop is below the tongue. At position K (knock over position) the old loop is pressed over the closed tongue and hook, and the new yarn is trapped in the hook.

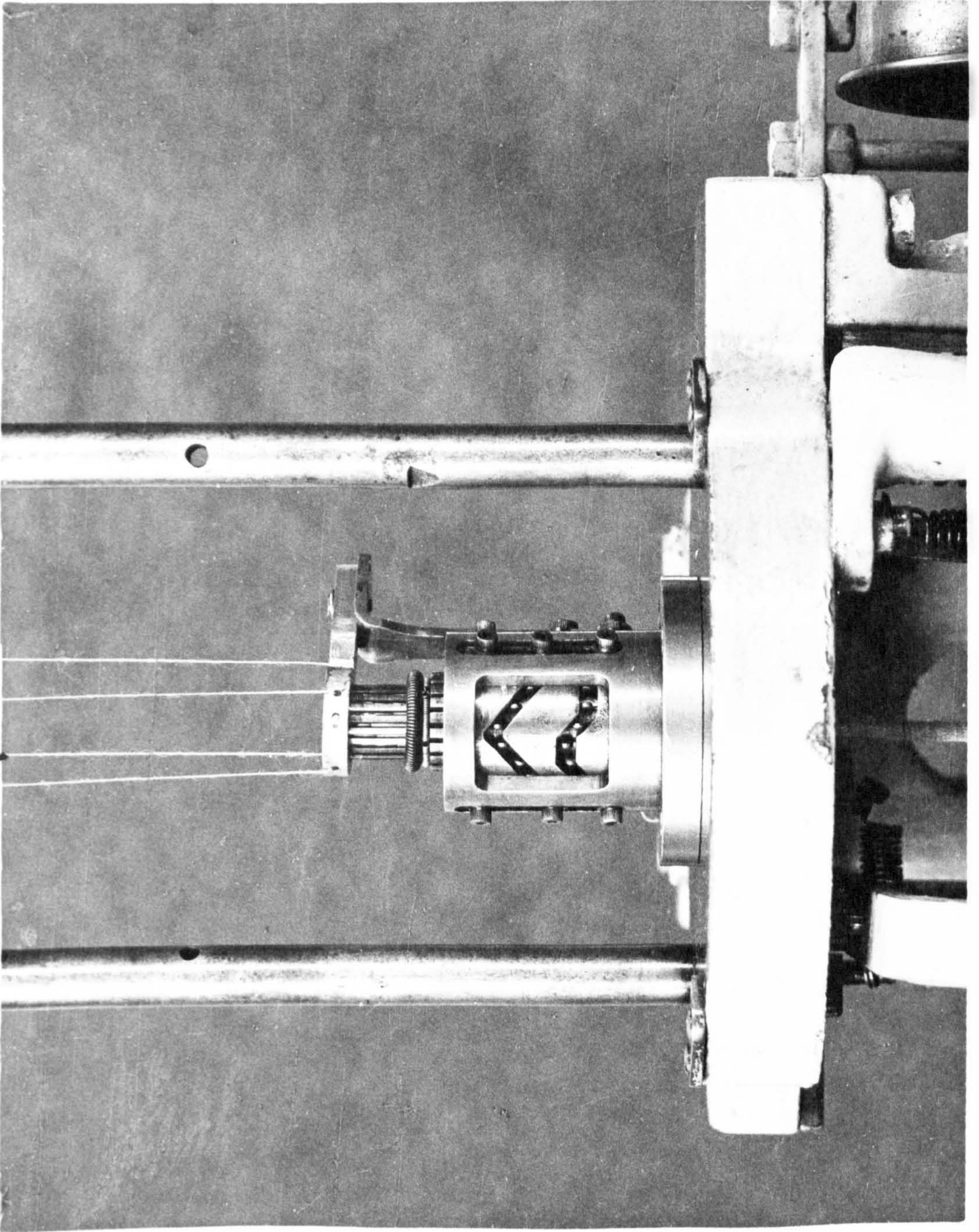
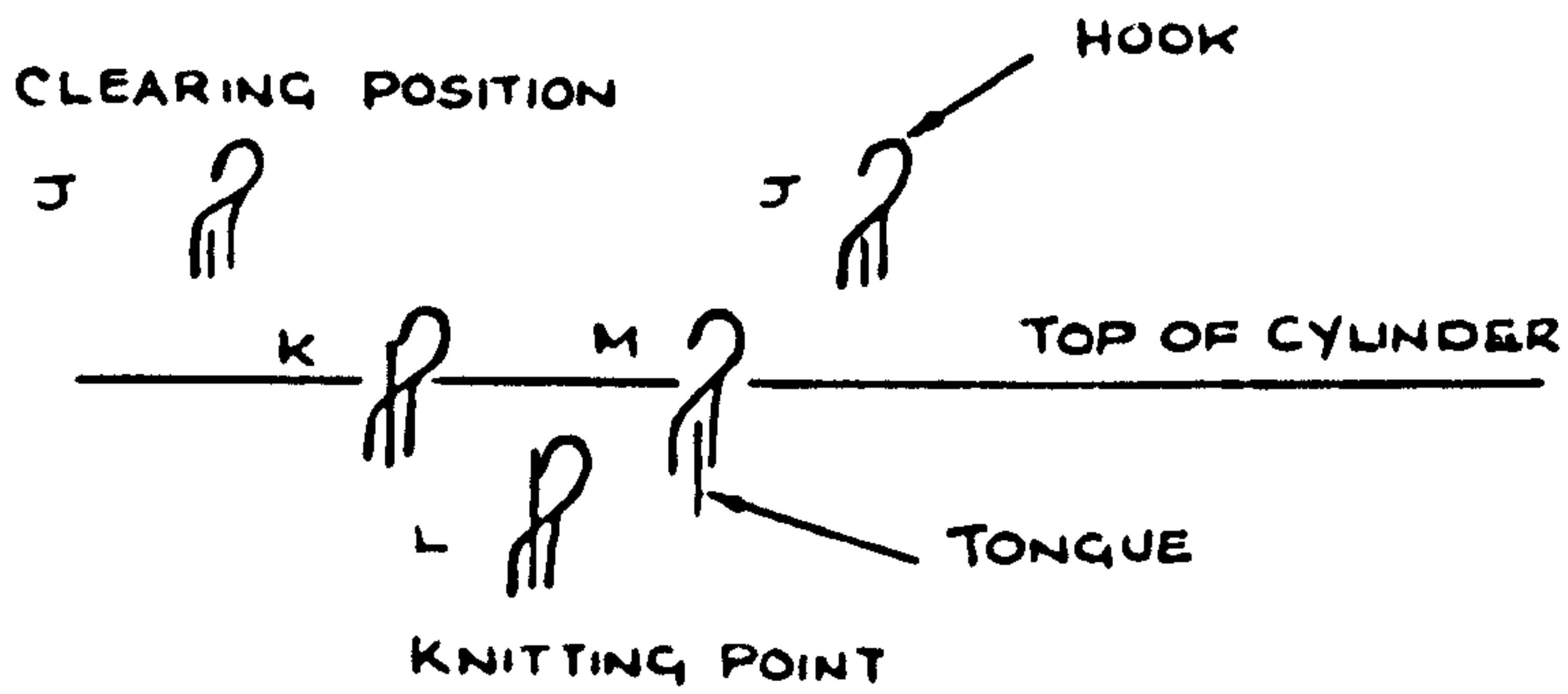


