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The Transmission of Power to High Speed Belts

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BELTCON 5

INTRODUCTION

Reading the past papers presented at these Beltcon Symposia I am surprised that two facets have received little attention. Firstly that today's conveyor designers are disadvantaged by having little or no physical contact with their creations. For one hundred years from the mid-nineteenth century designers and users were able to ride their conveyors and the majority did just that. The belt speeds were rarely greater than 1,5m/s and the conveyors had relatively short centres. My own initiators insisted that I ride along the coalface to the pit bottom so that I could 'feel' belt stability plus the jiggling and jerks. The majority of other transport systems, like ships, planes and trains have retained their human bond, while Rocketry has actually introduced it. Perhaps this 'lack' has something to do with our industry's rather tardy development when compared with other modes. But, let me be absolutely clear, this in no way detracts from the achievements of those magnificent young men with their computer machines and the 3D imagery of transient forces.

Secondly, and my reason for this paper, is that we appear to have avoided the weakest link, certainly the one we confidently apply, but in truth the mechanics of which we still know so little.

The early pioneers, like Evans[1] in 1795 and Westmacott and Lyster[2] in 1866 followed the principles then being used to transmit power by leather belts from a prime-mover to the new

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'industrial revolution' machines at speeds of 30 metres per second. Although I have copies of the Cornhouses into which these very early units were installed, the archives have not so far revealed their calculations.

The following list of some of the subsequent historical evidence helps in reaching a reasonable interpretation.

1. By 1900 the basic formula $T_1/T_2 = e^{\mu\theta}$ (its derivation is shown in Appendix A) was fully accepted with practical coefficients of friction of 0,25 and 0,35 for leather on bare or lagged drive pulleys respectively. These figures were known to be conservative but effective under both wet and dry conditions when starting or running underload.

2. 1906 saw the introduction in the U.K. of underground conveyors in coal mines by a mining engineer Richard Sutcliffe.[3] Because the 'handgot' system lacked any control of the loading rates, Sutcliffe provided a spring loaded Mangle Pressure Pulley to eliminate belt slip occurring with overloads. (Appendix B illustrates the belt reeving).

3. 1908/9 were the years of Wilfred Lewis and Carl Barth's[4] major professional achievements, as published by the American Society of Mechanical Engineers. The results of these experiments are illustrated in Figure 1. The findings were:-

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3.1 An indication of the increase in transmitted force compared with:

3.1.1 Varying amounts of Creep and Slip.

3.1.2 Varying ratios of T_1/T_2.

3.1.3 Varying values of T_2 and hence T_1.

3.2 By using the formula $\frac{T_1}{T_2} = e^{\mu\theta}$ and since they maintained an angle of total wrap equal to 180° they established a range of coefficients of friction (μ) totalling 0,25 to 0,72 for leather on a bare iron pulley.

4. In 1926 R F Jones,[5] a Professor of Ohio State University, described to the American Society of Mechanical Engineers how he had measured with a stroboscope the varying speeds of creep occurring on a drive pulley.

4.1 In addition he established the maximum and zero speed points on the arc of belt contact. The findings are summarized in Figure 2:

4.1.1 The maximum speed of creep identified at the point where the belt leaves the pulley.

4.1.2 The minimum speed of creep being at a point nearer the first point of contact of the oncoming belt with the pulley. This point varying with the power transmitted.

4.2 Jones deduced, using the same formula as Lewis and Barth eighteen years previously, that:

4.2.1 The coefficients of friction attainable on a bare Cast Iron Pulley were:

4.2.1.1 0,99 for Leather, and

4.2.2.2 0.44 for a Rubber Covered Fabric Belt.

4.3 There was inconclusive discussion on 'lag' (Hysteresis) between stress and strain at high speed (+40m/sec).

5. In the following year, 1928, H W Swift[6] acknowledging the work of Lewis and Barth, but also of Friedrichs in 1915, postulated "Why friction, which permits creep, is unable to prevent slip"?

5.1 Much of his experimental work confirmed previous evidence but expanded the comparisons of percentage slip against tension differences for many different types of belting.

5.1.1 In one case using a rubber covered belt after a period of bodily slip there appeared to be a permanent loss of the frictional properties. He did observe a peculiar wear characteristic of 'small rolls of rubber'.

5.1.2 There was much discussion on two principles, namely:
Loss of belt speed while passing over a drive pulley was the result of:

5.1.2.1 "A Variable Coefficient of Friction which rises from a very low value as the speed of slipping increases", or

5.1.2.2 "the elastic properties of the belt cause creep."

6. During the period 1906 to 1935 Conveyor Engineers exploited the θ component of $\frac{T_1}{T_2} = e^{\mu\theta}$ since they lacked understanding of the μ behaviour. Hence John Sheppard's Three Pulley Geared Drive appeared and subsequently literally thousands of such machines powered underground conveyor networks. Two such machines were installed here in South Africa at the Torbonite Mine in 1950. Appendix B illustrates the belt reeving.

7. In 1949 Desmond Sutcliffe,[7] grandson of the U.K. pioneer in underground belt conveyors initiated a study of friction on drive pulleys by the recently formed Development Department in his family's business. Don Clark[8] from Leeds University joined in the early investigations which terminated upon Desmond's death in 1950, and Clark moved on to the National Coal Board's Scientific Department. Retaining his enthusiasm for the project, his work was finally published in 1956.

7.1 The basic experimental principle was to insert strain gauges between the plies of a belt and record the results as the belt was moved around a pulley with driving or driven characteristics. A few tests were conducted successfully with a belt speed of 1m/sec. The following were identified:

7.1.1 Arcs of changing and constant tension for varying ratios $\frac{T_1}{T_2}$ (see figures 3 and 4).

7.1.2 Details of the strain pattern over the arc of changing tension. Figure 5 shows subsequent interpolation into Newtons/25.4mm against location on the pulley.

7.1.2.1 Table 1 shows the Tension at each 10° intermediate arc and the basic horizontal and vertical components (exactly as used in deriving the basic formula This illustrates that the site measurements display a variation in coefficient of friction over this 110° arc of changing tension, the average of which closely coincides with that determined from the basic formula so too the intermediate figures are an average over 10° . Note carefully that the creep speed is a maximum at 180° just when the belt is leaving the pulley and is zero at 70° or the junction with the arc of constant tension. See Figure 7.

7.1.2.2 Since the belt is contracting or creeping back over this arc, the hysteresis effect must be considered and Figure 6 illustrates the relationship against a belt being stretched.

7.1.3 Clark's work illustrated not only the specifics in the previous paragraphs, but confirmed the findings of the earlier experiments by an entirely different procedure. In addition, he illustrated very clearly the Flexing Arc characteristics shown in Figure 7.

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7.2 An understandable explanation of these mechanics was still not available so the majority of Conveyor Engineers still used coefficients of 0,25 and 0,35 in the basic formula and continued to be deceived by ignorance in what was happening to the belt and the pulley. Figure 8 shows how much.

8. Around 1960, the independently driven Dual Drive Pulley System was well established (see Appendix B). The need to provide creep absorbing measures for that passing to the secondary drive pulley as slip enhanced the need for high speed slipping fluid couplings with Squirrel Cage and the Tri-slot variety Motors.

8.1 Interestingly enough this was the entry to the era of 'aborted start' accidents resulting in major damage. In at least two accidents in the U.S.A. it was determined that the belt momentarily lost contact with the drive pulleys and, when slapping back again, the inertia impulse passing between the belt and pulley/s indicated a very high friction level and a considerable speed difference. One of the more recent occurrences resulted in all four of the power pack slow speed couplings breaking.

