



BELTCON 3

The Function and Mechanism of Conveyor Pulley Drums

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THE FUNCTION AND MECHANISM OF
CONVEYOR PULLEY DRUMS

S Y N O P S I S

As a result of modern constructions the typical failure pattern in conveyor belt pulleys has changed, putting more emphasis on the importance of shell design. Existing empirical and analytical methods have proved unable to cope with this challenge. The paper describes a new and complete analytical system. Using this model the reasons for shell failure trends are identified and cures demonstrated. The same model is used to show how shell life can be predicted and matched to plant life at little or no additional cost. The model is fully justified and tested against wear observations and finite element analysis.

Together with the Beltcon 2 paper "Belt Conveyor Pulley Design - Why the Failures" this paper provides a complete and simple analytical system for pulley design and assessment.

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Design Engineer to the Bosworth Group of Companies

2.0 INTRODUCTION

The current chaotic situation with regard to conveyor belt pulley construction continues.

At Beltcon 2 a paper was published (1) which contained a method for analysing pulleys as a whole. The method was general and simple so that different constructions could be assessed for suitability in a given application, at drawing office level.

That paper contained a major omission on drum shells summarised by the statement "it becomes very important to understand the way the drum behaves sadly our understanding is very incomplete".

The importance is that the drum shell life legitimately determines the life of a pulley. As will be shown knowledge of the shell life is the missing factor which has prevented us from fully designing pulleys matched to plant life. This paper gives an insight into a three year research programme designed to complete that understanding.

3.0 TRENDS IN PULLEY DRUMS

3.1 New Constructions

The dominant trends in local pulley construction in the last ten years have been:

- (a) The adoption of friction based shaft connections.
- (b) The placement of pulley end welds.

Originally the typical construction was that shown in Fig 1a where hubs are fitted to the shaft by shrink and key methods. A separate diaphragm plate joins hub and shell. The shaft and ends are pre-assembled prior to the shell/end weld being made. Fig 2a shows a typical mode of failure for such a pulley, the usual cause is excessive shaft deflection, and failure is easily preventable. These pulleys have an excellent record of reliability.

In Germany in the 1960's and 1970's a family of taper locking shaft connections was developed for keyless operation, clamping the shaft by friction alone. Figure 1b and 1c show constructions using this type of shaft connection. These constructions also eliminate the highly stressed hub/diaphragm weld and hence a possible source of failure. This type of connection is now almost universally used for heavy duty applications.

A further development of this construction is shown in Fig 1d where the end/shell weld is removed

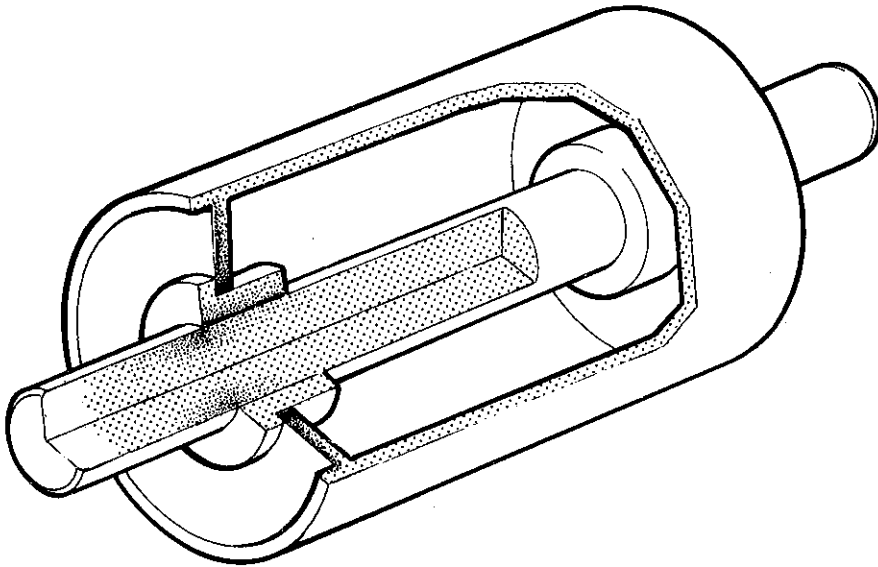


Fig. 1a
Weld-in-hub construction

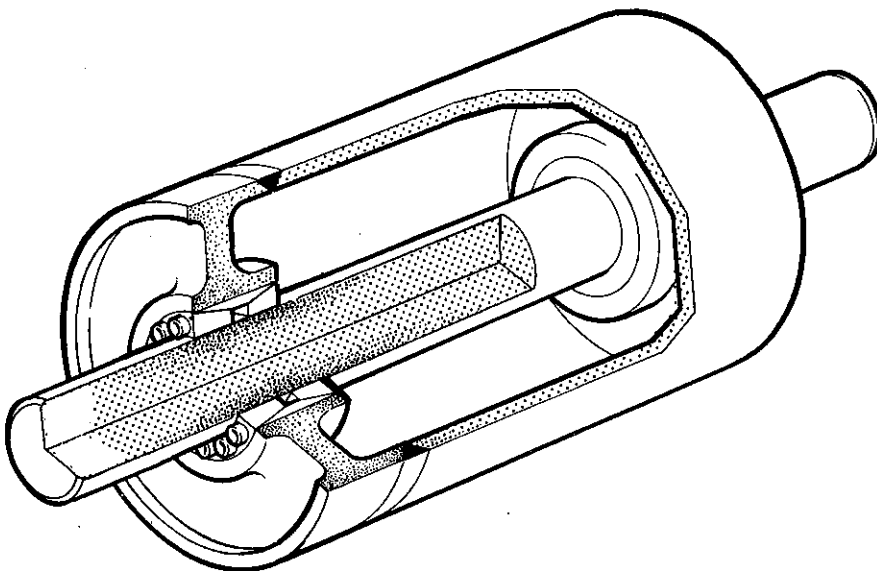


Fig. 1d
'T' Bottom pulley

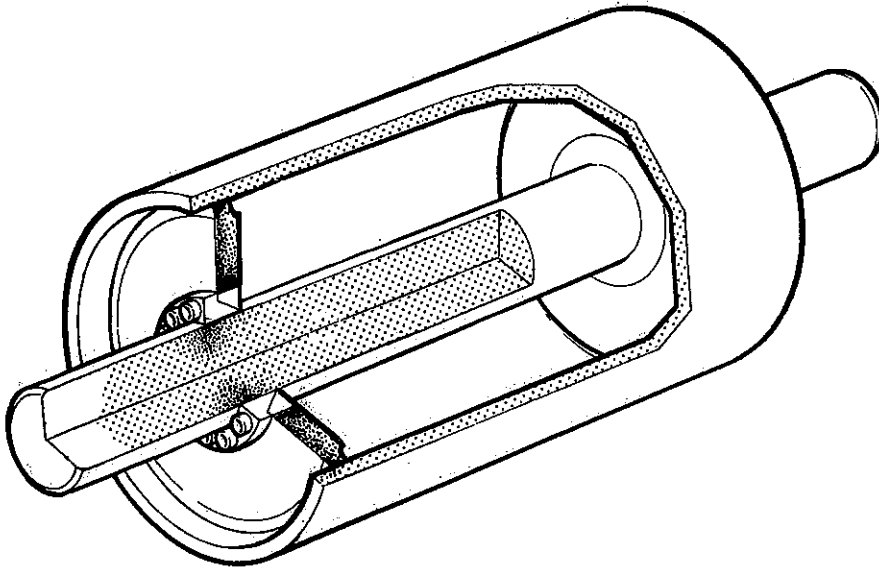


Fig. 1b
Plate end construction

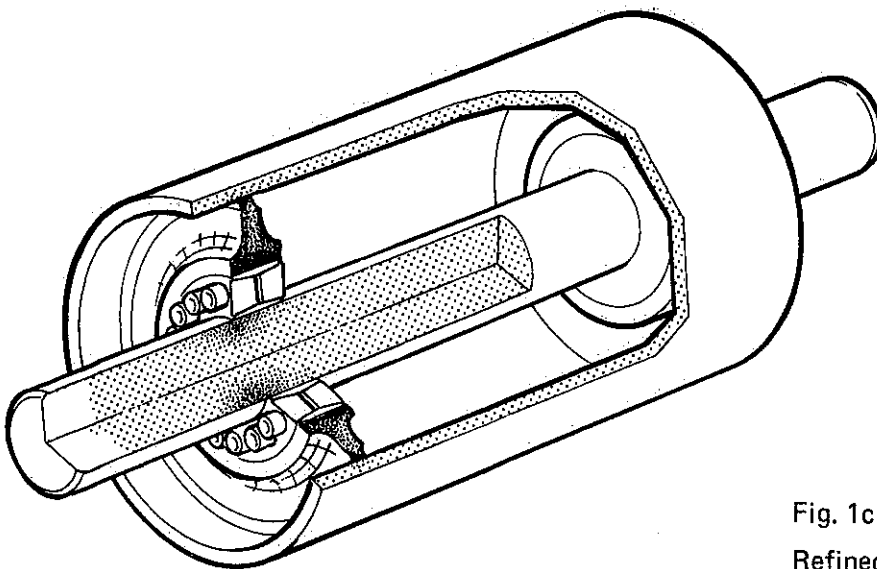


Fig. 1c
Refined plate end construction

to the drum surface. This is done to remove the other potential failure zone illustrated in Fig 2b.