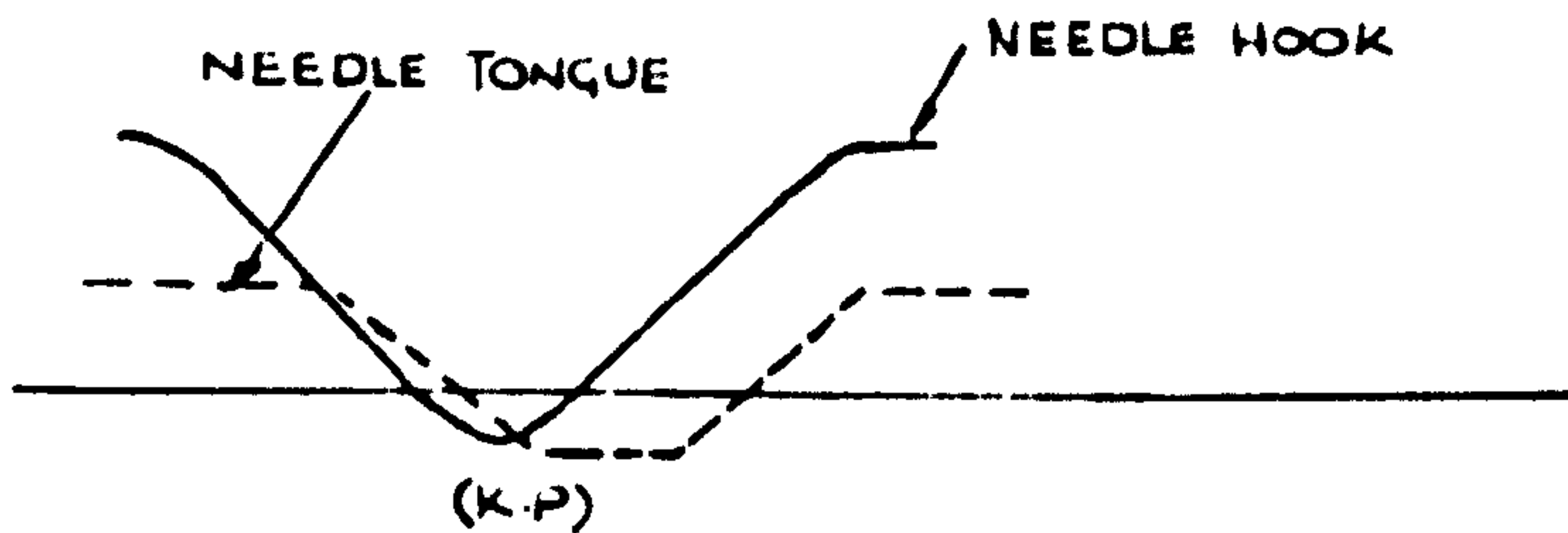
FIG 73

Fig. 74

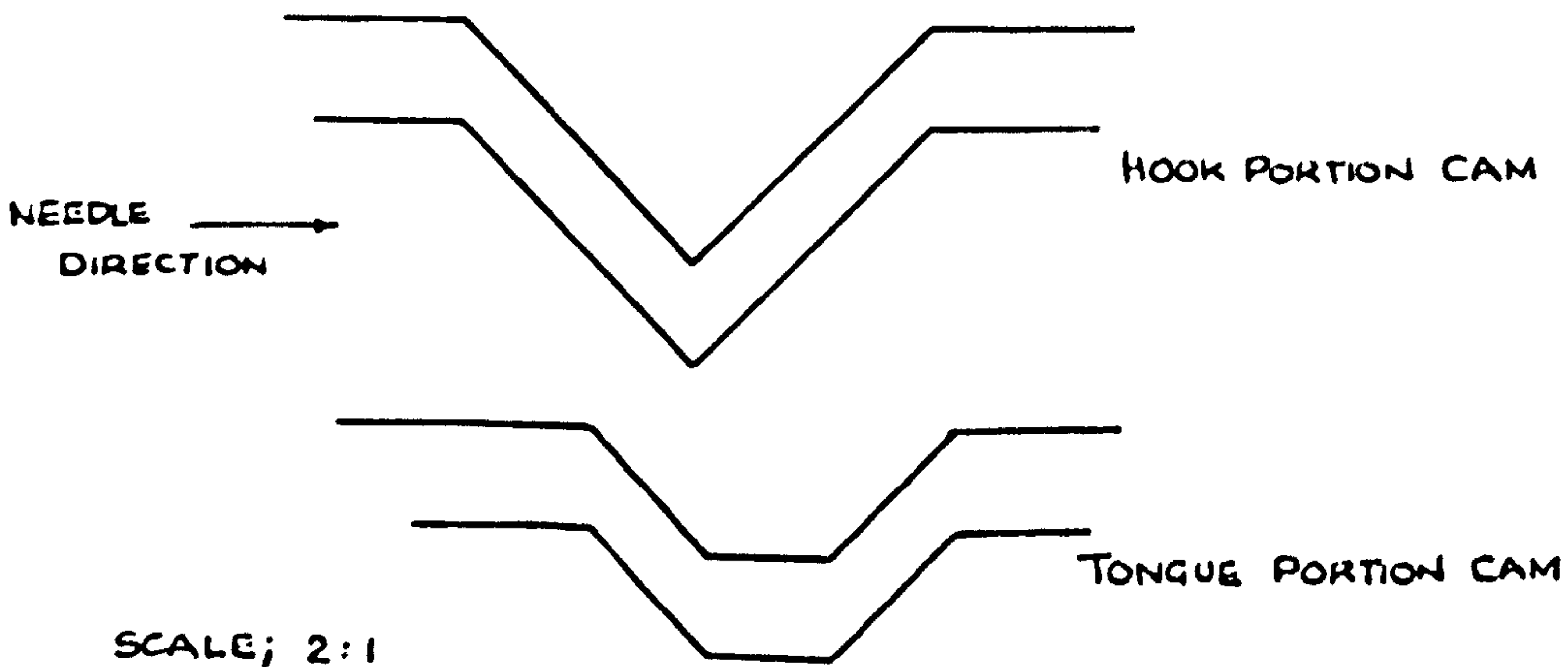
A. KNITTING ACTION WITH A COMPOUND NEEDLE



B. RELATIVE MOTION OF NEEDLE HOOK AND NEEDLE TONGUE



C. CAM DESIGN FOR A WEFT-KNITTING, COMPOUND NEEDLE





At position L (knitting point) the new loop is drawn below the cylinder top and the length of yarn in the loop is determined by the needle depth below the cylinder. At position M (running position) the yarn remains in the hook until clearing commences.

The relative motion of hook and tongue portions may be shown as a line drawing (see B, Fig. 74), and from the measurements of their vertical displacements, a cam system was designed (C, Fig. 74).

The cam systems were made from lengths of cast steel bar of 1.4" outside diameter and 1.0" bore. The bar was cut across the diameter to provide semi-circular pieces, and the cam shapes were filed carefully by hand. Great care was necessary in the making of these cams as correct motion of needle and hook parts depended upon the accuracy of the cams. Both a single feed and a four feed compound cam system were constructed for this work.

The compound cam system may be compared with the original latch needle cam system (see Fig. 75) and it is noted that the circumferential length has been reduced by approximately 50%; this suggests that four feeders can be fitted to the machine which means that twice the rate of fabric production may be achieved.

#### 4. Experimental Trials and Conclusions

The preliminary knitting trials with a single feed compound cam system were successful as satisfactory fabric was produced at commercial speeds (i.e. 750 r.p.m.). To complete the work, additional yarn feeding attachments were fitted to the machine and the four feed compound cam system was tested.