9. During the late 1960's and thereafter interest in motor racing and passenger cars mushroomed. This lead to more

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research and investigation into the performance of tyres. I am sure many conveyor engineers have recognised the advantage of using analogies to help solve puzzling situations. I certainly have, and my favourites are:

9.1 the motor car lying on its roof with the potholed road travelling over the wheels. How close have we come from Ford's Model T to Citroen's suspension, and

9.2 the railway train coupling clatter as transient shocks rebound. I asked a Railway Company's Train Dynamics Engineer for his comments on our common problem. "Ah" said he, "but we have the driver's feel". I found at this time that my favourite authors were people like E Southern and Desmond Moore[9] in order to understand more about the physics of tyre mechanics.

9.3 Moore in his book 'The Friction of Pneumatic Tyres' after suggesting that, some of Da Vinci, Amontons and Coulombs work and the Classic Laws of Friction, that evolved therefrom needed adjustment and particularly so when considering rubber and elastomers. He goes on to explain the interaction between a tyre and the road's surface. I quote:

"The contact area between an elastomer or rubber and a rough support surface is characterised by a draping of the elastomer about <u>individual</u> asperities ..."

He goes on:

"If a force F is now applied tangentially to the upper surface, relative motion of the frictional interface takes the form of a 'flowing' action as the elastomer conforms to the asperities of the base. A frictional force equal in magnitude and opposite in direction to the applied force is generated at the sliding interface and it includes both adhesional and hysteresis components, thus $F = F_A + F_H$ ".

(both the latter are affected by the normal load, R in our case the Radial Load)

then $\mu = \mu_A + \mu_H$ (being the coefficients).

9.3.1 The relevant characteristics are:

9.3.1.1 That is it invariably found that there are part Adhesion and part Hysteresis Coefficients, the proportions being μ_A and μ_H

9.3.1.2 μ_A peaks at low relative speeds

9.3.1.3 μ_H peaks at high relative speeds

9.3.1.4 μA troughs on wet rough surfaces

9.3.1.5 μH troughs on smooth dry rigid surfaces

The analogy appears good with our drive pulley an asperity.

9.3.2 Hysteresis friction is frequently referred to, by Moore as a 'Bulk Effect' meaning there is a 'bunching' or distortion of the elastomer just prior to the start of 'relative movement'.

9.3.2.1 This characteristic has been well publicised when considering 'indentation resistance' and despite knowing it related to shape factors I had never been able to physically measure it until investigating a series of major cuts in the top cover of a Steel Cord Belt. During the autopsy the markings or striations on the faces of the cuts were photographed for identification purposes. One set represented longitudinal striations compatible from previous exper-perience as representing 'fixed object' cutting. The others were a repetative 'integral sign', as illustrated in Figure 9. The murder weapon was eventually identified and reproduced by analysis (Figure 9 was part of this exercise). Ultimately, although the scene of the crime had been wiped almost clean, the exactly dimensioned weapons were unearthed. In Figure 9 the top curve of the integral sign is in fact caused by the 'elastomer distortion' and the dimensions were measurable.

10. An interpretation of this historical evidence. (Observe Figure 7).

10.1 Unquestionably we have a relative movement of a driven belt over the arc of changing tension.

10.2 This movement ceases at the junction of the changing and constant tension arcs. Therefore the speed varies, so too does the apparent coefficient of friction as determined from Clark's experimental measurement over the arc of changing tension.

10.3 If you accept Moore's characteristics (in paragraph 9.3.1, then the coefficient of friction has:

10.3.1 At the T_2 departure of the belt from the pulley a High μ_H derated for a smooth dry surface and Low μ_A under a small R

10.3.2 At the junction of the changing and constant tension arcs

A low μ_H derated for a smooth dry surface and a low μ_A under a high R.

10.3.3 The interpretation is that where there is relative movement such as creep over a defined arc (i.e. not a total slip of the belt over the whole contact surface of the pulley). The coefficient of friction is in fact varying due to many external causes resulting in a slip/stick type pattern of changes. This confirms that by using a Static Condition Formula $T_1/T_2 = e^{\mu\theta}$ determination of the coefficient of friction (μ) will in fact only be an average.

10.3.4. Consider a Dual Drive Horizontal Conveyor 2000 metres centres at a belt speed of 3,5m/sec under full load.

10.3.4.1 The troughed belt has a PEAK RATE OF TENSION INCREASE over the troughed portion equal to ± 1 BELT KILOWATT PER SECOND.

10.3.4.2 The belt is then subjected to a continuous PEAK RATE OF TENSION DECREASE over the arc of changing tension on the drive pulleys equating on each to ± 840 BELT KILOWATTS PER SECOND.

10.3.4.3 Compare this with the critical limit we found practical on underground installations of similar dimensions and powering to control transient vibration effects: ± 14 BKW PER SECOND. Many users have experienced the power of large sources of high rates of changing tension.

10.3.5 During the past decade whilst conducting 'perambulatory tutorials' over some 150 kilometres of conveyors per year, the records show a clear increase in poor belt tracking stability emanating from the drives. The majority of this results from 'cleanside belt creep' wearing both, the pulley surface and the bottom cover of the belt. Instances of excessive wear as referred to by Swift in 1928 (paragraph 5.1.1) have been observed here where higher than normal T_2 tensions have been used.

10.3.6 Until much more effective research is completed, the possibility of being more precise is nil. However, the need is to design out the obvious problem areas.

11. <u>The Future</u>

The requirements appear to be for lower capital and operating costs suggesting longer centres and high speed conveyors.

11.1 The foregoing evidence and an interpretation suggests that the transfer of power be conducted over a proportionally greater length. (See Figure 10).

11.2 The multiple drive pulley systems effectively reduced the cost of the belt by utilizing a lower slack side tension. Step two of this development is to use an intermediate drive system longitudinally. Mr Page's[10] concept in 1919 has now been in use for a decade, with one successful unit in South Africa so far.

11.2.1 Further stages of development are:

11.2.1.1 To design the carrying belt for principly longitudinal flexing and the Intermediate Drive Belts, narrower than the carrying belts and having a formal tread pattern, constructed for transverse flexing.

11.2.1.2 Pitching of the intermediate drives not only to reduce the carrying belt tensile rating but to minimise transient vibrations.

11.2.1.3 Providing multiple, easily mounted and dismountable, power packs on each intermediate drive. Not only for low inventory purposes but to increase reliability and availability of the system where one power pack failure can be absorbed by the others and the belt without a stop.

11.2.1.4 The photograph (Appendix C) taken in1960 was the first step in a programme to increase inherent belt stability. It did, in addition, control high speed payload trajectories very effectively. This profile is in fact similar to a much enlarged drive pulley with the inherently large distance over which power transfer can be transferred.

R E F E R E N C E S

1.	O Evans	American who published a 'Millers Guide' in 1795 describing a very early belt conveyor.
2.	P Westmacott & G Lyster	U K Engineers at Liverpool Docks in 1865 experimented with wooden idler rolls.
3.	Richard Sutcliffe	U K pioneer mining engineer provided the first underground colliery conveyors.
4.	Wilfred Lewis & Carl Barth	1908/9 presented a paper to the ASME on transmission belts.
5.	R F Jones	Professor of Ohio State University in a paper to the ASME in 1926 described measuring of creep speeds with a stroboscope.
6.	H W Swift	in 1927 presented a paper to the Institution of Mechanical Engineers, London, entitled 'Power Transmission by Belts'.
7.	Desmond Sutcliffe	Grandson of R Sutcliffe instigator of investigation into friction on drive pulleys.
8.	Don Clark	ex Leeds University, Messrs Richard Sutcliffe Ltd and N.C.B. Scientific Department. Experiments into Stress and Strains imposed on Multi-Ply Conveyor Belting by Flexing 1956.
9.	Desmond Moore	author of 'The Friction of Pneumatic Tyres' 1975.
10.	Mr Page	1919 patentee of a Conveyor Belt with Auxiliary Drive Belts Patent No. 1,313,111.