3.2 New Failure Patterns

These trends have proceeded empirically. So that in some of our largest installations we are now using pulleys we do not fully understand and for which only a few years local operational experience exists. The pattern of observed failures for such pulleys has also changed.

If we set aside failures due to shaft related problems - fully dealt with in the earlier paper (1) the dominant failure is now a shell crack on the pulley surface inboard of the drum end. Figs 3 and 4 show examples of such cracks. This type of crack has often been concealed by lagging as in Fig 3 and can reach serious proportions undetected.

Taken together the incidence of such failures, the strong commercial pressure to reduce shell thicknesses and the trend toward constructions such as 1d increase our need for a method of analysis for shells.

3.3 Previous analytical methods

Some good analytical methods do exist for drums due to work in Germany is the 60's and 70's (2,3). The method deals with very large drums with small effects due to the shaft connections. The results of such analyses give startlingly thin shells to sustain quite high loads for example a 1100mm

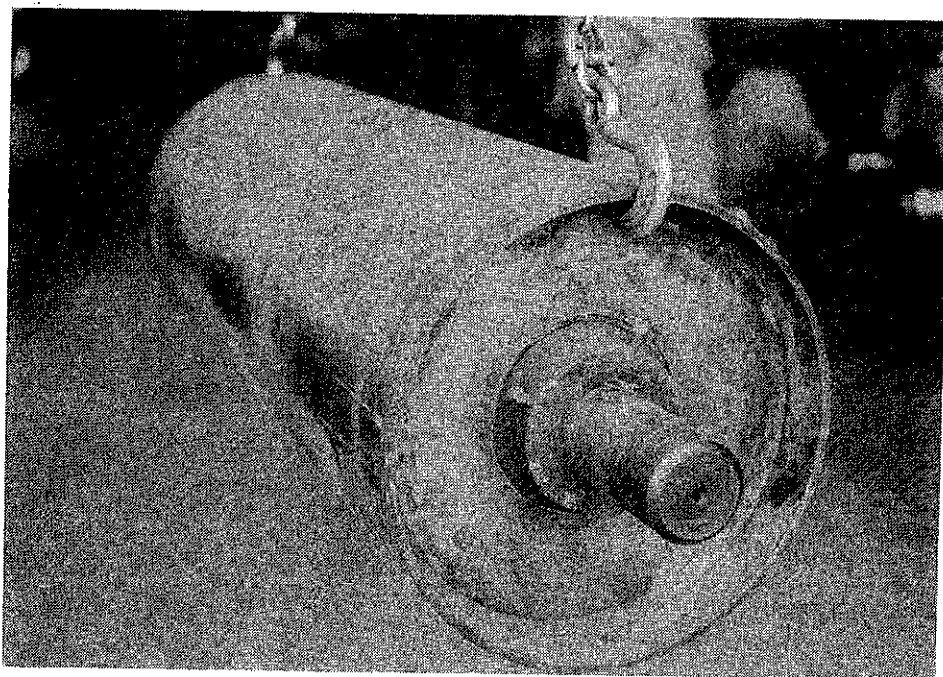


FIGURE 2a HUB WELD FAILURE

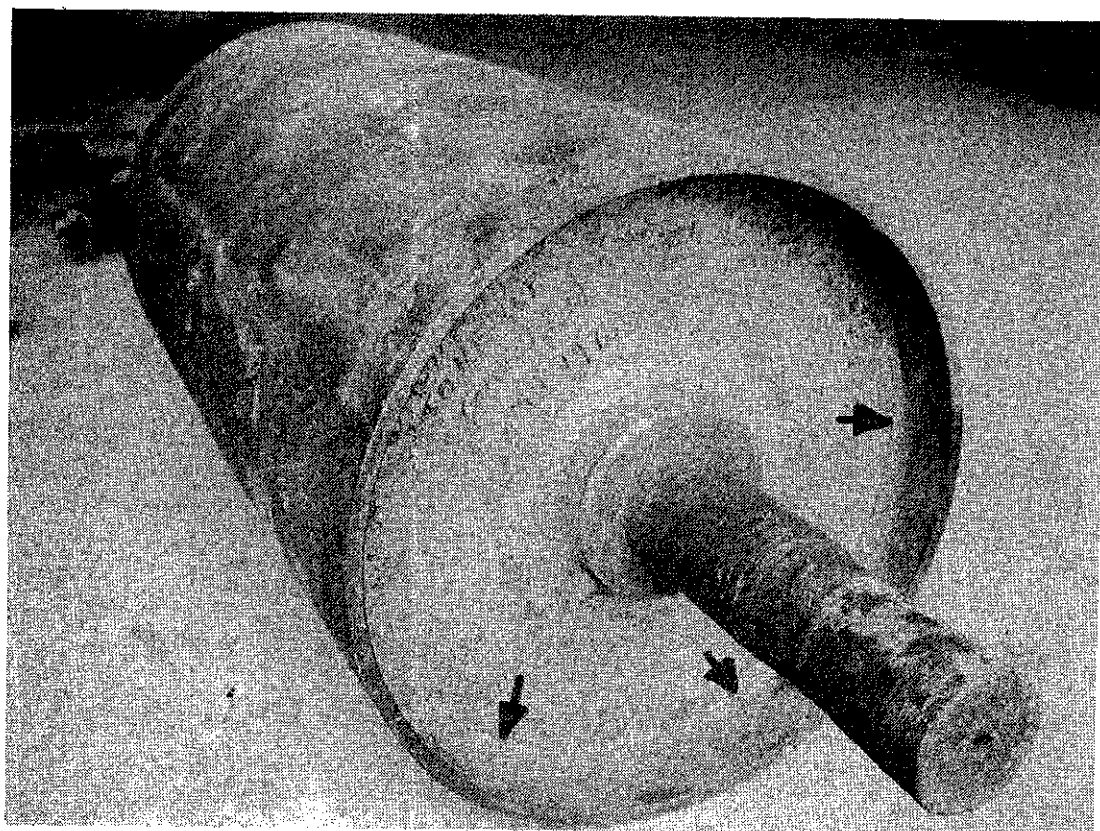


FIGURE 2b END TO SHELL FAILURE

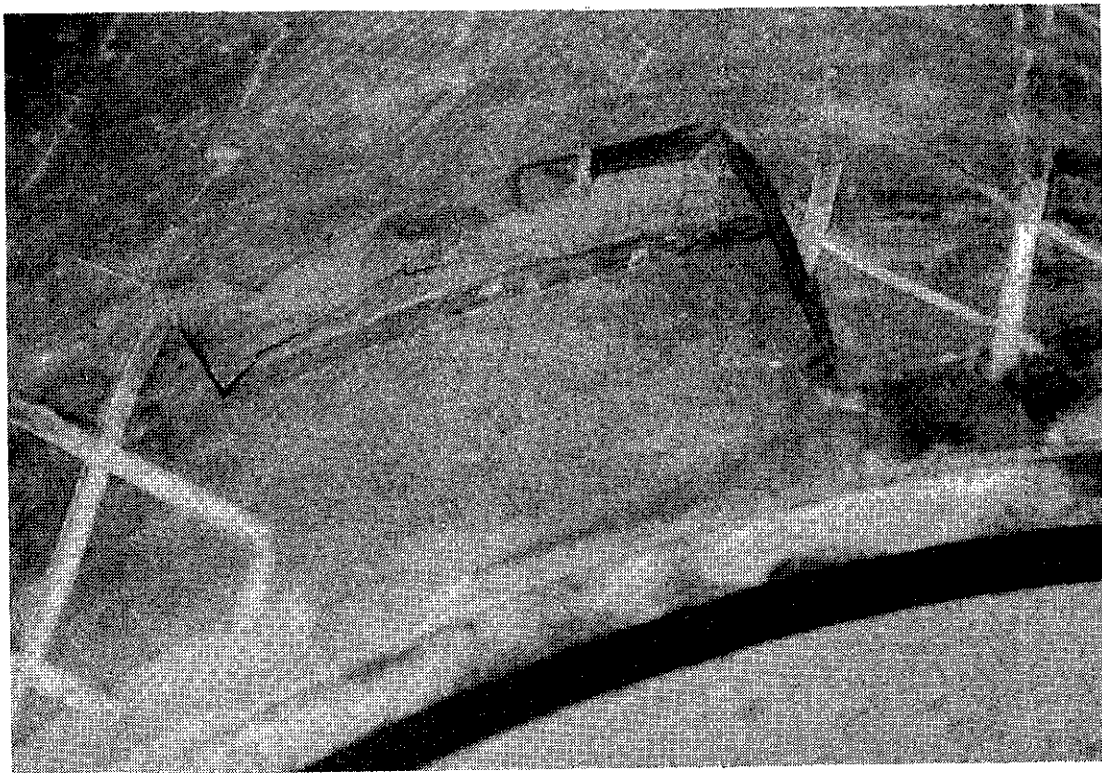
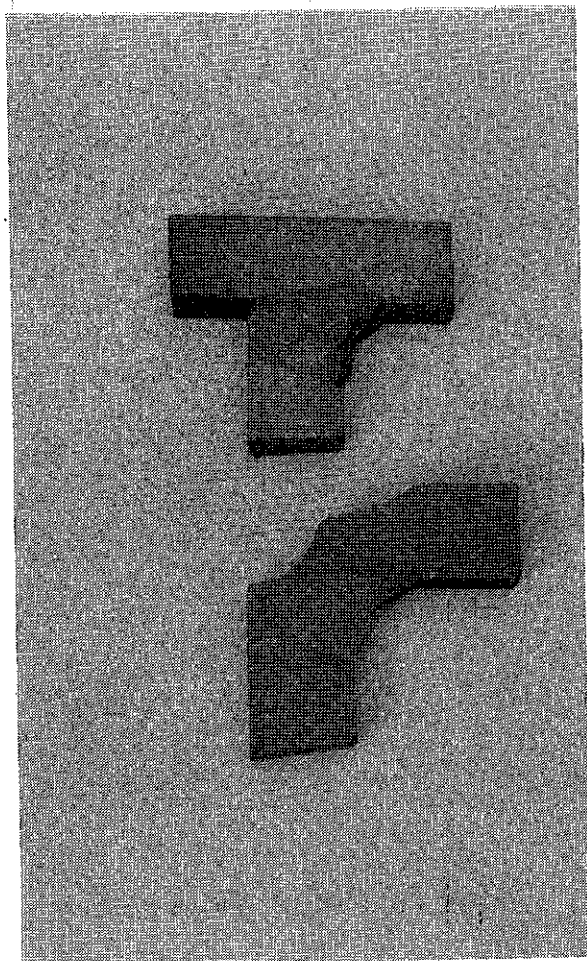