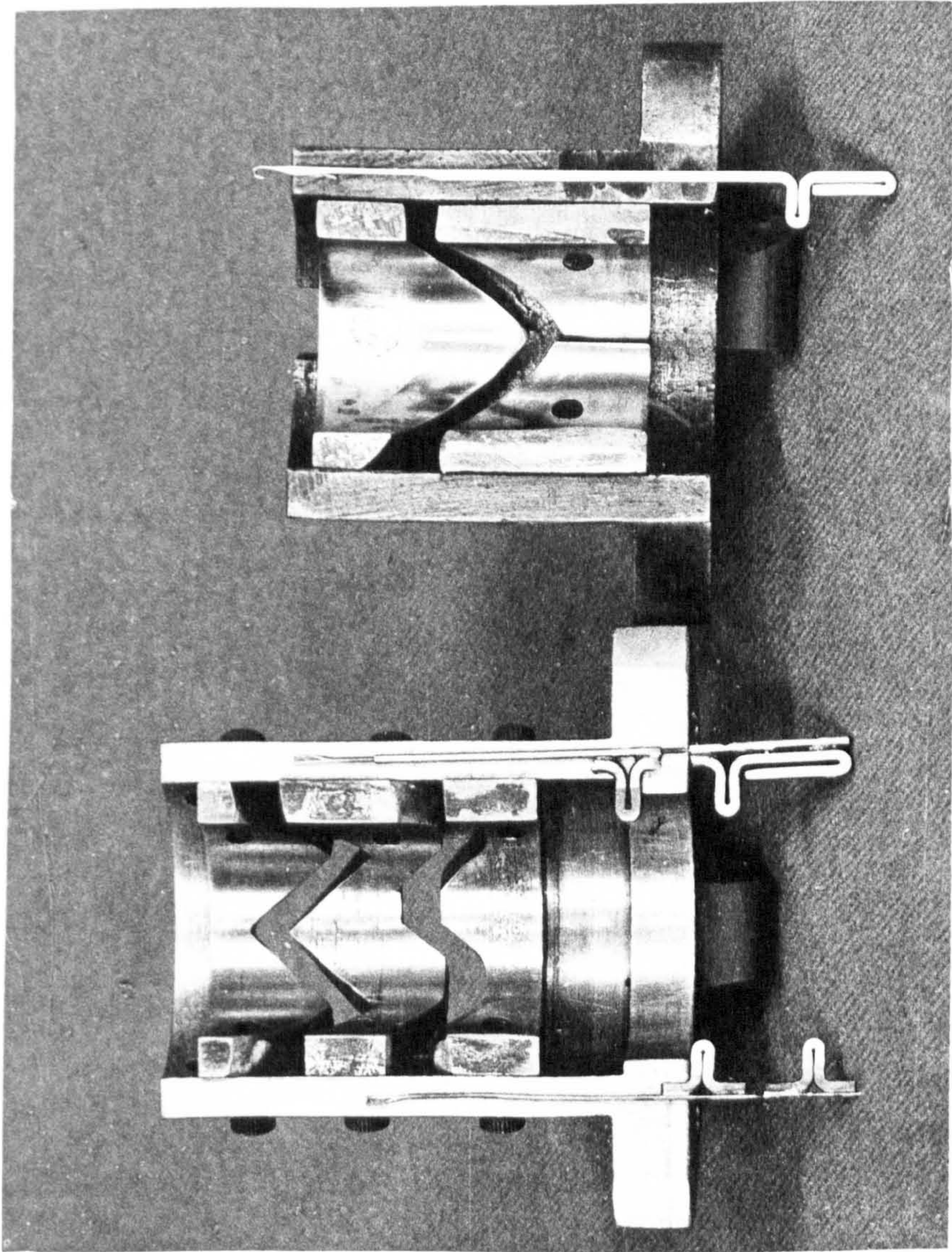


FIG 75

Knitting trials at commercial speeds with this cam system were also successful and satisfactory fabric was obtained.

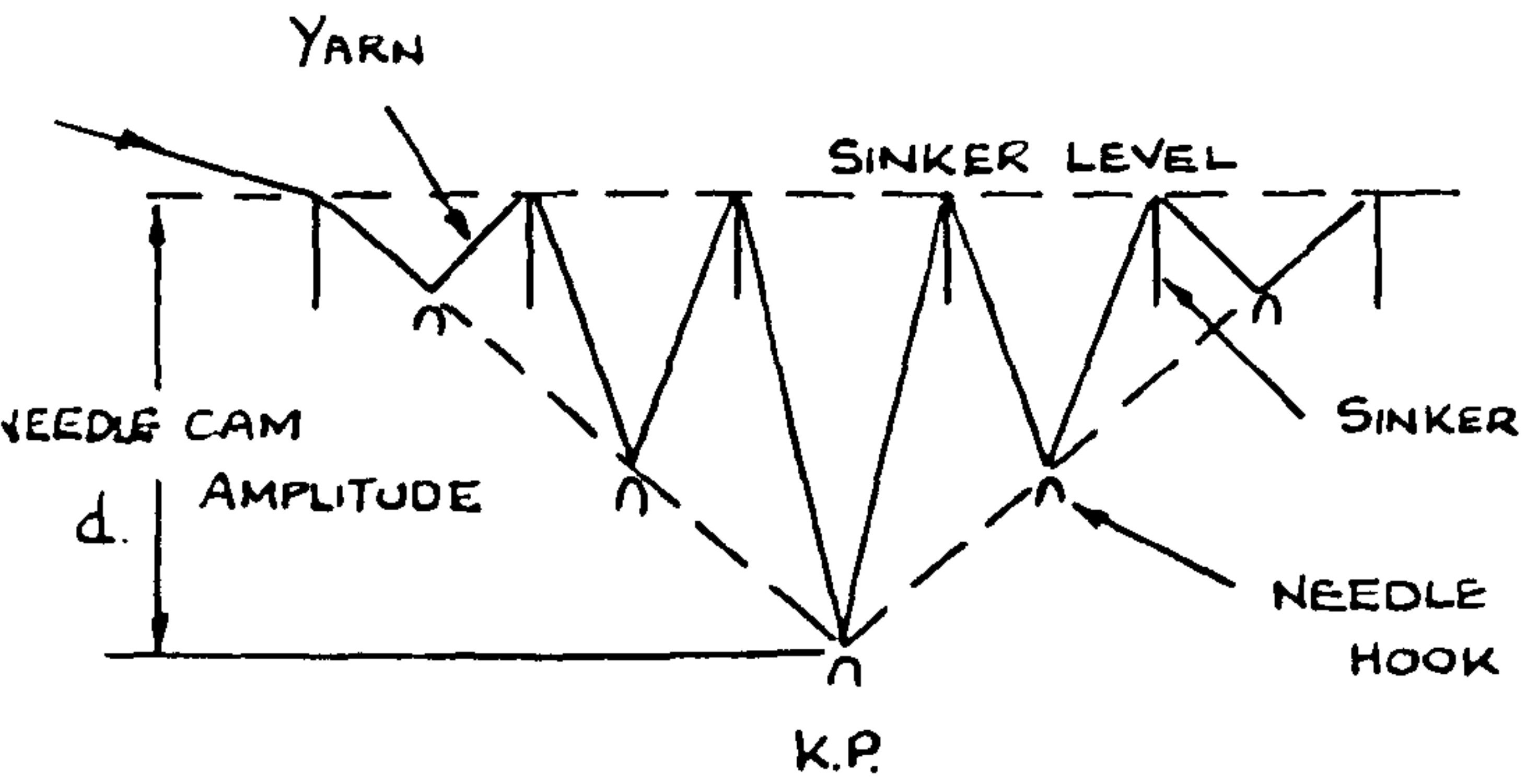
This investigation has shown, therefore, that compound needles would function satisfactorily on circular weft knitting machinery, and could be used to increase rates of fabric production without either increasing the cam angle or increasing the machine speed.

It was appreciated, however, that a combination of compound needles and the non-linear cams (discussed in Chapters 2 - 5) may be used to give much greater increases in rates of fabric production than can be obtained when using latch needles, as higher machine speeds as well as more feeders would be possible. It was also realised that for extremely high speed knitting, the compound needle would be more suitable than a latch needle; with the latch needle the continued impact of latch and hook could eventually cause hook breakage or latch pivot wear, whereas tongue and hook of a compound needle would be unaffected by increases in machine speed.

Another aspect of knitting with compound needles on circular weft machines is that they should prove to be as versatile as the latch needle and therefore could be used to produce the most complicated knitted structures that involve tuck and miss stitches. For instance, tuck stitches could be achieved by lowering the tongue prior to its reaching the knock-over height, thus drawing more yarn into the hook without knocking over the previously knitted loop. Miss stitches could be obtained by lowering both tongue and hook portions prior to clearing height and therefore avoiding drawing more yarn from the package.