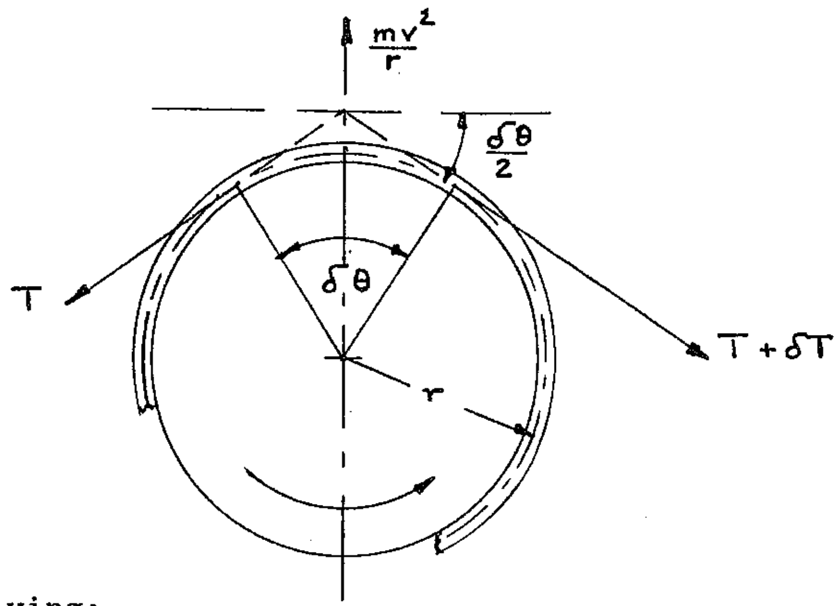
A T T A C H M E N T S

(In order of reference in the papers)

APPENDIX A	Derivation of Basic Formula.
APPENDIX B	Conveyor Drive Configurations.
FIGURE 1	A Lewis - Experimental Results 1909
FIGURE 2	R F Jones - Experimental Results of measuring creep speed 1926.
FIGURE 3	D Clark's Pattern of changing tension arcs 1955.
FIGURE 4	D Clark's Constant arc of changing tension but varying angles of total wrap 1955.
FIGURE 5	D Clark's Interpolated tension figures over an arc of changing tension ± 1955.
TABLE 1	D Clark's Experimental tension figures with calculated Coefficients of Friction.
FIGURE 6	Hysteresis Relationship Curves.
FIGURE 7	Diagramatic Interpretation of Current evidence.
FIGURE 8	Comparison of Ratios $\frac{T_1}{T_2}$ v Arc of Changing Tensions.
FIGURE 9	Analysis of Cutting Weapon leaving an Integral Shaped Striation.
FIGURE 10	Alternative Concept for very long high speed conveyor systems.
PHOTOGRAPH	illustrating inherent stability for a conveyor.

BASIC FORMULA AND ITS DERIVATION

1. The currently used method of determining the required relationship between the Tension, Angle of Wrap and Coefficient of Friction is from an assumed stationary system (belt relative to the pulley) or an inelastic belt, derived by resolving Radially and Tangentially.



Resolving:

1.1 Radially . where R is the resultant force

$$R = T \sin\frac{\delta\theta}{2} + (T + \delta T)\sin\frac{\delta\theta}{2} - \frac{mv^2}{r}$$

1.1.1 The centrifugal force is normally considered to be insignificant since m is small and v is small.

1.1.2 Where $\sin\frac{\delta\theta}{2} \simeq \frac{\delta\theta}{2}$ as $\delta\theta \to 0$ and $\delta T \frac{\delta\theta}{2}$ is negligible

Then

$R = T\,\delta\theta$ (if the mass and/or velocity is high then:- $R = \left[T - \frac{mv^2}{r}\right]\delta\theta$)

1.2 <u>Tangentially</u>

$$T\cos\frac{\delta\theta}{2} + \mu R = (T+\delta T)\cos\frac{\delta\theta}{2}$$

1.2.1 $\cos\frac{\delta\theta}{2} \simeq 1$ as $\theta \to 0$

Then $T + \mu R = T + \delta T$

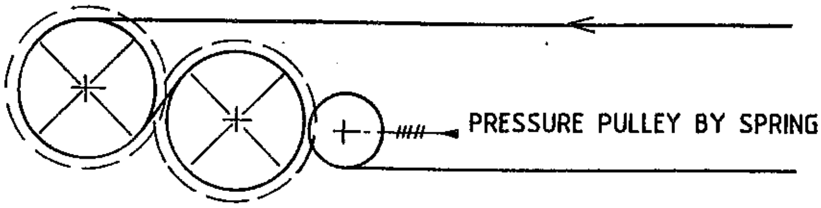
and $\mu R = \delta T$

$$\text{Then } \int_0^\theta \mu\,\delta\theta = \int_{T_2}^{T_1} \frac{\delta T}{T} \quad \text{or} \int_{T_2}^{T_1} \frac{\delta T}{T - \frac{mv^2}{r}}$$

$$\text{and } \frac{T_1}{T_2} = e^{\mu\theta} \quad \text{or } \frac{T_1 - \frac{mv^2}{r}}{T_2 - \frac{mv^2}{r}} = e^{\mu\theta}$$

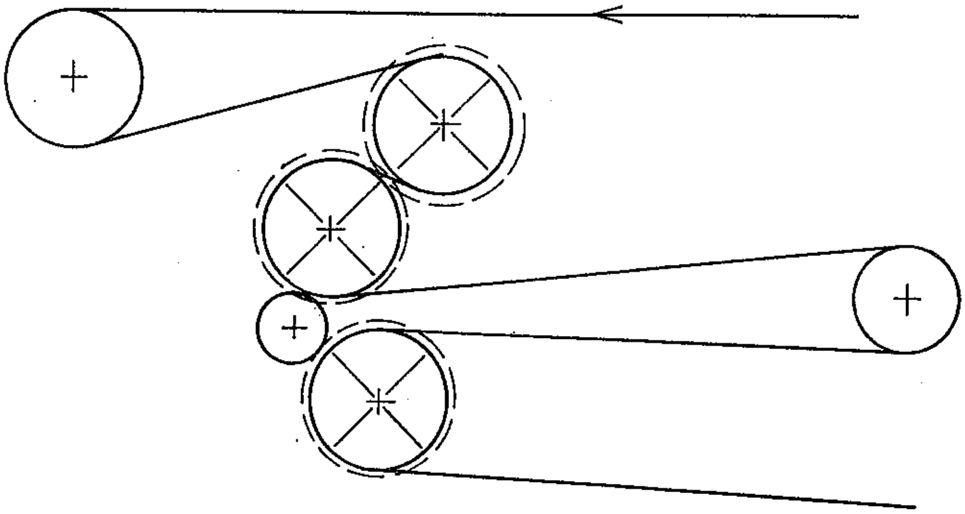
CONVEYOR DRIVE CONFIGURATIONS

1906



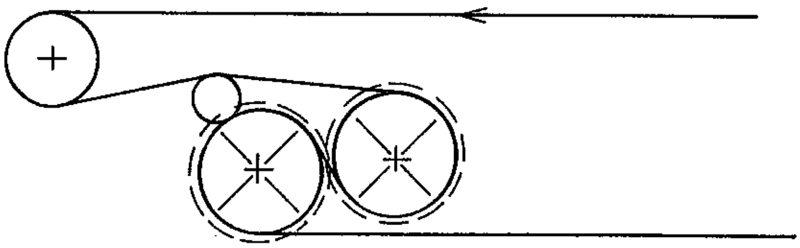
RICHARD SUTCLIFFE'S LONGWALL FACE CONVEYOR
BRITISH PATENE 11156/05.