FIGURE 3 TYPICAL SHELL CRACK



**'T' BUTT WELD
SHOWING START OF CRACK**

**FRACTURE RUN FULLY
THROUGH SHELL**

FIGURE 4

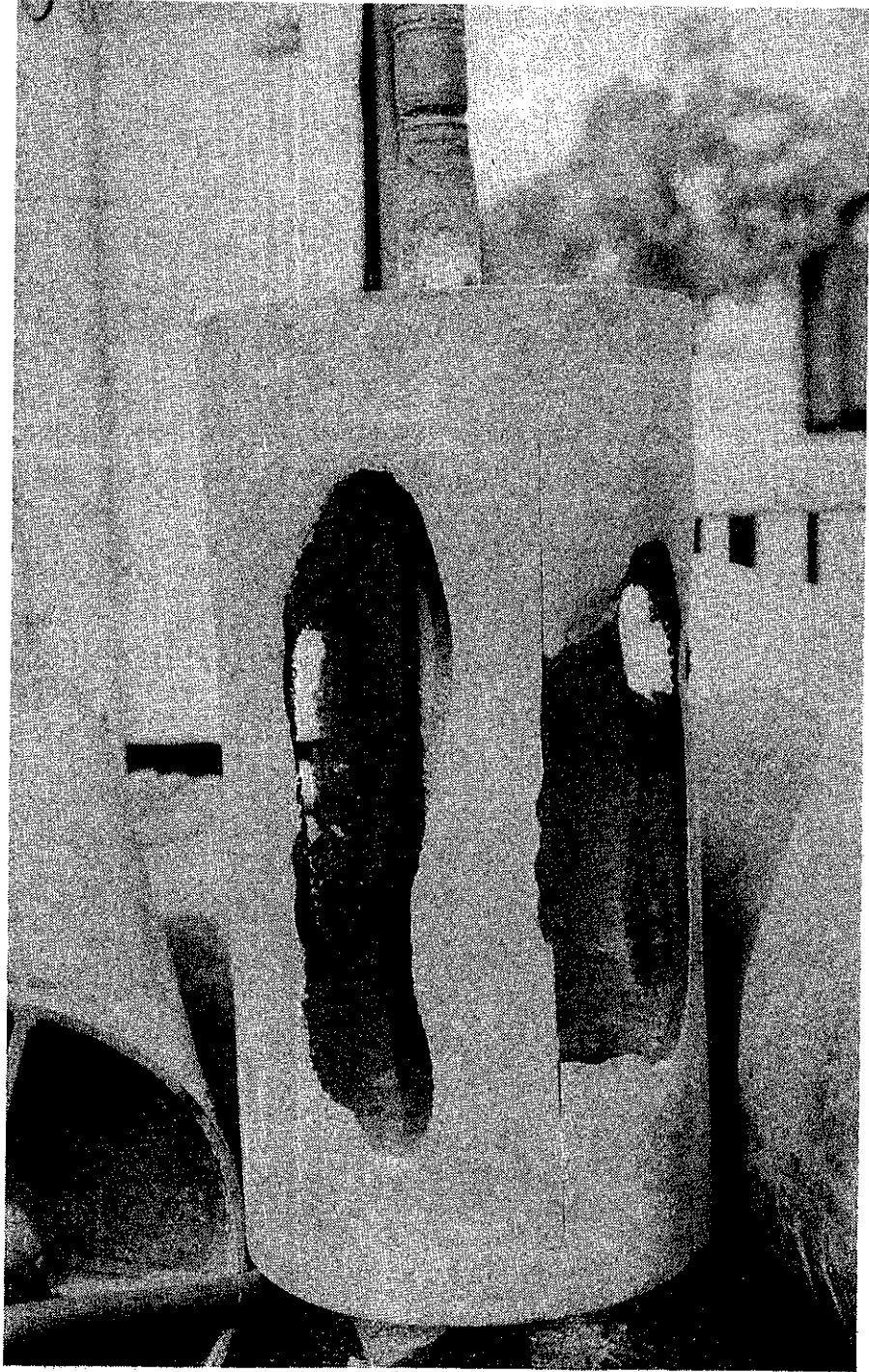


FIGURE 5 THIN SHELL PULLEY

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face pulley with 30 tons of resultant load requires only 4mm plate.

The reason for this is that the highest stress is assumed to occur centrally across the drum and hence the end effects normal to closed end shells and the effects of the shaft connection are ignored. This is an unjustifiable assumption as we shall see.

Similarly, the empirical methods developed in Australia are limited to the sizes investigated, and show illogical results at the ends of their ranges, for instance in pulleys of similar face width and diameter required shells exceed 30mm thick irrespective of loading, or construction.

3.4 Analytical Methods in Practice

There is good evidence to suggest the German method works well on relevant pulleys as for instance Fig 5 where an average shell thickness of 0,5mm served until bearing failure was experienced. Sadly, this pulley is by no means typical of the most critical modern installations.

Observation and measurement show a very typical pattern for the damage to pulley shells after long service. Fig 6 illustrates this. Note how the pulley has formed "saddlebacks" midway between the ends and the central inner stiffener. At the lowest point of the saddleback the shell thickness is reduced (in this case to zero). This pattern has been found typical across a large sample of pulleys returned for repair. Fig 7a and b shows a set of diameter and thickness measurements from a pulley with the saddleback clearly visible. Note that the saddleback is formed in

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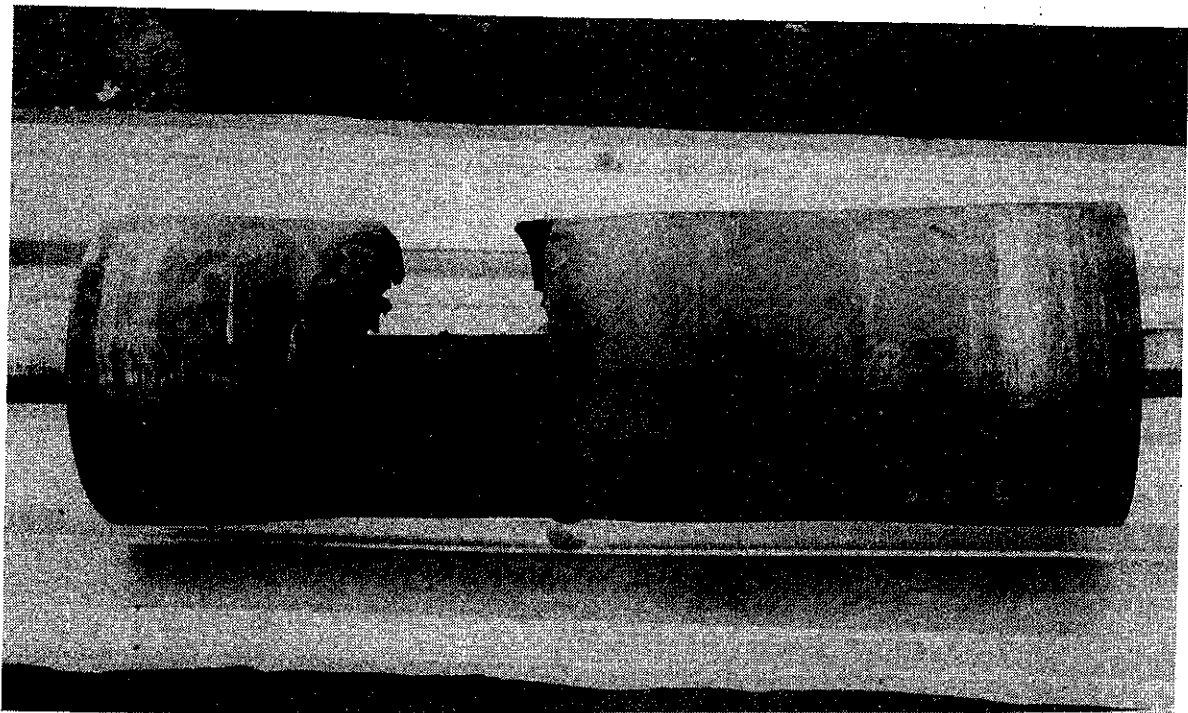