FIG 76

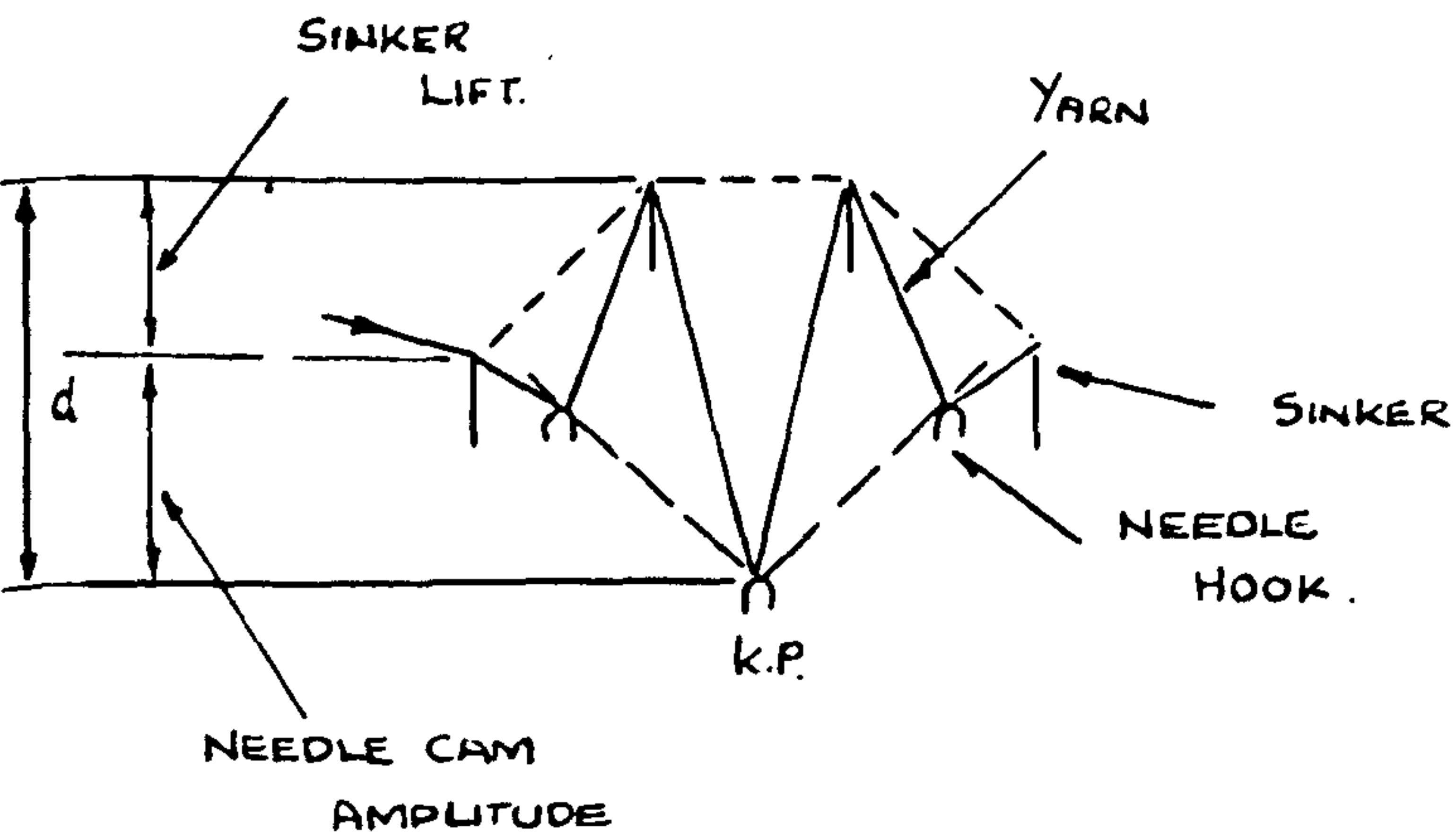
A LOOP FORMATION WITH A  
CONVENTIONAL SINKER



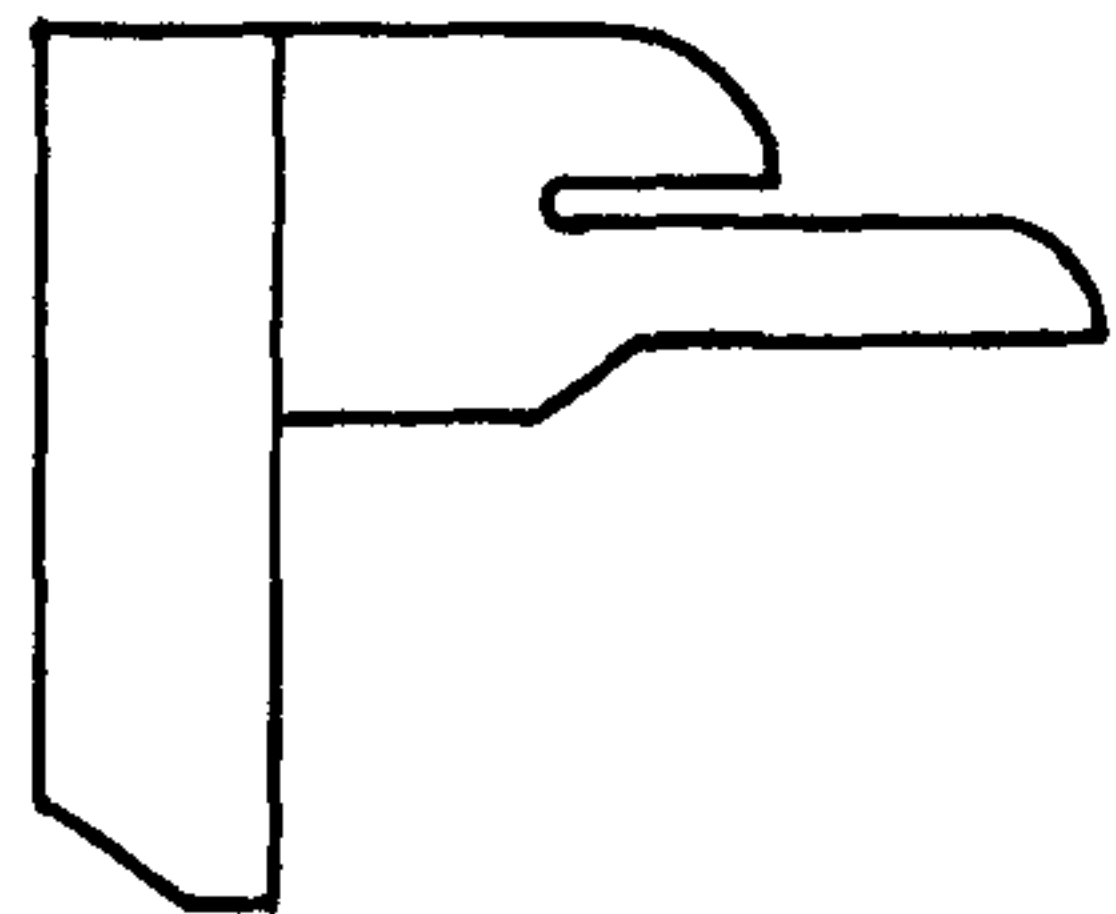
C. CONVENTIONAL SINKER



B. LOOP FORMATION WITH  
A MODIFIED SINKER



D. MODIFIED SINKER



With stitches such as single piqué, double piqué, etc., the selection to tuck or miss would be controlled by cam design, and with structures such as double jersey and patterned fabric independent needle butt selection could be achieved by a pattern wheel.

C. The Application of a Lifting Sinker Motion to a Circular Weft Knitting Machine

1. Loop-forming action and a sinker modification

With sinker type knitting machines, which are employed for making single jersey fabric, the only movement of the sinker is the withdrawal motion at the feeding point to allow the descending needle to move down to the knitting point and draw a loop. It was considered that if the sinker could be controlled to give an upward vertical displacement at the knitting point, then the amplitude of a cam system could be reduced considerably.

The relative positions of sinker and needle hook during loop formation are shown in A, Fig. 76, for the existing commercial machine, and in B, Fig. 76, for the proposed machine where the sinkers rise at the knitting point.

From B, Fig. 76, it may be seen that a lifting sinker movement could provide a reduction of 50% of cam amplitude, thus allowing twice the number of feeders to be fitted to the circumference of the machine, in the same way as has been achieved by redesigning the compound needle. To provide for a vertical sinker movement, it was necessary to modify the commercial sinker. C, Fig. 76 shows a typical commercial sinker and D, Fig. 76 shows the type of sinker that would be suitable for this project. It was considered that an upward lift of the modified sinker could be achieved by a cam system

acting on the sinker butts. Several sinkers, of the type shown in D, Fig. 76, were made by soldering rectangular steel pieces to the web holding section of the commercial sinker.