1935



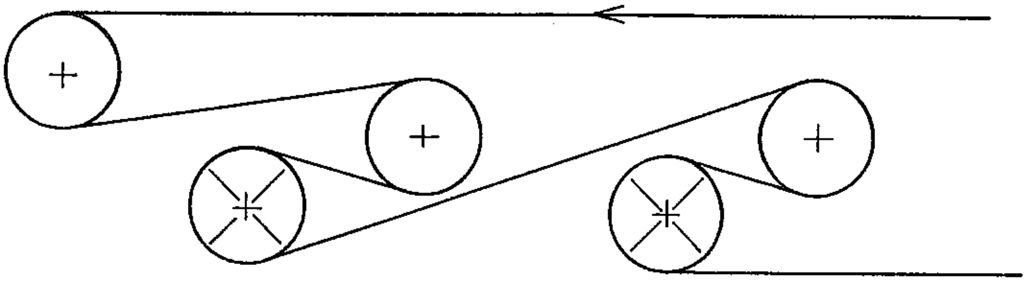
JOHN SHEPPARD'S TRIPLE PULLEY GEARED UNIT

1949



GEARED TANDEM DRIVE WITH AUTOMATIC PRESSURE PULLEY

1960



INDEPENDENTLY DRIVEN DUAL DRIVE

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FIGURE 1

W.LEWIS'S EXPERIMENTAL RESULTS AS PRESENTED BY CARL BARTH
AT THE A.S.M.E. TRANSACTION IN 1909
CONVERTED TO THE EQUIVALENT lbs / INCH / PLY.

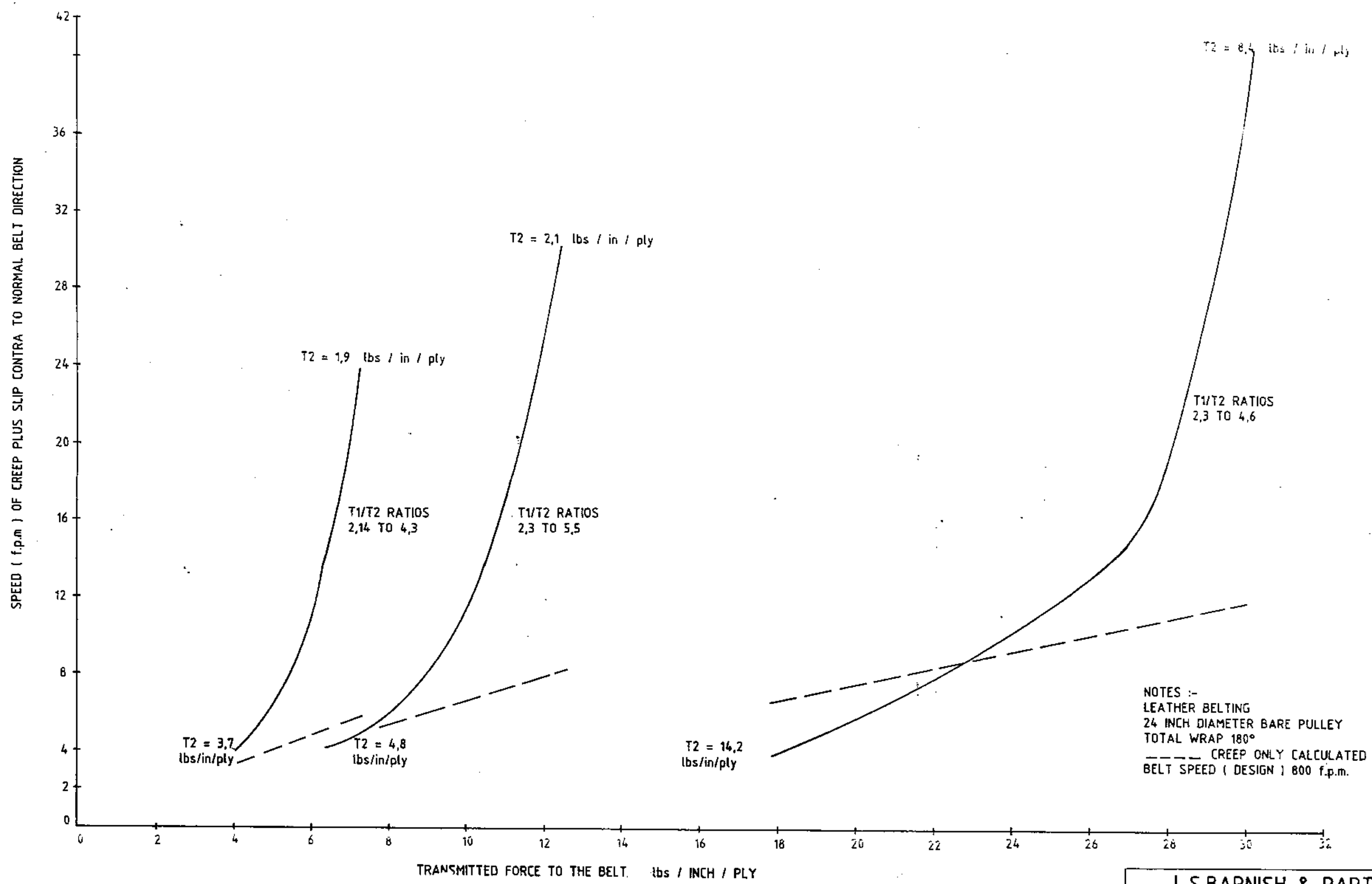

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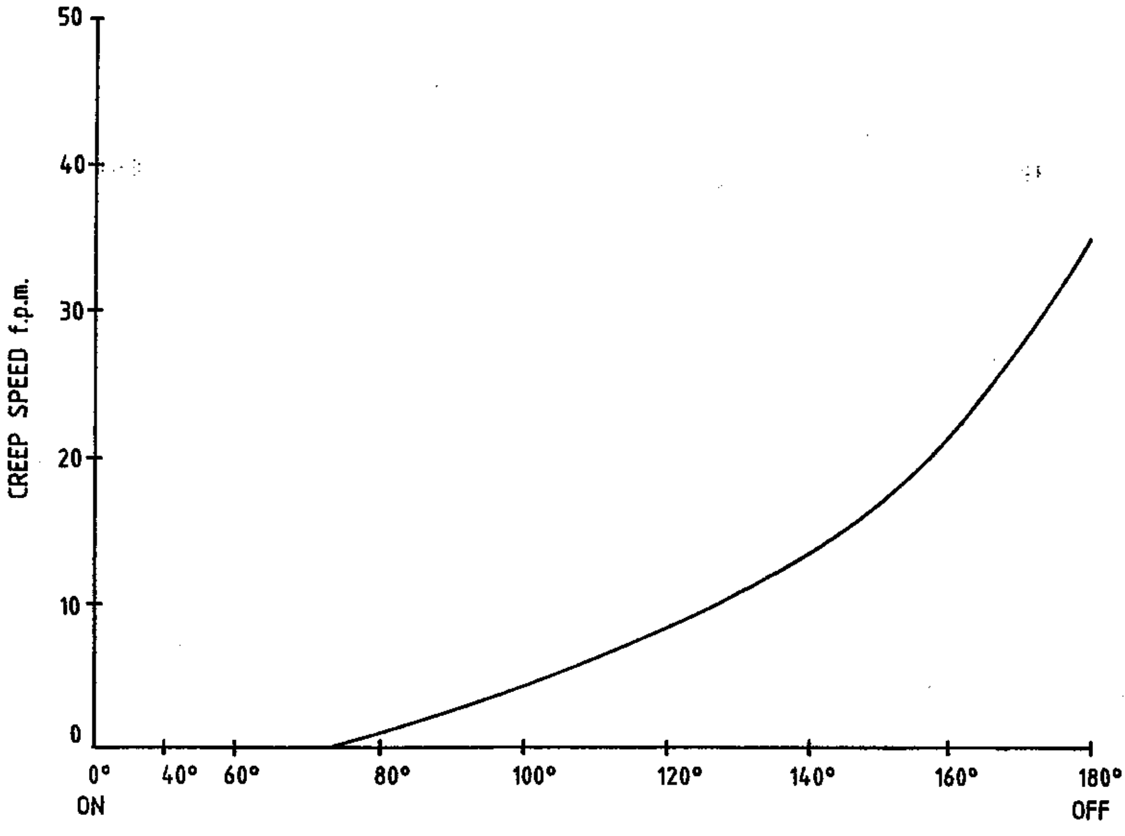
FIGURE 2