FIGURE 7a A TYPICAL PULLEY AT THE END OF ITS WEAR LIFE

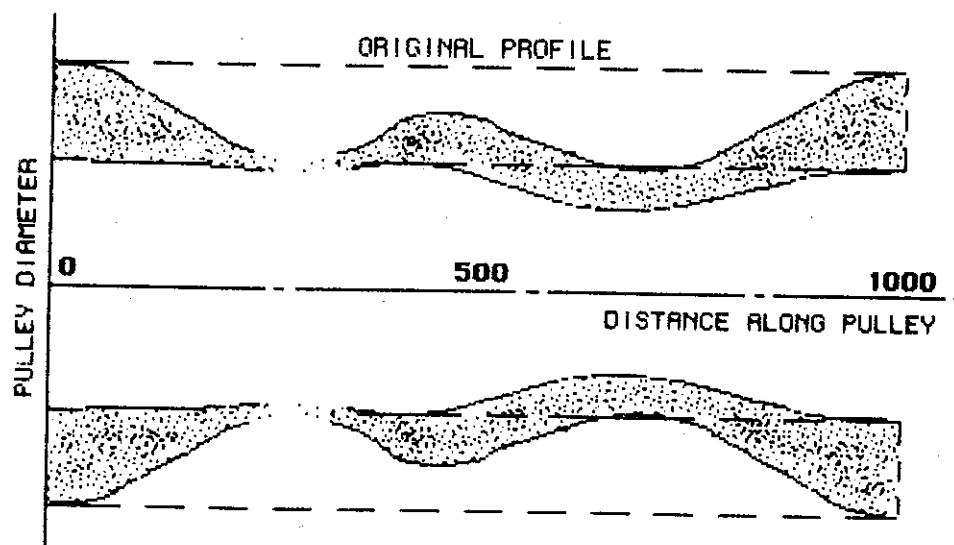


FIG 7b SHAPE OF PULLEY SURFACE AFTER FULL WEAR LIFE



FIGURE 6 SINGLE INNER STIFFENER PULLEY-IN A WORN STATE

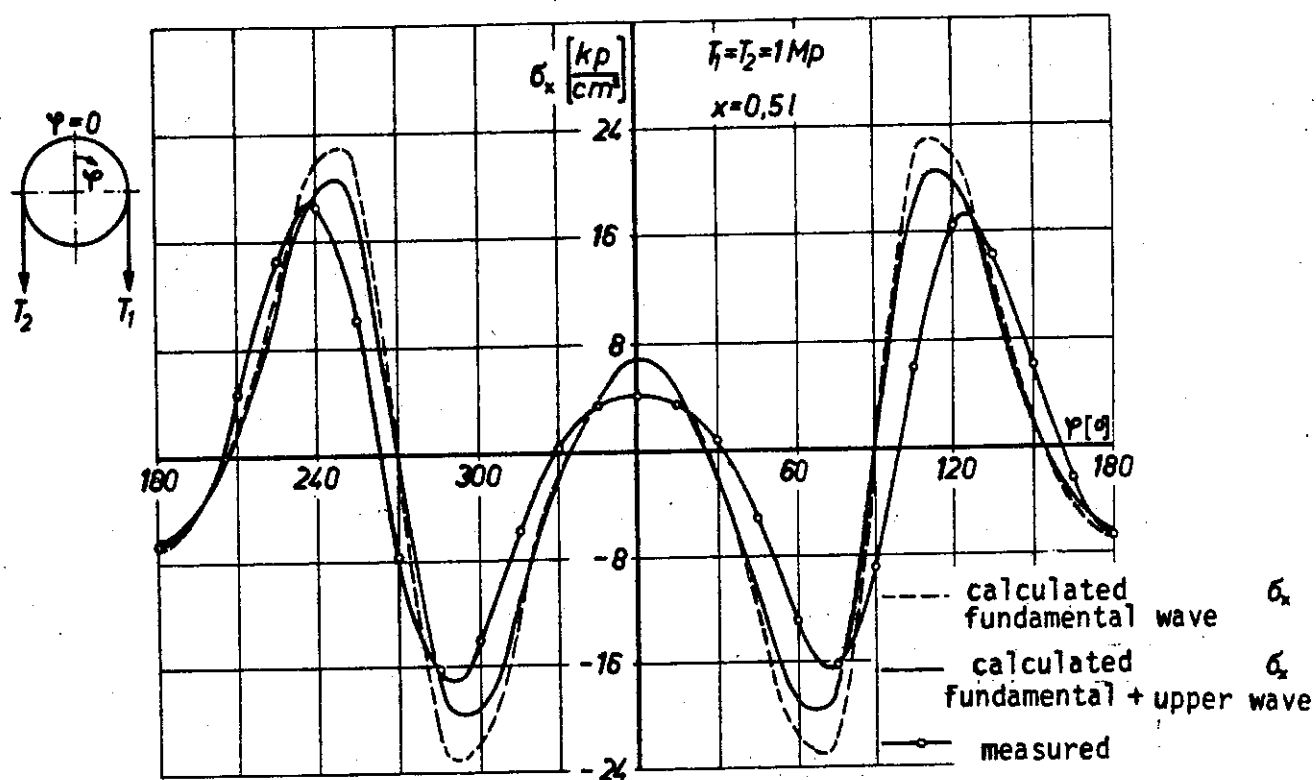


Fig 8 : Axial stresses in the pulley shell

4.1 Background

To produce a complete model of pulley shell stress requires a very complex approach since the problem cannot be dealt with in less than three dimensions, in loading and geometry. This would be far beyond the scope of this present paper and forms the basis for the continued research programme between Bosworth and the University of the Witwatersrand Technology Centre.

We can however provide an approximate model which agrees well with practice and is supported by the sophisticated methods. The proposed model is conservative but demonstrates clearly most of the problem areas and the effects of varying construction.

This model has specifically been designed for pulley user checks at drawing office level.

4.2 The Model

The stress picture is built up by superposition of a number of effects calculated only at their positions and directions of peak value. Fig 9 illustrates the effects considered.

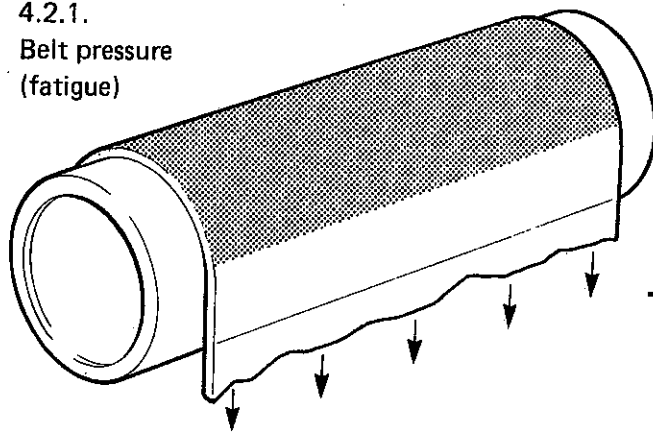
An important aspect is that the calculations should be performed for both the worn and unworn conditions taking into account the "saddleback" effect between stiffened points.

None of the elements may be neglected since different effects predominate in different pulleys.

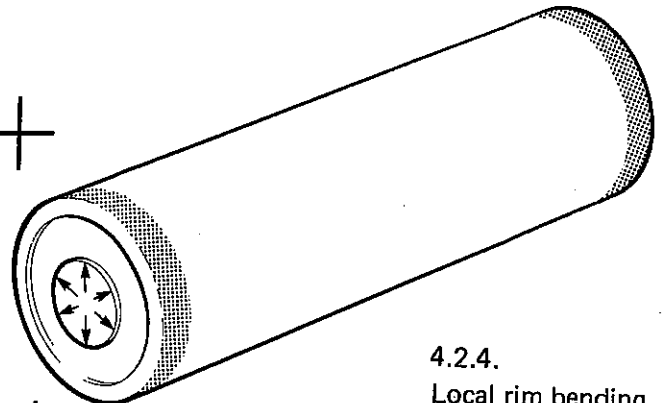
almost equal proportions by wear (metal loss) and plastic deformation. None of the current analytical systems predict high enough stresses to cause such deformation even given the rate of cycling of the imposed fatigue stress which has been shown to be 3 times the pulley rotation speed (2) as in Fig 8.