2. Modification to a circular weft knitting machine and design of the cam systems

A machine, similar to the one used for the compound needle work was acquired for the sinker project, (i.e. a 1" diameter, two feed, rotating cam box machine). The advantages of such a small diameter machine are that modifications or new parts could be made easily with standard machine tools, and only a few modified sinkers were necessary.

The main modification made to the machine was to cut completely through the existing slots of the sinker ring, so that the new sinker butts would protrude through and would be acted upon by a sinker cam system. Other machine modifications included new cam sections and a redesigned sinker top; the modified machine is shown in Fig. 77.

The cam systems used for the sinker project consisted of two separate cams - one to move the sinker butts up and down, and one to control the needle butts. It was not necessary to make a new cam system for the needle butts, as the upper cam system constructed for the compound needle hook portion had been designed to provide a vertical movement equal to half the amplitude of a commercial machine (C, Fig. 74), and, therefore, was entirely suitable for the preliminary trials on the sinker project.

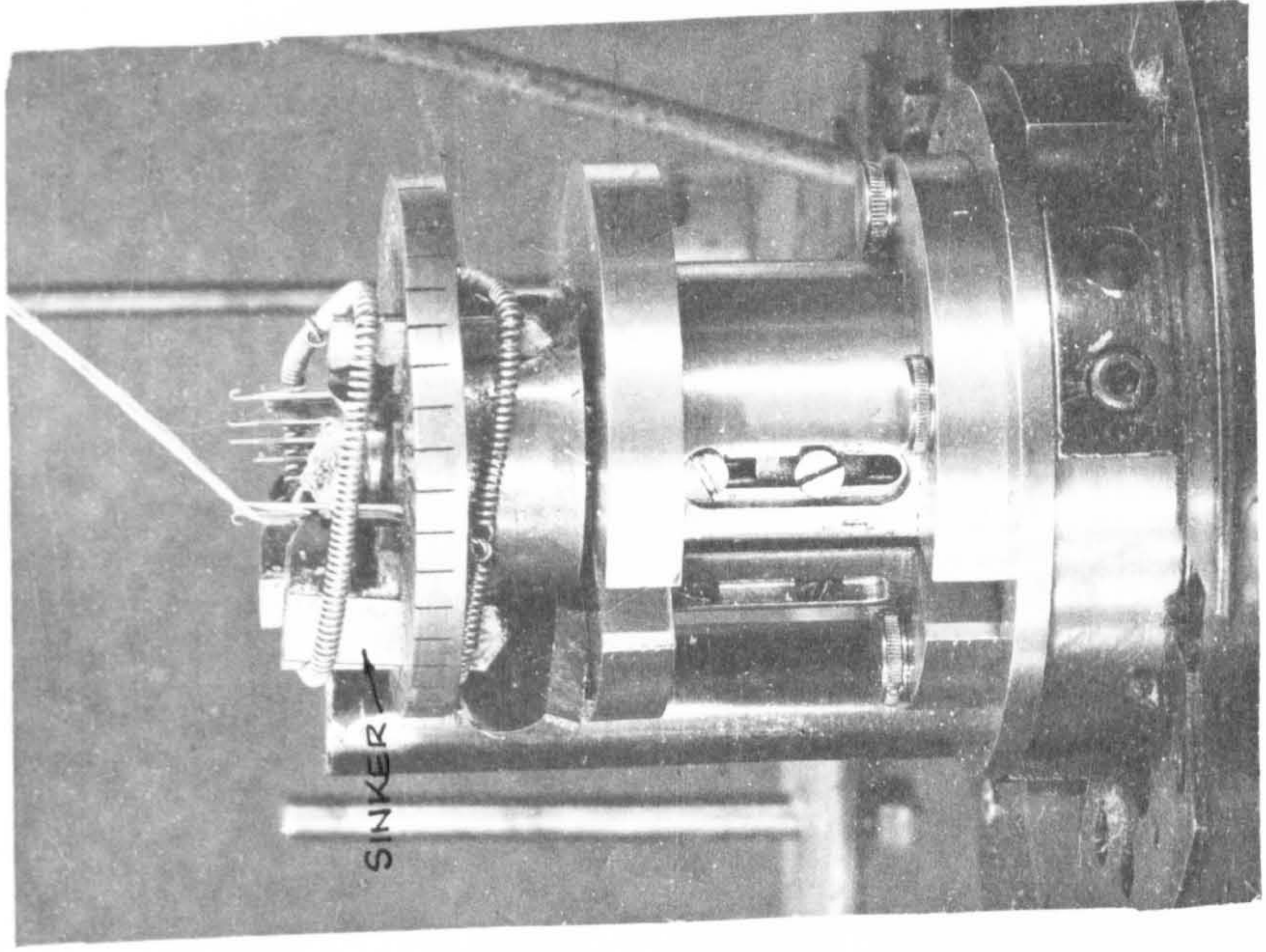
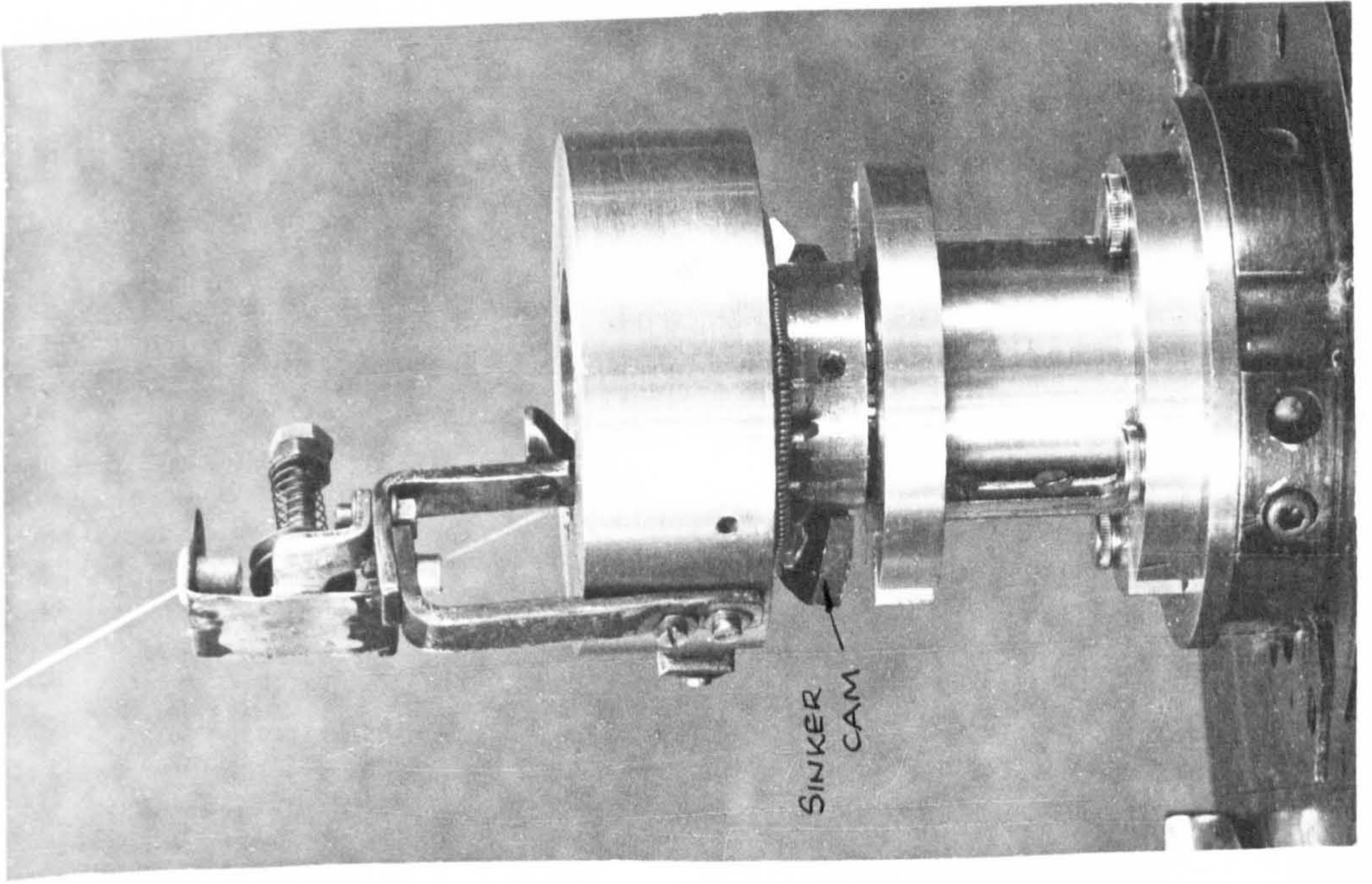


FIG 77

For the sinker cam (see Fig. 77), it was necessary not only to lift the sinkers up and down at the knitting point, but also to move the sinker back so that the needle could pull through a new loop. To achieve this the sinker cam system constructed for this work pushed the sinker butt both upwards and outwards. Springs were attached to the sinkers to ensure correct vertical displacement and the downward movement of the sinkers was controlled by the inclined cam in the sinker top, which pushed the sinkers back to the running position.