R.F.JONES'S EXPERIMENTS MEASURING CREEP SPEED AND LOCATION 1926



ANGLE OF WRAP 180°
24 INCH PULLEY WITH A BELT SPEED OF 2000 f.p.m.

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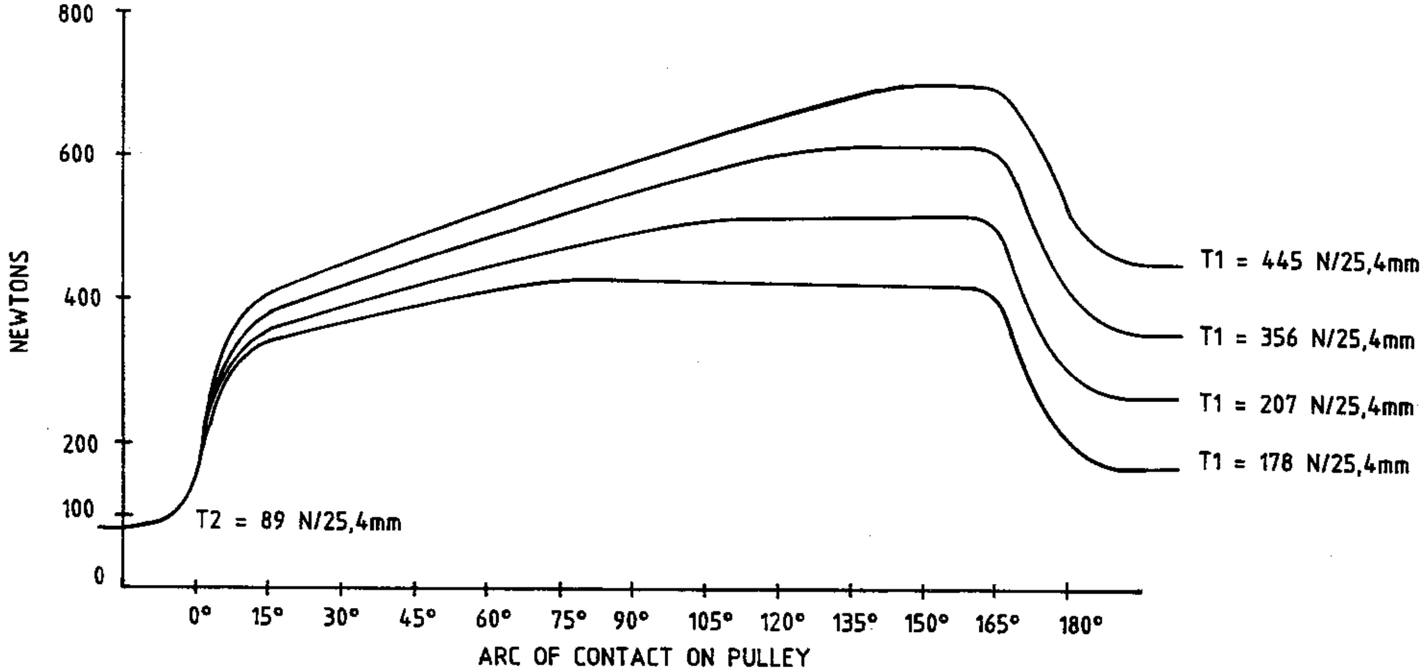
FIGURE 3



PATTERN OF ARC OF CHANGING TENSION FOR VARYING T1/T2 RATIOS
DATA FROM OUTER GAUGE LYING BETWEEN 4TH AND 5TH PLY OF A 5 PLY
32 OZ COTTON DUCK BELT WITH RUBBER COVERS 24,5mm WIDE
FROM CLARK'S EXPERIMENTS ±1955

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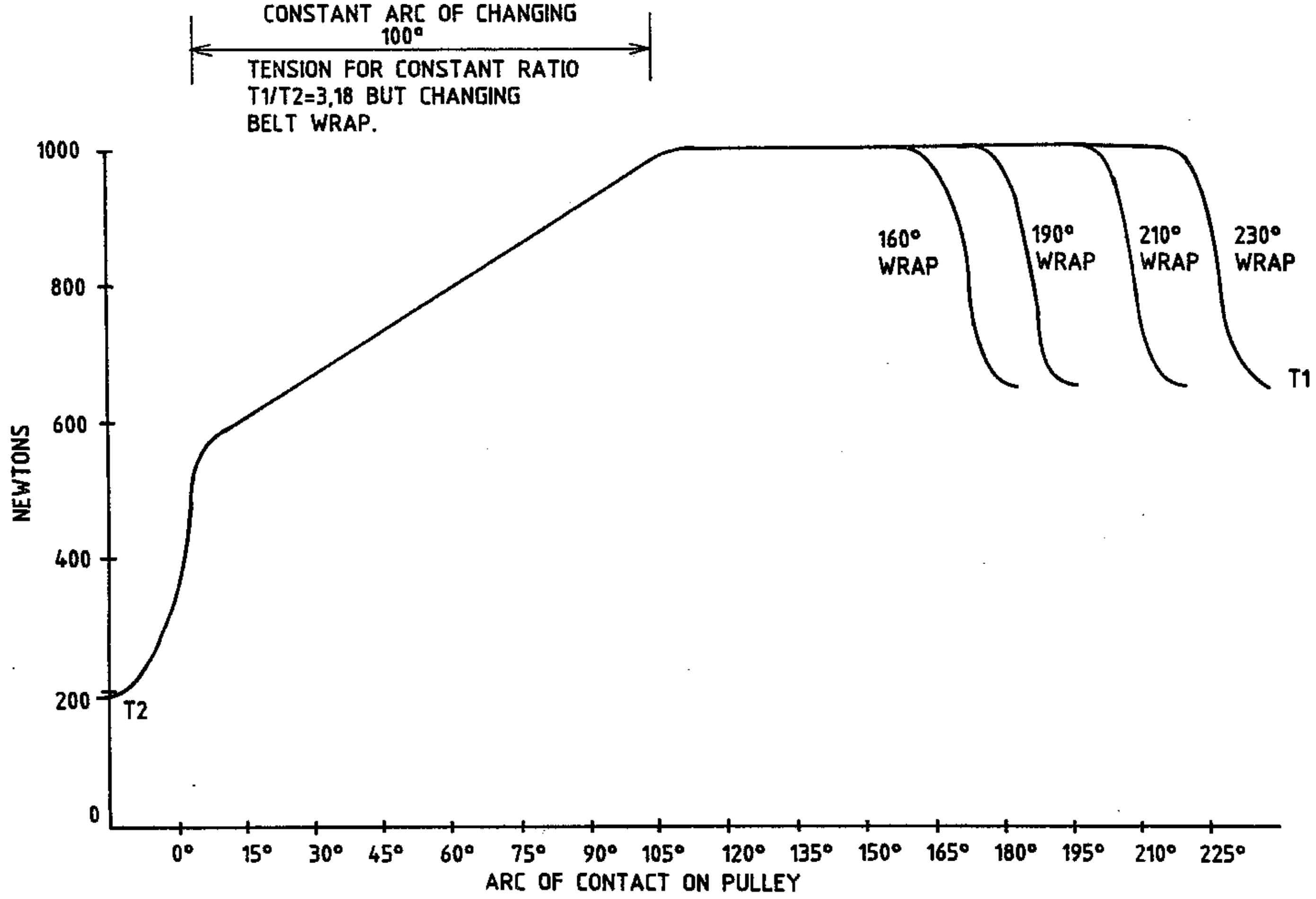
FIGURE 4