Finally, it should be observed that the plastic deformation occurs in the same region as the failures discussed earlier.

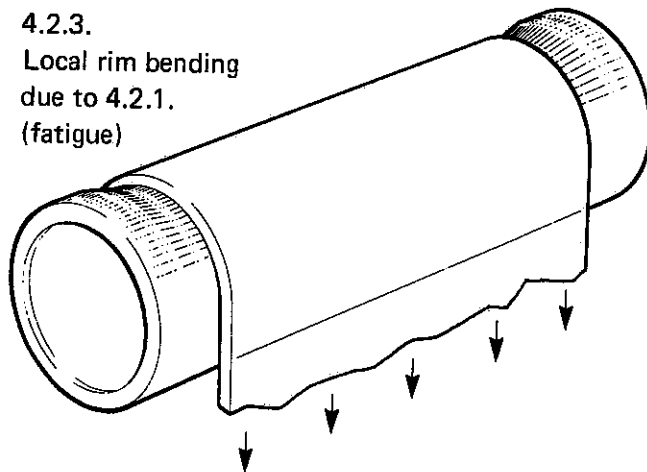
4.2.1.
Belt pressure
(fatigue)



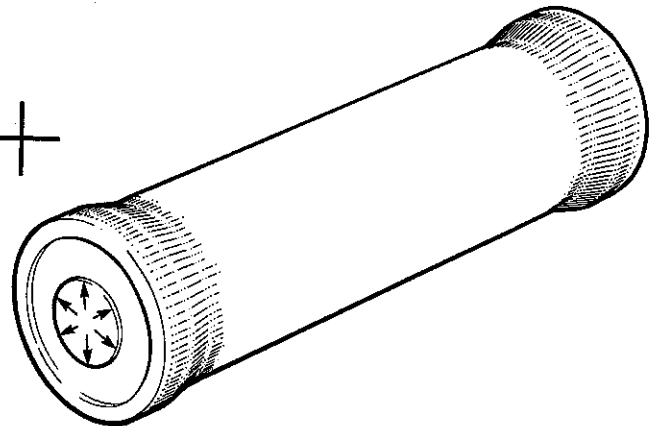
4.2.2.
Shaft connection
induced expansion
(static)



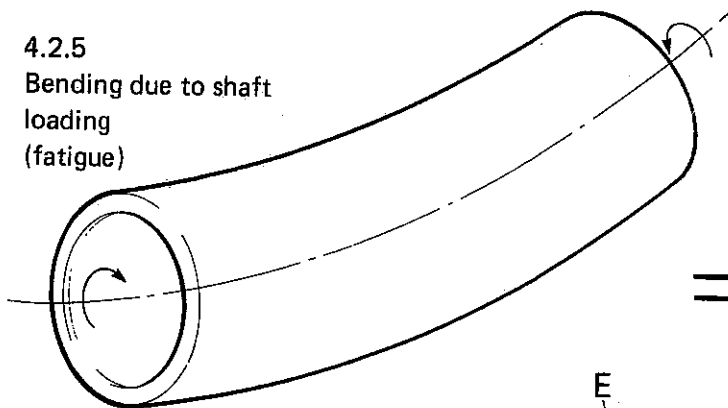
4.2.3.
Local rim bending
due to 4.2.1.
(fatigue)



4.2.4.
Local rim bending
due to 4.2.2.
(static)



4.2.5
Bending due to shaft
loading
(fatigue)



4.2.6
Net stress values

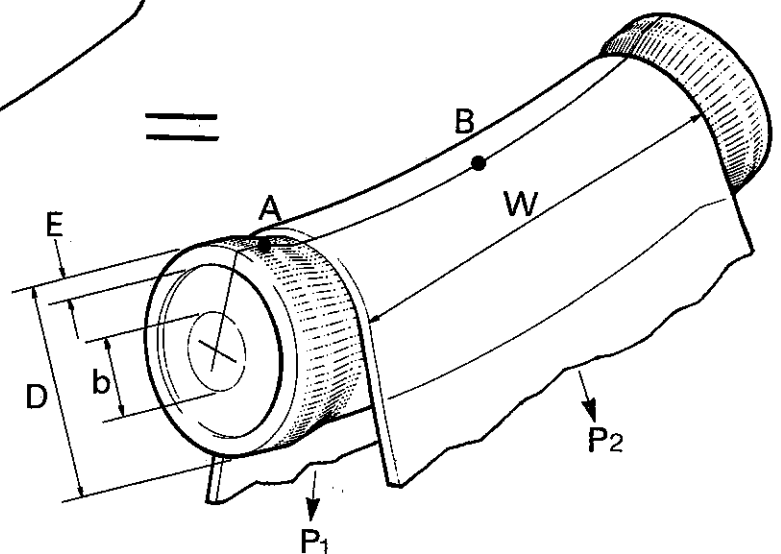


Fig. 9.
THE STRESS MODEL
(Static and fatigue)

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4.2.1 Belt Pressure.

Belt pressure on the surface can be shown from first principles to act as in Fig 10, where pressure is solely a function of curvature and instantaneous belt tension (T). Then for a non-drive pulley

$$P_1 = P_2 = T \quad (1)$$

and

$$\text{Surface pressure } z_b = \frac{2T}{W.D.} \quad (2)$$

Where W = belt width, D = Pulley diameter
To find stress use thin wall tube theory
(resulting error close to ends is not significant)

$$\text{Hoop stress } S_b = \frac{z_b D}{2t} \quad (\text{compressive}) \quad (3)$$

Where t = shell thickness after machining,
Recalculate for thickness after wear.

Drive pulley subroutine - In the case where belt tension is assymetrical, an equivalent tension (T) is found and assumed to act evenly over the belt lap.

First calculate the active drive arc (2) in practice for the highest probable coefficient of friction (μ).

$$\theta = \log_e (P_1/P_2) / \mu \quad (\text{radians}) \quad (4)$$

Then $T_e = \frac{P_1}{\mu\alpha} (1 - e^{-\mu\theta}) + P_2 (1 - \frac{\theta}{\alpha}) \quad (5)$

Where α = wrap angle in radians.

Use T_e instead of T in equations (2) and (3) to find central Hoop compressive stress.

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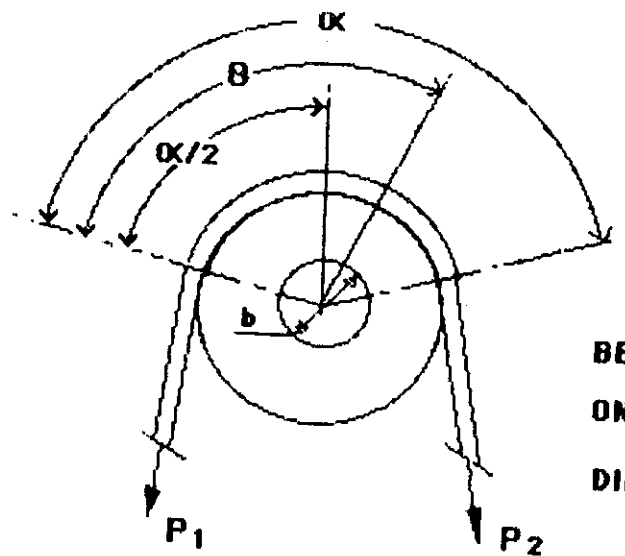


FIG 10

BELT LOADING
ON A PULLEY
DIAMETER 'D'

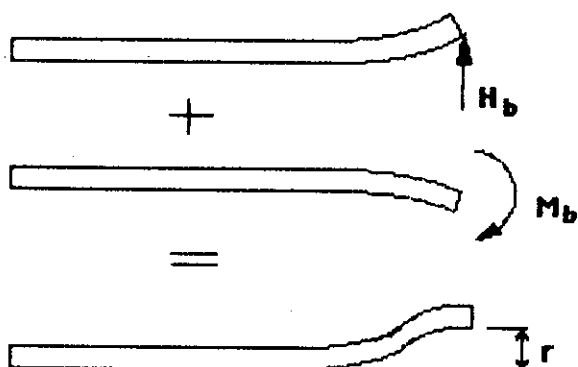


FIG 11 RIM BENDING OF SHELL

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- 4.2.2 Shaft connection induced expansion. Lamé's method is used for this solution (4). This works well for constructions of the type in Fig 1b. A larger and significant error may be involved in type 1c and 1d constructions. In these cases an iterative approach can be adopted to give a solution, taking into account the variations in through thickness of the drum ends. Such a programme is available as a cross.check.