### 3. Experimental Trials and Conclusions

Preliminary trials indicated that a lifting motion of the sinker could be used to reduce the amplitude of vertical needle movement by 50%, since yarn was easily drawn over the knock-over portion of the rising sinker by the needle hook. The main drawback with such a machine, however, is that the fabric must rise with the sinkers; this introduces a problem which involves the fabric take-down tension and the size of the loops. When large loops are being produced and an average take-down tension is applied, the loops stretch as the sinkers rise and occasionally a loop will fall over the sinker knock-over portion and go below the sinker level. With smaller loops this does not occur but the construction of the fabric is too tight for normal commercial use. Also with these smaller loops a high take-down tension is necessary to clear the loop from the end of the latch.

It was obvious from these preliminary investigations that more elaborate modification of the machine was necessary in order to produce a satisfactory machine operating with a lifting sinker motion. One possible modification would be to incorporate sinkers with longer knock-over portions.



The principle, however, appeared to be sound and, with suitable modifications, it is suggested that the application of a lifting sinker motion would enable a rate of fabric production twice that of commercial production to be achieved. Even greater increases should be possible if non-linear cam systems of the type described in Chapter 2 were used.



CHAPTER 7

SUMMARY

CHAPTER 7

SUMMARY

The purpose of Phase I of this work, described in detail in Chapters 2 - 5, was to determine whether curved cam systems, designed with mathematically controlled acceleration characteristics, could be used on weft knitting machinery to provide higher rates of fabric production.

Two linear cam systems (i.e.  $45^{\circ}$  and  $55^{\circ}$  cam angle) which provide no acceleration control for the needle butts were constructed and the performance of these commercial type cams was compared with that of three non-linear cams, (i.e.  $52^{\circ}$ ,  $57^{\circ}$  and  $62^{\circ}$  maximum cam angle) and a  $55^{\circ}$  straight and curved cam, which were of polynomial design and provided acceleration control for the needle butts.

In the analysis given in Chapter 5, calculations of the yarn tension build up within the loop-forming portions of each cam showed that, for all cam systems, the values of maximum yarn tension and the stitch lengths calculated from these positions were in close agreement with the experimental results. The calculated maximum tension values for non-linear cams were greater than those of linear cams, due to the much slower rate of change in displacement of the non-linear cam, which involved more knitting elements in the loop-forming process. For example, a  $52^{\circ}$  non-linear cam provided a maximum tension value of 240 gms when calculated from a high yarn input tension of 20 gms, whereas the corresponding maximum tension value for the  $45^{\circ}$  linear cam was 110 gms. When knitting fabric with these cams at 20 gms input tension, holes appeared in the fabric with the  $52^{\circ}$  non-linear cam but not with the  $45^{\circ}$

linear cam. This was easily explained as the measured breaking load of the yarn was 228 gms and this value had clearly been exceeded in the case of the non-linear cam.

At input tensions below 20 gms the maximum tension values of all the non-linear cams were below the breaking load of the yarn, and satisfactory fabric was produced.

The stitch length values calculated from the positions of maximum tension and measured from the fabric for an input tension of 20 gms, for example, were respectively 0.156" and 0.159" for the 52° non-linear cam, and 0.140" and 0.144" for the 45° linear cam, thus illustrating extremely close agreement.

This close agreement between the calculated and measured results suggested that the method of calculating the tensions within the loop-forming portions might be used to predict the knitting performance of any new cam shape prior to its manufacture.

To measure the machine performance, a torque device was acquired and mounted to the main drive of the machine; this instrument measured the total torque required to drive the machine. In addition, a strain gauge force measuring device was designed and constructed, which measured directly the force of the needle butts on the cam. It was evident that the force measuring device, which was used in conjunction with an oscilloscope, provided much more detailed information of the needle butt performance than did the machine torque device. For instance, large needle forces caused by damaged needle tricks, bent needles or badly aligned cams, etc., could be detected easily on the oscilloscope traces, recorded on film and the faulty component located. These results suggested that a direct force measuring device of this type would be extremely useful for knitting machine builders, to test new cam systems, knitting elements, etc.

The results obtained from the strain gauge force measuring device showed that an increase in machine speed produced an increase in the horizontal force for all cam systems. In attempting to explain these force increases, the acceleration forces of the needles were calculated for different machine speeds. These values showed that the calculated horizontal force acting on the needle under dynamic conditions did not contribute to an increase in the horizontal force. However, experiments with and without lubricant on the needles showed that the measured force increase could be explained from a condition of boundary lubrication. It was suggested that the supply of lubricant between the needle shank and the trick wall produced an increase in the shear force needed to overcome the viscous drag of the oil when the machine speed was increased - thus the force resisting the vertical movement of the needle in its trick increased with an increase in machine speed.

Further measurements of the forces showed that the total horizontal force was similar for both the linear and non-linear cams. It was explained that although non-linear cams produce much higher maximum yarn tension values than do the linear cams, the effort to drive the machine was unaffected, as the cam angles in the loop-forming portions of the non-linear cam were always less than the angles of the linear cams.

The needle butt performance and the fabric appearance were examined at knitting speeds much higher than commercial speeds to see which cam systems would provide increases in the rate of fabric production. Knitting trials with the  $45^{\circ}$  and  $55^{\circ}$  linear cams showed that the limiting speeds were approximately 160 r.p.m. and 120 r.p.m. respectively, as above these speeds fabric distortion occurred, holes appeared in the fabric and occasional needle butt breakage occurred.

With the non-linear cams (i.e.  $52^{\circ}$  and  $57^{\circ}$ ) and the  $55^{\circ}$  straight and curved cam, satisfactory knitted fabric was produced at 200 r.p.m. (i.e. three times commercial speed) with no needle butt breakage. The limiting speeds for the non-linear cams could not be obtained since the maximum safe speed for the other moving parts of the knitting machine was 200 r.p.m. However, calculations of the acceleration forces of the needles were used to show that speeds as high as 30 times commercial speed might be obtained with perfectly made non-linear cams, before needle butt breakage occurred.

From a comparison of the circumferential lengths of the various cam systems, it was concluded that the  $52^{\circ}$  and  $57^{\circ}$  non-linear cams and the  $55^{\circ}$  straight and curved cam could be used on commercial machines to provide two to three times the normal rate of fabric production. It was suggested that a high speed machine should be developed in order to establish the maximum speeds of the curved cam systems, and to determine whether the limiting speeds were due to poor fabric appearance, needle butt breakage or needle and hook failure.

The purpose of Phase II of this work, described in Chapter 6, was to determine whether reductions in the amplitude of vertical needle movement could be achieved, so that more feeders and hence increases in rate of fabric production could be obtained.

Warp knitting compound needles were modified and furnished with latch needle butts and a four feed compound cam system was fitted to a 1" diameter circular weft knitting machine - a machine which normally functions with a maximum of two feeders when latch needles are used.

The application of the modified compound needle to a circular weft knitting machine was successful, and satisfactory fabric was produced at commercial speeds. This work showed

that twice the commercial rate of fabric production could be achieved with modified compound needles.

Further reduction in the vertical amplitude was attempted by providing a lifting motion to the sinker. This involved the redesign of the sinker and modification to the sinker ring of a small diameter machine. The lifting motion of the sinker allowed loops to be produced with a cam system of half the normal amplitude. The inevitable lifting of the fabric at the knitting point, however, did provide a problem and to overcome this the sinkers require a further modification.