DATA FROM OUTER GAUGE LYING BETWEEN 4TH AND 5TH PLY OF A 5 PLY
32 OZ COTTON DUCK BELT WITH RUBBER COVERS 24,5mm WIDE
FROM CLARK'S EXPERIMENTS ±1955

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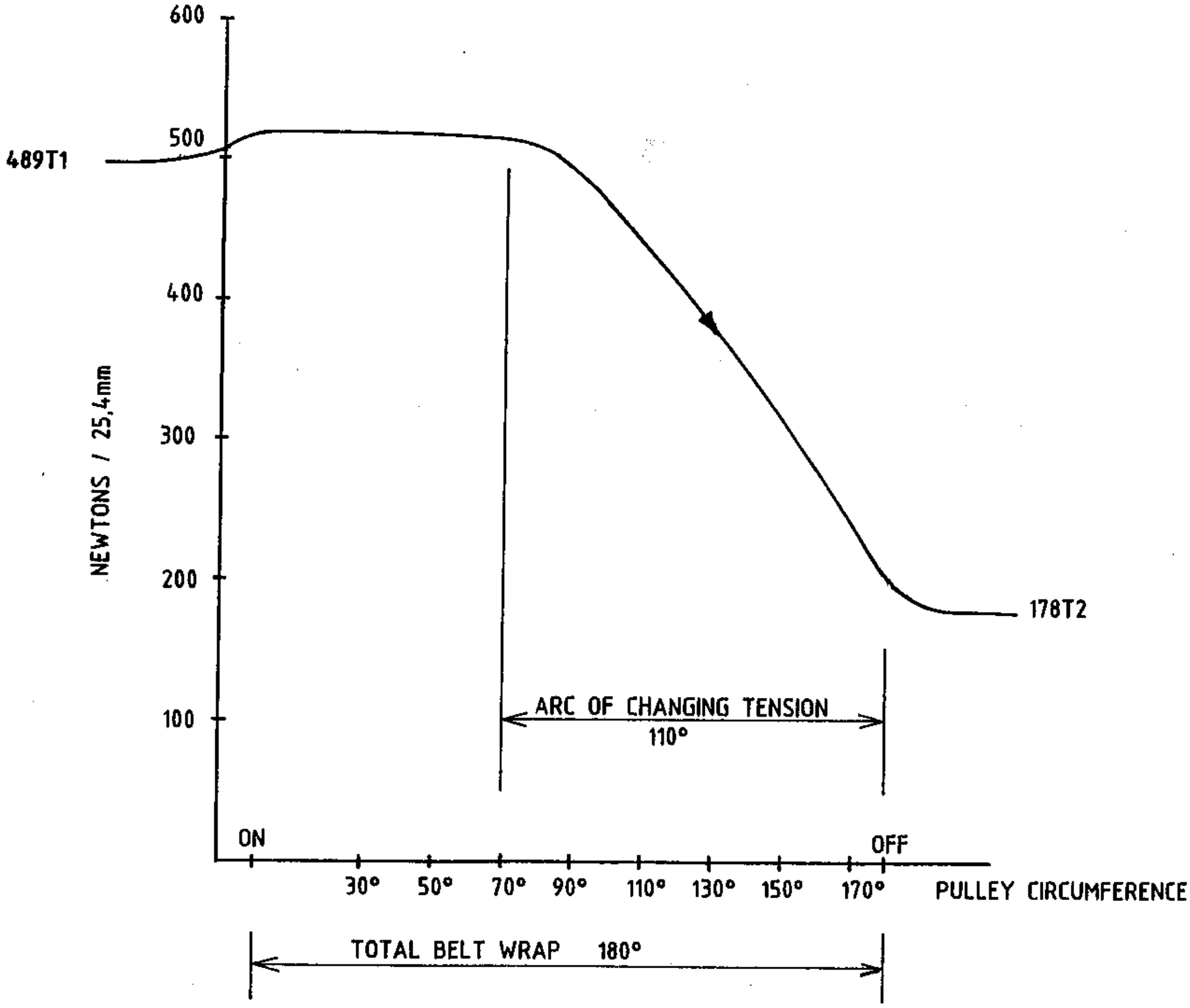
FIGURE 5



INTERPOLATED NEUTRAL AXIS OF BELT TENSION VARIATIONS WHILST BEING DRIVEN AROUND A 610mm DIAMETER UN-LAGGED PULLEY - DATA TAKEN FROM CLARKS EXPERIMENTS ± 1955

(BELT 25,4mm WIDE OF 5 PLY 32 oz COTTON (AMERICAN) DUCK WITH P.V.C COVERS BELT MODULUS E= 1508 kN/m)