By Lamé's method shell internal hoop stress S_1 is :

$$S_1 = \frac{pb^2(D^2 + D_i^2)}{D_i^2(D^2 - b^2)} \text{ hoop, tensile} \quad (6)$$

Note p = shaft connection pressure (Mpa)

b = bore of pulley end, usually shaft connection diameter

D_i = Shell inside diameter

- 4.2.3 Rim bending - due to belt pressure.

In modern pulleys this is often the dominant effect. It describes the stress caused when two adjacent sections of a plate deflect to different positions as shown in Fig 11. For a detailed treatment of this phenomenon see (5,6 and 7) this has not previously been applied to pulleys, most of this work being done on large pipelines with hoop stiffeners. The governing equations after (6) are; for radial deflection r_b

$$M_b = \frac{2r_b t.E.Z^2}{D^2} = \text{Rim bending section moment} \quad (7)$$

$$H_b = 2M_b/Z = \text{Rim section force} \quad (8)$$

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Where z = shell constant

$$= (Dt/2) \sqrt{3(1-\nu^2)}^{\frac{1}{2}} \quad (9)$$
$$= (Dt/3,3)^{\frac{1}{2}} \text{ for our purposes}$$

and S_{ab} = axial rim stress $= \frac{6 M_b}{t^2}$ (10)

$$S_{hb} = \text{hoop rim stress} = \frac{D}{t z} (H_b - M_b/z) \quad (11)$$

E = Youngs Modulus ν = Poissons ratio

Thus for belt pressure z_b , we have from thin shell equations.

$$r_b = \frac{z_b D^2}{4tE} (1 - \frac{\nu}{z}) \quad (12)$$

and S_{ab} and S_{hb} can be calculated.

4.2.4 Rim bending - due to drum end expansion.

This is calculated exactly as above except that the radial expansion r_1 comes from Lamé's theorem with the limitations noted in 4.2.2

$$r_1 = \frac{p}{E} \frac{Db^2}{(D^2 - b^2)} \quad (13)$$

Hence calculate M_{b1} , H_{b1} , S_{a1} and S_{h1}

Observing the direction of the deflections shows that the sense of S_{a1} and S_{h1} is the same as S_{ab} and S_{hb}

4.2.5 Shell bending.

This effect has been shown (8) when in isolation, to follow closely conventional engi-

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needs bending theory. Applying this in the usual way. The axial bend stress, S_a , is:

$$S_a = \frac{32 MD}{(D^4 - (D-2t)^4)} \quad (14)$$

Where $M = P(B-L)/4$ = Shaft bending moment
 B = Bearing centres
 L = Spacing of pulley ends
 P = Resultant tension

NOTE : Reference (1) gives a method of calculating the bending actually transmitted to the shell (M_d). This can be substituted for M above to reduce the conservatism of the solution.

4.2.5 Total stress.

Local stresses are taken at two points and treated as representative. Firstly at mid-wrap and central on the face of the drum, S_a and S_b are the nett axial and hoop stresses respectively, both of these are fatigue stresses.

Secondly at mid-wrap, inside the drum, adjacent to the end in the axial direction, the nett stress S_{amax} becomes

$$S_{amax} = S_{a1} + \bar{S}_{ab} + \bar{S}_a \text{ (compressive)} \quad (15)$$

Similarly at the same point the peak hoop stress S_{hmax} becomes

$$S_{hmax} = S_{h1} + \bar{S}_{hb} + S_{hl} \text{ (tensile)} \quad (16)$$

Stresses shown thus \bar{S} are fatigue stresses.

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4.3 Typical Values

An example will serve to show how the model works in practice. Also some interesting implications.

If we take a non-drive pulley from a recent typical contract and use the data as follows:-

Belt Tension	$T_1 = T_2 = 17,7$	Kn
Wrap Angle	$\alpha = 180^\circ$	
Load Factor	$F = 1,4$	(applies to T_1 and T_2)
Drum diameter	$D = 450$	mm
Shaft diameter	$d = 150$	mm
End bore	$b = 200$	mm
Internal pressure	$p = 128$	Mpa
Belt width	$W = 1500$	mm
Shell thickness	$t = 16$	mm after machining
Shell bending moment	$M = 2104 \times 10^3$	Nmm (by method in (1))

Following the above system we can calculate the stresses with the results shown in table 1.0

TABLE 1.0 Typical pulley stress components (unworn)

Hem	Stress Source	Symbol	Value	Remarks
1	Belt induced	S_b	1,0	Hoop fatigue
2	Shaft connection induced	S_1	68	Hoop static
3	Rim bending due to belt	S_{1b}	0,9	Hoop fatigue
4	Rim bending due to connection	S_{h1}	63	Hoop static
5	Rim bending due to belt	S_{ab}	1,6	Axial fatigue
6	Rim bending due to connection	S_{a1}	114	Axial static
7	Shell bending	S_a	0,9	Axial fatigue

5.0 THE MODEL AND THE REAL WORLD

5.1 Shell failure patterns

A look at the values in Table 1.0 will show why the change in failure pattern. For a shrink and key type pulley (Fig 1a) the dominant stresses are 1,3,5 and 7 all of these are very low level fatigue stresses hence the longevity of such shells and the suitability of very thin shells. This is the effect predicted by the German studies. These stresses rotate typically at $2-5 \times 10^6$ cycles/annum for high duty applications.

The introduction of shaft connector related stresses 2,4 and 6 immediately increases failure risk. These stresses appear along a weld line, on the concealed side of a single sided 'T' butt. Such as Fig 12. In such cases only a low level of stress should be tolerated. The highest nett variable stress of 120 Mpa in Table 1.2 is close to an allowable maximum.

5.2 'T' Bottom Pulleys

Use of constructions such as 1d remove the weld from the immediate area of the end. This does not necessarily help. We must look at the nature and origin of the stress components 4 and 6 in Table 1.0 which are local bending stresses generated by a forced deflection in the thin shell. The dissipation of such stresses follows a damped sinusoidal form (8) decaying at a rate related to the shell thickness. Complete decay is achieved in a few thicknesses. Fig 13 shows the decay of these stresses and the effect of shell thickness.

TABLE 1.1 Pulley nett stress levels at mid wrap
(Tensile stress shown +ve)

Position - Direction	Symbol	Value	Remarks
Central - Axial	S_a	$\pm 0,9$	Unworn
Central - Hoop	S_b	$\pm 1,0$	Unworn
Adjacent to end - Axial	S_{amax}	- 116,5 - 41,5	Unworn
Adjacent to end - Hoop	S_{hmax}	+ 132 + 130	Unworn

TABLE 1.2 Pulley nett stress levels at mid-wrap calculated
after full anticipated wear.

Position - Direction	Symbol	Value	Remarks
Central - Axial	S_a	$\pm 2,5$	After 10mm wear
Central - Hoop	S_b	$\pm 1,6$	After 10mm wear
Adjacent to end - Axial	S_{amax}	- 124 - 116	After 5mm wear
Adjacent to end - Hoop	S_{hmax}	+ 135 + 134	After 5mm wear