It is considered that, ultimately, the maximum rates of fabric production for a circular weft knitting machine will be achieved by a combination of non-linear cams, compound type needles and a lifting sinker motion.

A P P E N D I X



TABLE 15

Time (mins)	Torque (lbs ft)	Temperature (°C)
0	1.28	23.4
15	1.12	24.6
30	1.01	25.8
45	0.96	26.3
60	0.93	26.6
75	0.91	27.0
90	0.90	27.2
105	0.90	27.3
120	0.89	27.4
135	0.89	27.4
150	0.89	27.4

Room Temp 23.2°C

TABLE 16

Measured Stitch Length $l_a$ (ins)	G-ins $l_t$ -ins	Cam Systems					
		Linear		Non-Linear			St. & Curv.
		45°	55°	52°	57°	62°	55°
Slack - 0.196	G	0.125	0.123	0.101	0.111	0.116	0.114
	$l_t$	0.274	0.270	0.226	0.246	0.256	0.252
	§ RB	28.5	27.4	13.3	20.3	23.4	22.2
Medium - 0.176	G	0.105	0.101	0.089	0.096	0.100	0.101
	$l_t$	0.234	0.226	0.202	0.216	0.222	0.226
	§ RB	24.8	22.1	12.9	18.5	21.6	22.2
Tight - 0.156	G	0.085	0.079	0.077	0.082	0.084	0.088
	$l_t$	0.194	0.182	0.178	0.188	0.192	0.200
	§ RB	20.1	14.8	12.9	17.5	19.3	22.5

TABLE 17

Yarn Input Tension (gms)	Measured Stitch Length (ins); Cam Setting - Medium					
	Linear		Non-Linear			St. & Curv.
	45°	55°	52°	57°	62°	55°
2.5	0.196	0.197	0.181	0.184	0.185	0.187
5	0.176	0.176	0.176	0.176	0.176	0.176
10	0.160	0.154	0.168	0.169	0.167	0.166
15	0.151	0.143	0.163	0.165	0.162	0.159
20	0.144	0.133	0.159	0.161	0.156	0.151
25	0.140	0.131	-	-	0.153	0.146
% Robbing Back						
2.5	16.3	12.9	10.4	14.5	16.7	16.4
5	24.8	22.1	12.9	18.5	20.7	22.1
10	31.6	31.8	16.8	21.3	24.8	26.1
15	35.5	36.3	19.3	23.6	27.0	29.6
20	38.4	40.0	21.3	25.0	29.7	33.2
25	40.1	41.5	-	-	31.1	35.4

TABLE 18

NEEDLE											
$\theta = 60^\circ$		$\theta = 90^\circ$		$\theta = 120^\circ$		$\theta = 150^\circ$		$\theta = 180^\circ$			
$T_2/T_1$	$\log_e T_2/T_1$	$T_2/T_1$	$\log_e T_2/T_1$	$T_2/T_1$	$\log_e T_2/T_1$	$T_2/T_1$	$\log_e T_2/T_1$	$T_2/T_1$	$\log_e T_2/T_1$	$T_2/T_1$	$\log_e T_2/T_1$
7/5	0.3365	9/5	0.5878	11/5	0.7885	12/5	0.8755	13/5	0.9555		
14/10	0.3365	18/10	0.5878	21/10	0.7419	22/10	0.7885	23/10	0.8329		
20/15	0.2874	25/15	0.5104	29/15	0.6590	31/15	0.7242	32/15	0.7575		
28/20	0.3365	33/20	0.5008	36/20	0.5878	38/20	0.6419	40/20	0.6931		
SINKER											
8/5	0.470	11/5	0.7885	14/5	1.0296	16/5	1.1632	18/5	1.2809		
16/10	0.470	21/10	0.7419	25/10	0.9163	27/10	0.9933	31/10	1.1314		
23/15	0.4272	29/15	0.6590	34/15	0.8167	37/15	0.9014	41/15	1.0054		
31/20	0.4383	38/20	0.6419	43/20	0.7655	47/20	0.8544	52/20	0.9555		

TABLE 19

NEEDLE; R = 0.228 m/m								
$\theta = 120^\circ$			$\theta = 150^\circ$			$\theta = 180^\circ$		
$T_2/T_1$	$\log_e \log_e \frac{T_2}{T_1} - \log_e \theta$	$T_2/T_1$	$\log_e \log_e \frac{T_2}{T_1}$	$\log_e \log_e \frac{T_2}{T_1} - \log_e \theta$	$T_2/T_1$	$\log_e \log_e \frac{T_2}{T_1}$	$\log_e \log_e \frac{T_2}{T_1} - \log_e \theta$	$\log_e R/T_1$
11/5	- 0.9769	12/5	- 1.0953	- 1.1903	13/5	- 1.1903	- 3.0897	
21/10	- 1.0378	22/10	- 1.2001	- 1.3276	23/10	- 1.3276	- 3.7827	
29/15	- 1.1562	31/15	- 1.2901	- 1.4225	32/15	- 1.4225	- 4.1886	
36/20	- 1.2705	38/20	- 1.4057	- 1.5114	40/20	- 1.5114	- 4.4760	
SINKER; R = 0.1016 m/m								
14/5	- 0.7096	16/5	- 0.8114	- 0.8971	18/5	- 0.8971	- 3.8962	
25/10	- 0.8267	27/10	- 0.9691	- 1.0217	31/10	- 1.0217	- 4.5896	
34/15	- 0.9417	37/15	- 1.0663	- 1.1400	41/15	- 1.1400	- 4.9949	
43/20	- 1.0064	47/20	- 1.1197	- 1.1906	52/20	- 1.1906	- 5.2825	

TABLE 20

Knitting Element Positions	Yarn Input Tensions (Gms) - 57° Non-Linear	Robbing Back Tensions (gms)
T <sub>1</sub>	20	
S <sub>1</sub>	-	
N <sub>1</sub>	28.0	
S <sub>2</sub>	67.5	-
N <sub>2</sub>	116	-
S <sub>3</sub>	220	-
N <sub>3</sub>	334	295
S <sub>4</sub>	-	192
N <sub>4</sub> (K.P.)	-	98.8
S <sub>5</sub>	-	56.4
N <sub>5</sub>	-	22.3
S <sub>6</sub>	-	9.8
N <sub>6</sub>	-	2



TABLE 22

Speed (r.p.m.)	Measured Force Without Yarn (gms)					
	Linear		Non-Linear			St. & Curv.
	45°	55°	52°	57°	62°	55°
20	540	600	480	630	660	600
40	660	750	630	780	810	720
60	780	900	720	930	960	870
80	900	1020	840	1050	1110	960
100	960	1110	930	1140	1170	1020
120	990	1130	950	1150	1180	1040
Measured Force With Yarn (gms)						
20	780	840	750	990	870	870
40	900	990	900	1110	1050	1020
60	1020	1140	1020	1260	1170	1140
80	1110	1230	1140	1350	1320	1260
100	1200	1290	1230	1440	1380	1320
120	1210	1300	1250	1450	1400	1330



TABLE 23

Needles in 57° non-linear cam (Fig. 55)	P <sub>F</sub> (gm)	m $\ddot{y}$ (gms) 25 r.p.m	S <sub>3</sub> or S <sub>4</sub> (gms)	m $\ddot{y}$ (gms) 145 r.p.m.	S <sub>3</sub> or S <sub>4</sub> or S <sub>5</sub> (gms)
(a)					
1	10	0.32	1.03	11.65	2.19
2	10	0.65	2.06	23.30	6.43
A 3	10	0.76	4.23	27.20	14.63
4	10	0.65	8.25	23.30	25.85
5	10	0.43	11.89	15.50	29.10
6	10	0.11	14.05	3.88	19.29
(b)					
7	10	0.32	16.64	11.65	- 1.47
8	10	0.71	10.59	25.60	- 11.40
D 9	10	1.04	6.94	37.30	- 12.90
10	10	1.19	2.55	42.70	- 2.46
(c)					
11	10	1.10	3.22	39.60	14.40
12	10	0.95	5.54	34.2	22.36
13	10	0.82	10.19	29.5	37.22
A 14	10	0.65	14.80	23.3	46.30
15	10	0.39	22.65	14.0	52.30
16	10	0.11	24.26	3.88	33.31
(d)					
17	10	0.22	21.33	7.76	4.90
18	10	0.54	20.62	19.40	2.40
D 19	10	0.67	16.05	24.1	- 12.6
20	10	0.87	12.70	31.10	- 18.8
21	10	0.91	10.36	32.60	- 16.5
22	10	1.02	5.70	36.5	- 9.6
(e)					
23	10	1.04	1.12	37.3	4.78
24	10	0.89	3.16	31.8	12.11
A 25	10	0.76	8.34	27.2	28.85
26	10	0.45	11.92	16.3	30.0
27	10	0.11	17.39	3.88	23.86