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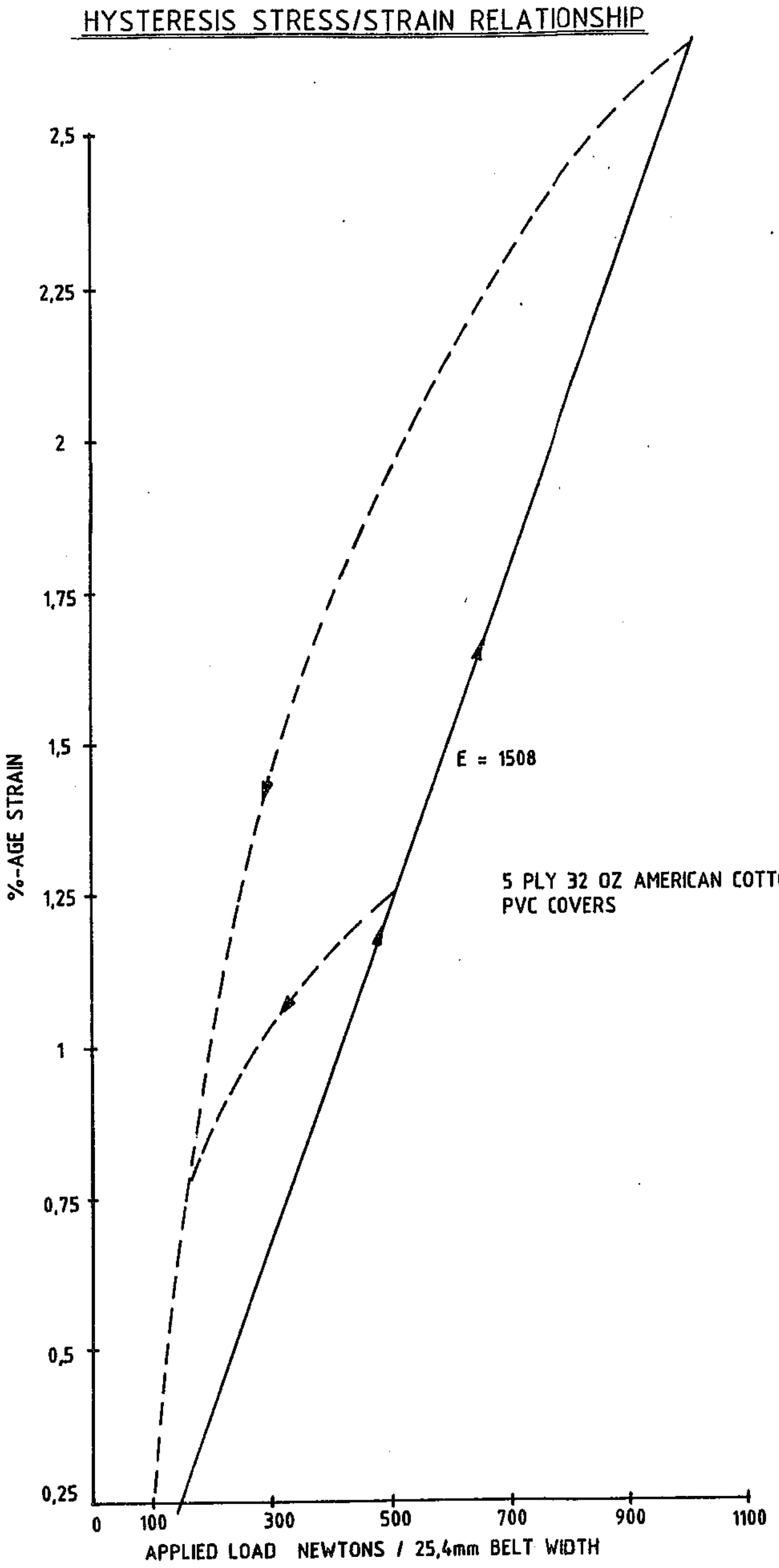
FIGURE 6



DATA FROM CLARK'S EXPERIMENTS

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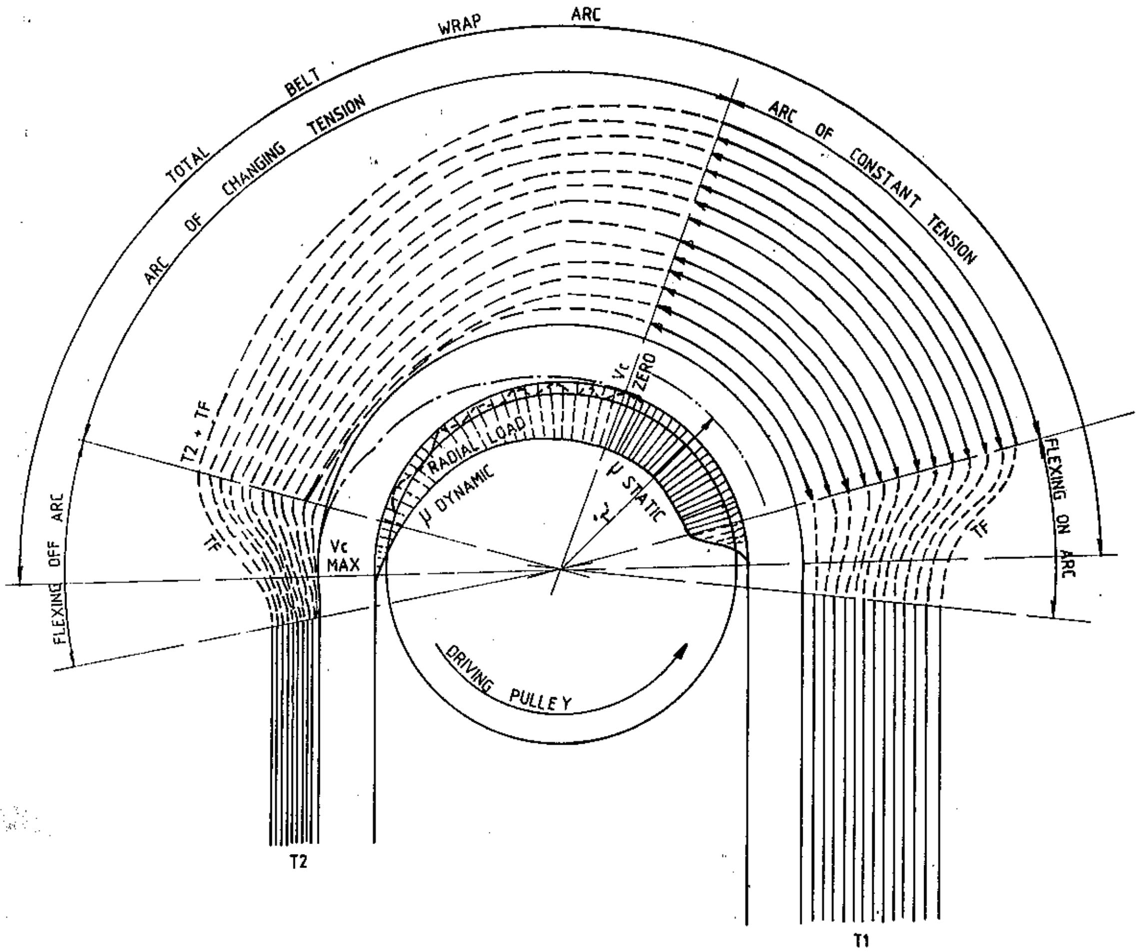
FIGURE 7



DIAGRAMMATIC INTERPRETATION OF CURRENT EVIDENCE

T1 = HIGH TENSION
T2 = LOW OR SLACKSIDE TENSION
TF = TENSION DUE TO FLEXING STRAIN IN THE BELT AROUND THE PULLEY
Vc = CREEP VELOCITY
R = RADIUS TO NEUTRAL AXIS OF THE BELT

NB SOLID LINES REPRESENT NO CHANGE
BROKEN LINES REPRESENT CHANGE

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FIGURE 8

DRIVING PULLEYS

COMPARISON OF RATIO OF TENSIONS v ARC OF CHANGING TENSION FOUND DURING CLARK'S EXPERIMENTS WITH 5 PLY COTTON DUCK BELTING.