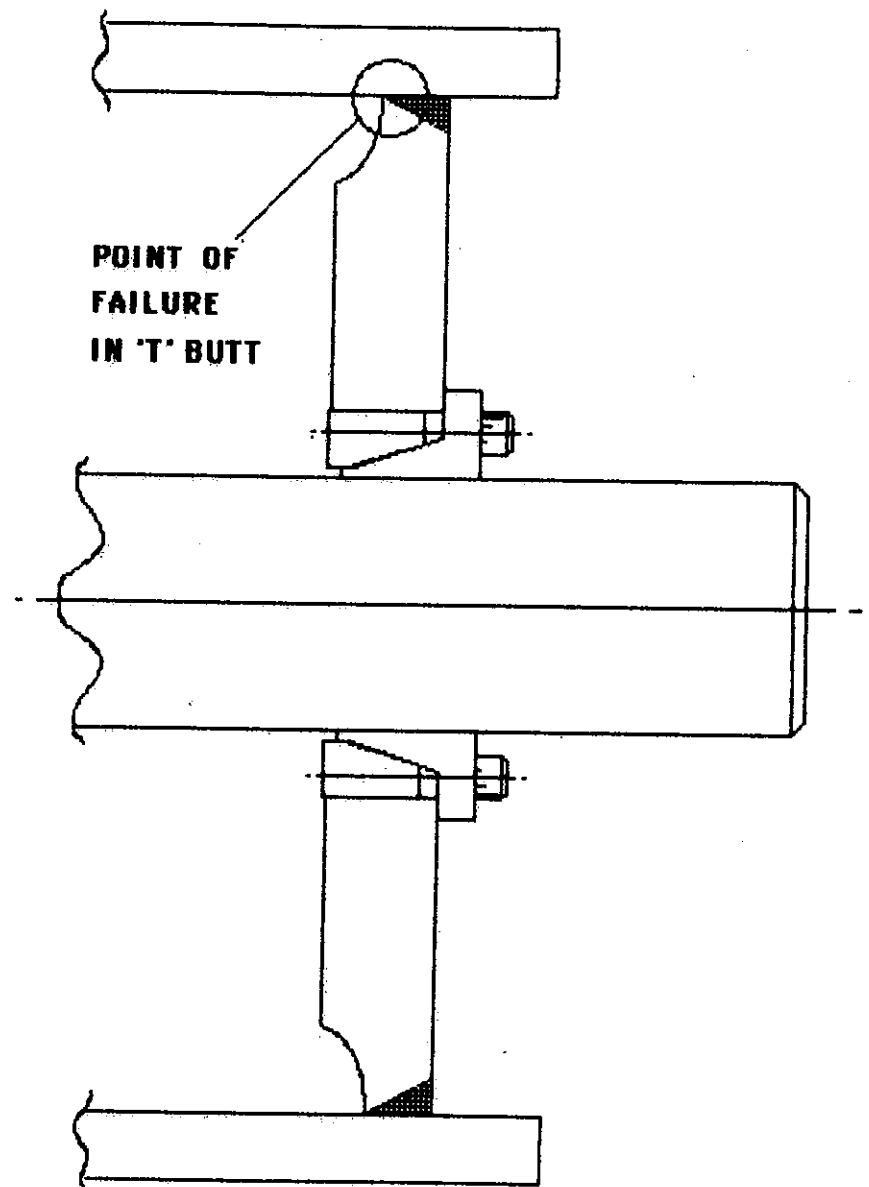
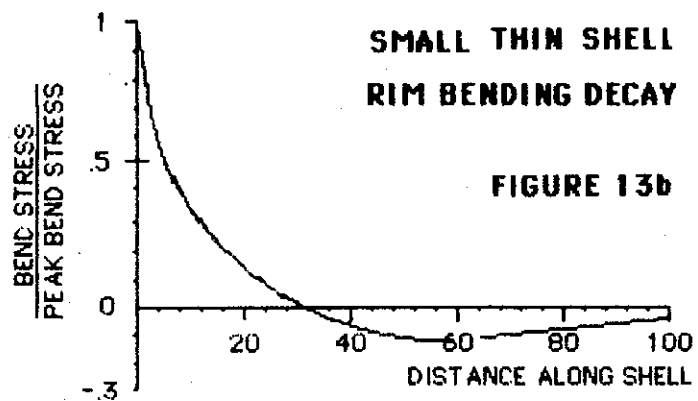
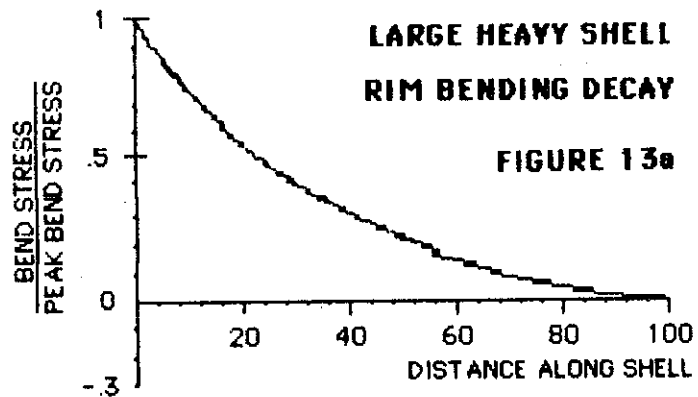


FIGURE 12 LOCATION OF MAX STRESS IN SHELL
— ALSO TYPICAL FAILURE POINT



5.4 Other Constructions

The 'T' bottom pulley has frequently been specified regardless of application. This is a very expensive method of construction, due to the thickness of material required to remove the shell/end weld from the high stress area. Further, it is not necessarily effective.

We can however produce a construction that does effectively remove the weld from the danger area, by using the properties of wide double taper shaft connections. If these are allowed to dictate the end disc thickness and the end profile is offset, we have a 'L' bottom construction as shown in Fig 14. This has the properties of minimum material thickness and cost, also maximum benefit to the weld area. An indent on the end face to allow bearing inset remains possible, although very expensive.

5.5 The Value of Standards

To make best use of such techniques as the 'L' and 'T' bottom pulley requires the ability to tool for forgings or castings and hence best material defect rate and raw material mass.

In a nine month period recently Bosworth manufactured pulleys with 47 different diameters, 25 different shaft sizes and 6 different shell thicknesses giving a required 7050 different tools for optimum effective production.

By contract one major user has in their standard only 780 variations to cover the same range. This

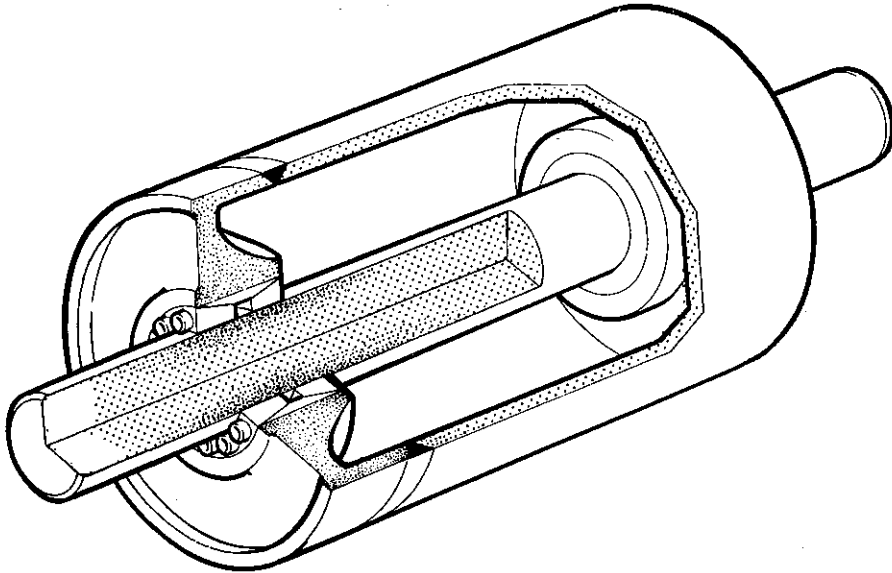
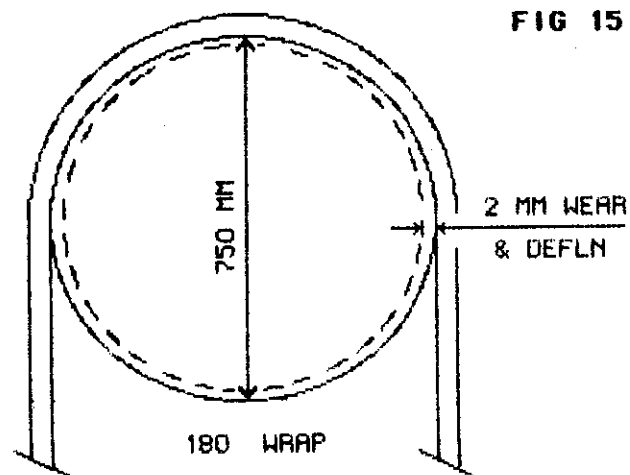


Fig. 14.
'L' Bottom pulley



DIAMETER DIFFERENCE CAUSING WEAR

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has been found effective over a number of years.
similarly, we strongly recommend that a standard
such as 150 1536 (10), even if supplemented, be
adopted in every pulley using organisation.

Assuming that adequate attention is given to the shaft connector effects, we then require to control wear rate on the drum to give a wear life matched to the fatigue life. This is entirely possible.

Lagging is one means that has been effective, however it requires periodic re-work to maintain the lagging. Even removal of the pulley, and hence holding of a spare for vulcanising, in the ideal case.

A study of the wear effects on pulley shells shows that the root cause of metal loss is the diameter differences created in an operational pulley. These arise from the deflections described earlier and result in "scrubbing" of the belt as it attempts to pass over a surface with differing linear velocities. This can be illustrated by a simple example, Fig 15 shows a pulley with a 2mm radial difference so that the linear "scrub" over the pulley surface is

$$\begin{aligned} S_d &= \frac{\pi}{2} (750 - (750-4)) \text{ mm} \\ &= \frac{\pi}{2} (4) = 6,28 \text{ mm} \end{aligned}$$

This occurs at each half cycle giving a net slipping velocity per 1/2 cycle of;

$$S_v = \frac{2.3.6,28}{\pi \cdot 0,75} = 16 \text{ mm/s}$$

We have observed pulleys with diametral differences as high as 14mm and slipping velocities up to 60 mm/s.

Fig 16 is an example.