Continued

TABLE 23 (Continued)

Needles in 57° non-linear cam (Fig. 55)	P <sub>F</sub> (gm)	mȳ (gms) 25 r.p.m	S <sub>3</sub> or S <sub>4</sub> (gms)	mȳ (gms) 145 r.p.m.	S <sub>3</sub> or S <sub>4</sub> or S <sub>5</sub> (gms)
(f)					
28	10	0.22	13.60	7.76	3.1
29	10	0.58	10.73	21.0	- 8.05
D 30	10	0.87	7.05	31.1	- 10.00
31	10	0.87	3.62	31.1	- 3.50
32	10	0.43	1.85	15.5	- 0.42
(g)					

A = Acceleration

D = Deceleration

$\Sigma(S_3 + S_4)$

= 325

$\Sigma(S_3 + S_4 + S_5)$

= 302

TABLE 24

Speed (r.p.m.)	Measured Force Without Yarn (gms) 57° Non-Linear Cam
20	630
40	780
60	930
80	1050
100	1140
120	1150
140	1150
160	1140
180	1150
200	1150

TABLE 25

Needles (Fig. 55)	$\theta^\circ$	Horizontal Force $S_1$ (gms) - $P_F = 10$ gms			
		$\mu = 0.1$	$\mu = 0.2$	$\mu = 0.25$	$\mu = 0.3$
1	0	1.01	2.08	2.66	3.3
2	5	1.93	3.11	3.78	4.53
3	15	3.93	5.50	6.45	7.56
4	30	7.75	10.65	12.79	15.5
5	40	11.40	16.70	21.00	28.2
6	45	13.90	21.40	28.40	41.7
7	50	17.20	28.80	40.10	76.0
8	40	11.40	16.70	21.00	28.2
9	30	7.75	10.65	12.79	15.5
10	10	2.90	4.24	5.00	6.0
11	10	2.90	4.24	5.00	6.0
12	20	5.06	6.94	8.15	9.46
13	35	9.42	13.25	16.20	20.4
14	45	13.90	21.40	28.40	41.7
15	55	21.80	42.00	75.00	324.0
16	57	24.00	50.40	107.0	"
17	55	21.80	42.00	75.0	324.0
18	55	21.80	42.00	75.0	324.0
19	50	17.20	28.80	40.10	76.0
20	45	13.90	21.40	28.40	41.7
21	40	11.40	16.70	21.00	28.2
22	25	6.34	8.67	10.15	12.2
23	0	1.01	2.08	2.66	3.3
24	10	2.90	4.24	5.00	6.0
25	30	7.75	10.65	12.80	15.5
26	40	11.40	16.70	21.00	28.2
27	50	17.20	28.80	40.10	76.0
28	45	13.90	21.40	28.40	41.7
29	40	11.40	16.70	21.00	28.2
30	30	7.75	10.65	12.80	15.5
31	15	3.93	5.50	6.45	7.56
32	5	1.93	4.53	3.78	4.53
$\Sigma S_1 =$		328	539	797.36	"

TABLE 26

Speed (r.p.m.)	Measured Force (gms); 45° Linear Cam		
	Ord. Oil	No Oil	Thick Oil
20	540	880	1400
40	670	950	1720
60	800	890	1810
80	920	830	1820
100	980	830	-
120	1000	810	-

TABLE 27

Loop-Forming Needle Position (Fig. 45)	Actual Needle Position (Fig. 55)	57° Non-Linear Cam	
		$P_{\theta}$ (gms)	$S_5$ (gms)
$N_1$	20	12.0	16.7
$N_2$	21	69.1	78.6
$N_3$	22	198.3	129.0
$N_4$ (K.P.)	23 (K.P.)	65.7	6.6
$N_5$	24	8.5	2.0

$\Sigma S_5 = 232.9$

TABLE 28

Yarn Input Tension (gms)	Torque (lbs ft) - Medium Cam Setting					
	Linear		Non-Linear			St. & Curv.
	45°	55°	52°	57°	62°	55°
5	1.55	1.56	1.55	1.57	1.56	1.55
10	1.57	1.58	1.58	1.58	1.57	1.56
15	1.61	1.60	1.60	1.59	1.58	1.59
20	1.66	1.64	1.62	1.60	1.58	1.62
25	1.71	1.69	-	-	1.59	1.67

TABLE 29

Take-Down Tension (lbs)	Measured Stitch Length (ins)					
	Linear		Non-Linear			St. & Curv.
	45°	55°	52°	57°	62°	55°
2	0.176	0.176	0.176	0.176	0.176	0.176
4	0.179	0.180	0.180	0.179	0.178	0.177
6	0.182	0.182	0.181	0.179	0.179	0.176
8	0.183	0.185	0.183	0.180	0.182	0.175
10	0.186	0.186	0.185	0.183	0.184	0.176
Torque (lbs ft)						
2	1.55	1.57	1.55	1.57	1.55	1.55
4	1.57	1.60	1.58	1.59	1.59	1.56
6	1.59	1.63	1.60	1.62	1.60	1.58
8	1.63	1.65	1.64	1.65	1.64	1.62
10	1.66	1.68	1.68	1.67	1.66	1.64



TABLE 30

Yarn Count (Wor- sted)	Measured Stitch Length (ins) - Slack Setting					
	Linear		Non-Linear			St.&Curv.
	45°	55°	52°	57°	62°	55°
1/32	0.199	0.195	0.198	0.195	0.197	0.197
1/28	0.196	0.196	0.196	0.196	0.196	0.196
1/24	0.191	0.193	0.195	0.193	0.199	0.187
2/32	0.191	0.194	0.196	0.193	0.203	0.184
2/28	0.192	0.195	0.198	0.194	0.204	0.179
2/24	0.194	0.192	0.196	-	0.203	-

Cover Factor (C.F.) & Torque (T) lbs.ft.

Y.C.	C.F.	T.	C.F.	T.	C.F.	T.	C.F.	T.	C.F.	T.	C.F.	T.
1/32	0.89	1.51	0.91	1.50	0.89	1.51	0.91	1.52	0.90	1.50	0.90	1.51
1/28	0.97	1.52	0.97	1.53	0.97	1.53	0.97	1.55	0.97	1.53	0.97	1.53
1/24	1.07	1.56	1.06	1.52	1.05	1.56	1.06	1.61	1.02	1.56	1.09	1.58
2/32	1.31	1.63	1.29	1.54	1.28	1.60	1.30	1.64	1.21	1.58	1.36	1.63
2/28	1.39	1.67	1.37	1.56	1.35	1.62	1.38	1.66	1.32	1.62	1.48	1.76
2/24	1.49	1.91	1.51	1.80	1.48	1.86	-	1.95	1.42	1.73	-	2.3

TABLE 31

Speed (r.p.m.)	Measured Stitch Length (ins)					
	Linear		Non-Linear			St. & Curv.
	45°	55°	52°	57°	62°	55°
40	0.172	0.168	0.176	0.175	0.178	0.173
60	0.176	0.176	0.176	0.176	0.176	0.176
80	0.183	0.185	0.177	0.174	0.177	0.179
100	0.187	0.189	0.176	0.175	0.177	0.181
120	0.187	0.189	0.177	0.176	0.178	0.183
140	0.186	-	0.177	0.175	.	0.184
160	0.188	-	0.176	0.174	.	0.185
180	-	-	0.177	0.175	.	0.187
200	-	-	0.176	0.175	0.180	0.188

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