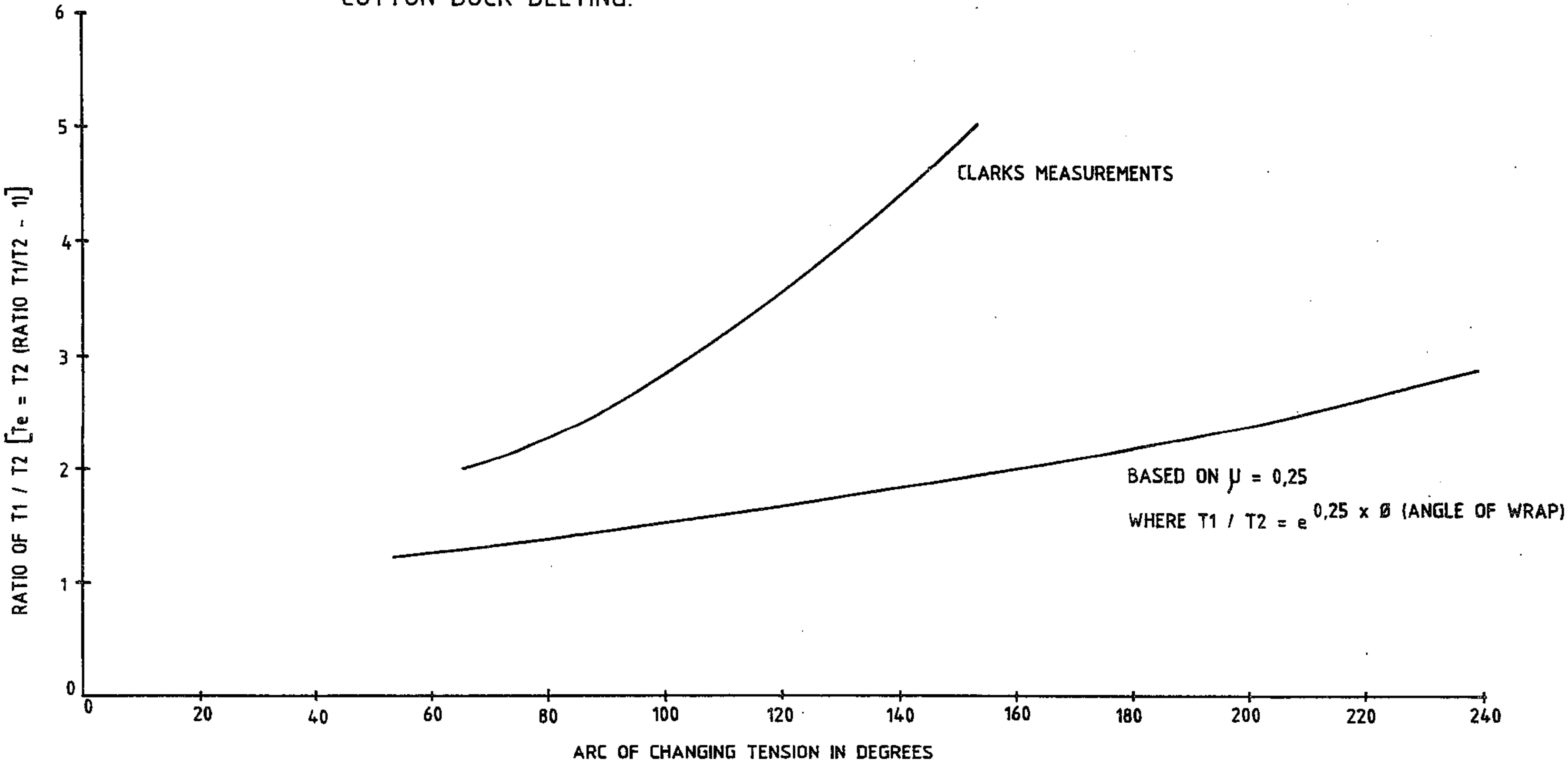

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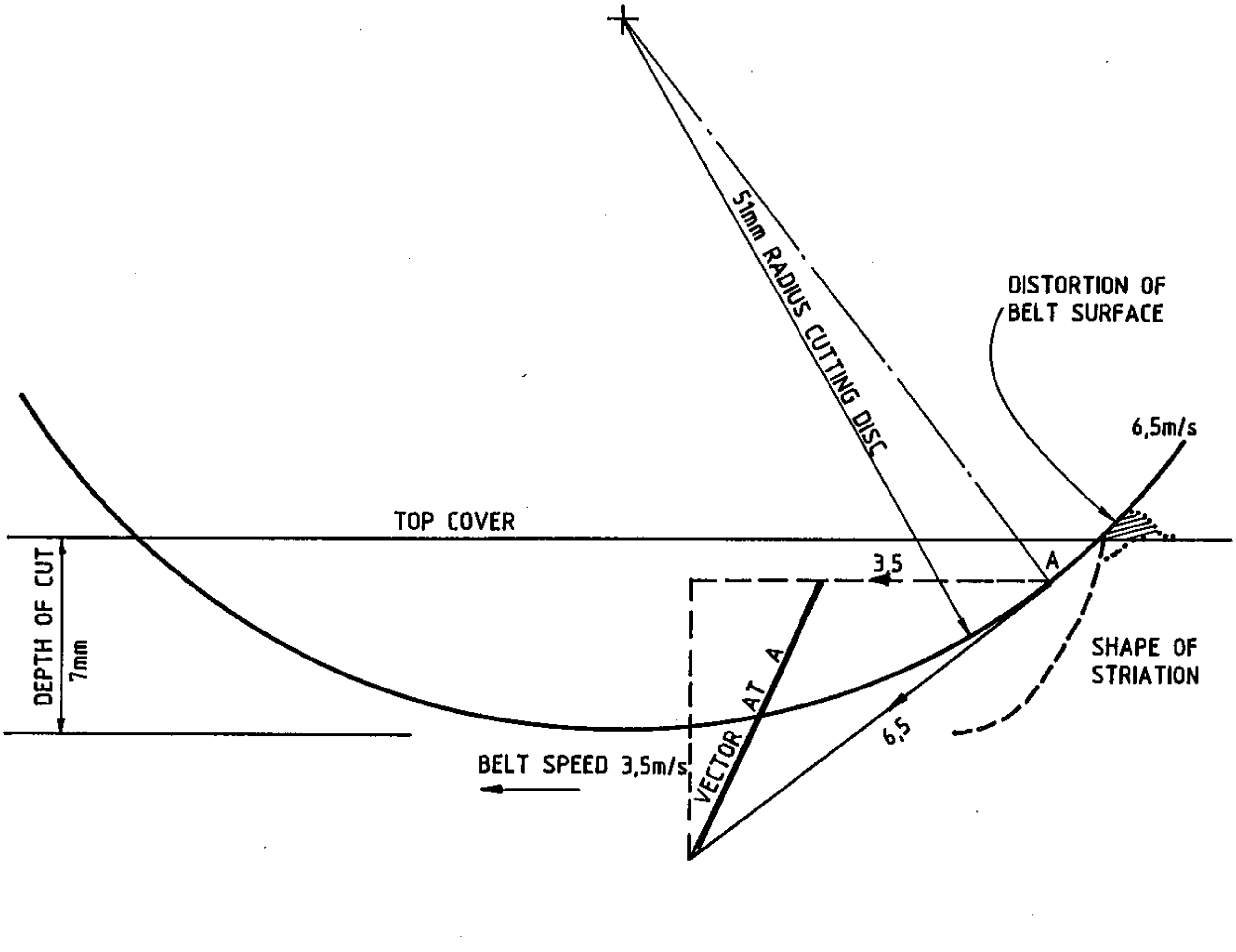
FIGURE 9

ANALYSIS OF CUTTING WEAPON LEAVING AN INTEGRAL SHAPED STRIATION

ASSUME BELT SPEED 3,5m/s
CUTTING DISC 102mmØ
CIRCUMFERENTIAL SPEED $\frac{102 \times 3,5}{55} = 6,5$m/s



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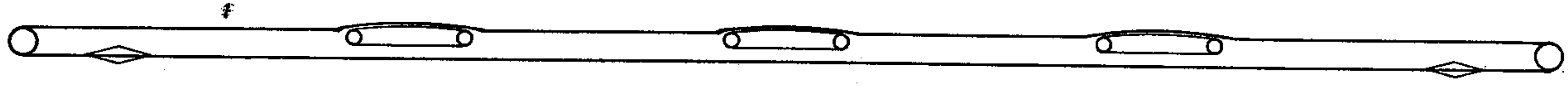

AN ALTERNATIVE CONCEPT FOR VERY LONG HIGH SPEED
CONVEYOR SYSTEMS UTILIZING INTERMEDIATE DRIVES.

FIGURE 10

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