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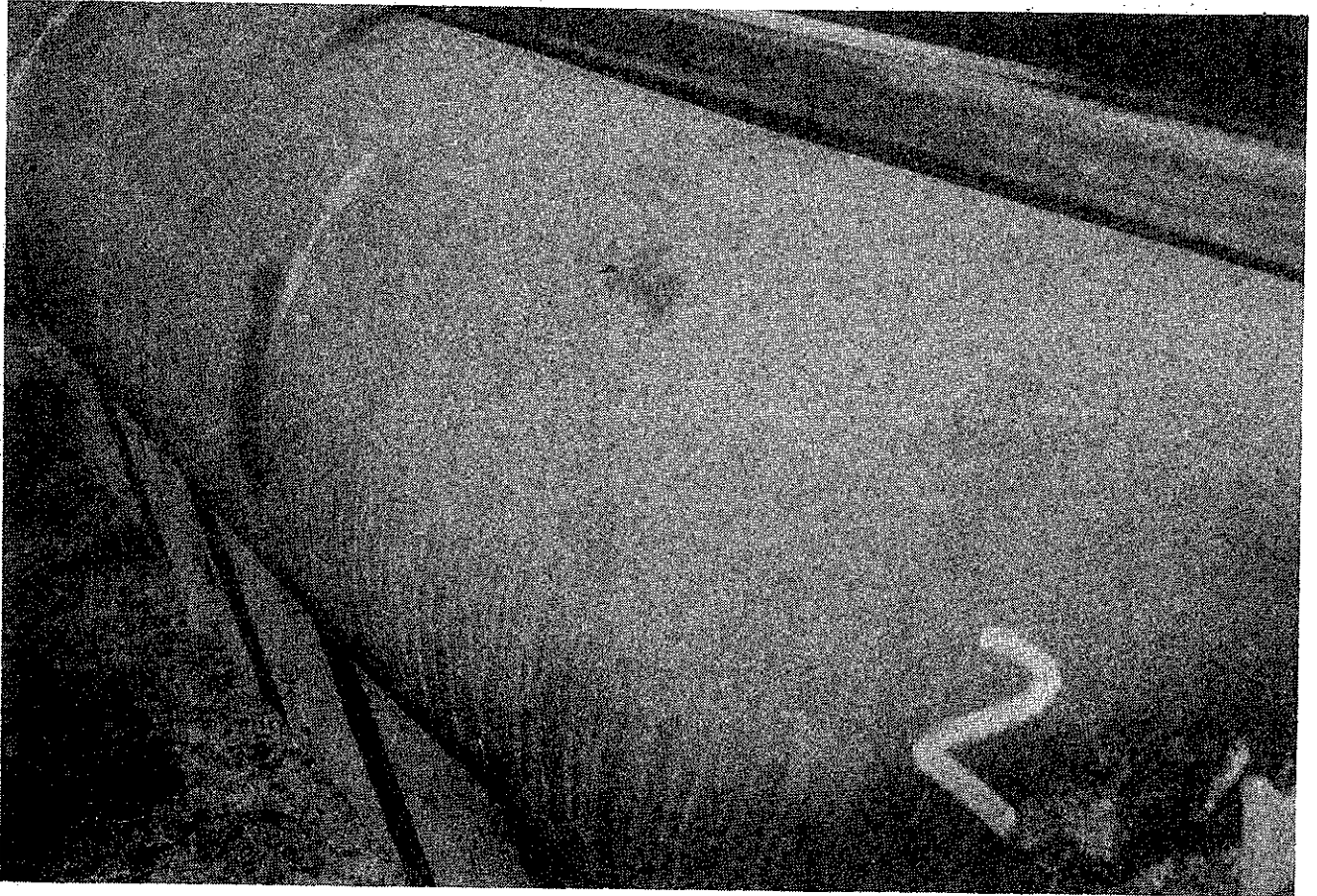


FIGURE 16 PULLEY IN FULLY WORN CONDITION

**NOTE: SHELL DEFORMATION / WEAR VISIBLE
AT THE TOP RIGHT CORNER OF THE PICTURE.
ARROWS INDICATE INNER STIFFENER LOCATIONS**

We can demonstrate that this is a real phenomenon by microscopic measurement of the wear scratches which are related to S_d , Fig 17 shows such a worn surface and the resulting scratches. We therefore have a good model for wear prediction and control.

Referring back to Fig 7a and b it can be seen that wear concentrates at the smaller diameters, it is accompanied by plastic deformation. As these effects proceed they also accelerate because of the diameter effects noted above. These effects are most pronounced in small pulleys with high belt speeds.

It is recommended that measures to reduce shell deflection such as light inner stiffeners can be considered as a mechanism to reduce wear on bare shells to a predictable level matched to plant and pulley fatigue life.

At the later stages of wear the inner stiffeners will also act as protection against impact damage which is accounted for only in the load factor.

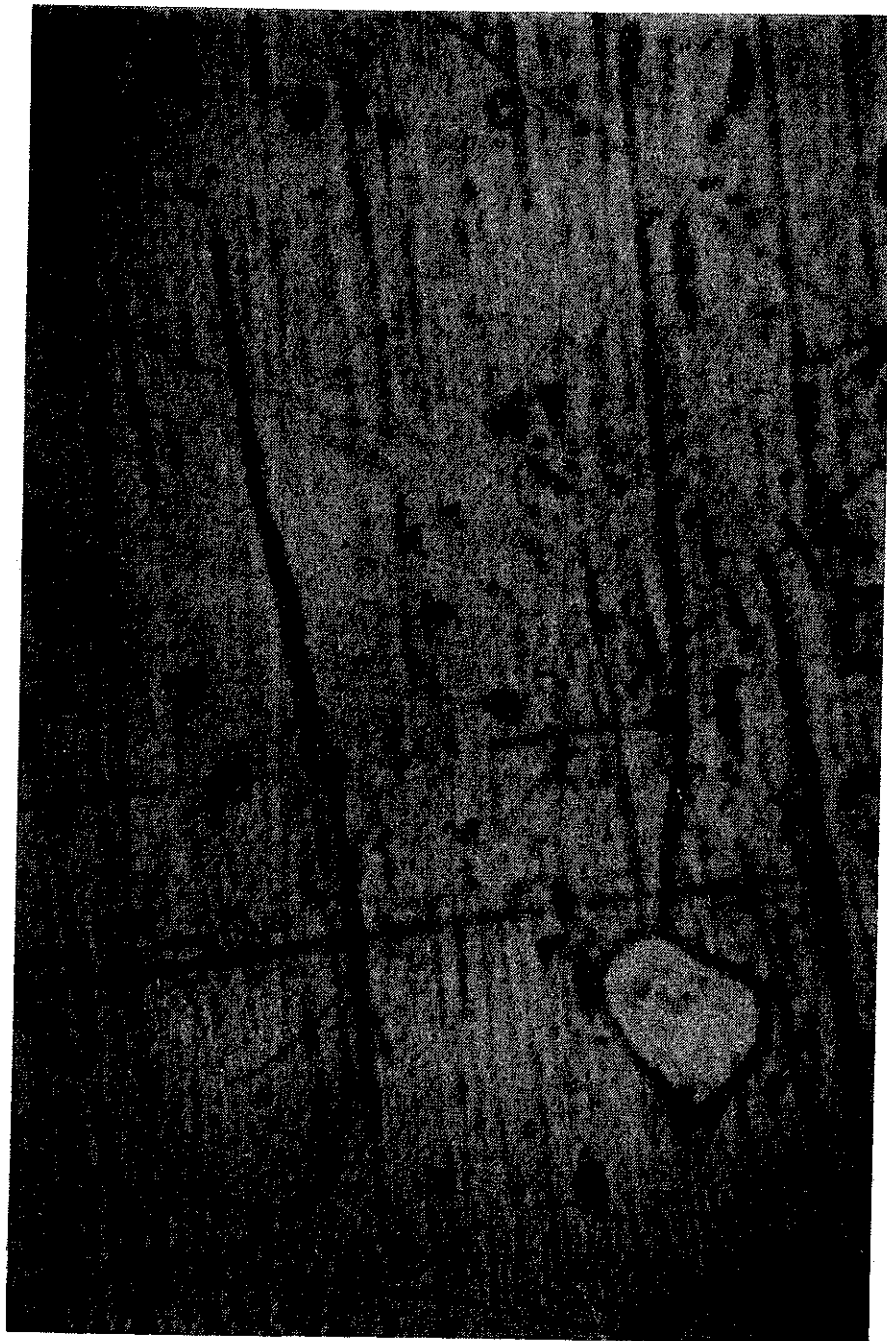


FIGURE 17 TYPICAL WEAR MARKS ON PULLEY SURFACE

40 X MAGNIFICATION

**MEASURED LENGTH OF MARKS VARIES ACROSS
PULLEY FACE IN RESPONSE TO SHELL DIAMETER**

7.0 THE FUTURE OF SHELL DESIGN

7.1 Analytical Targets

The qualitative correspondence between the crude analytical model used and the phenomenon observed in practice is good but can be improved.

Outside the scope of this paper methods for direct analysis of belt pressure and shell bending effects is being performed so that the modelling of shell axial stress and cyclic stress patterns can be refined to give better deformation models.

7.2 Finite Element Models Compared

As a step toward this, finite element methods have been applied to a typical pair of pulleys and the analytical results compared. Fig 18a and b illustrates the correspondence between the results of the given analytical method and the more sophisticated method.

The differences in stress level adjacent to the hub reflects the modelling of shell/end tilt in the more precise method. The level of conservatism of the analytical method is not always as high as in this example.

The extremely high costs of local finite element modelling (several thousand Rand per study) prevents adopting such methods for regular pulley analysis. The example illustrates very well that the analytical model does provide a usable alternative.

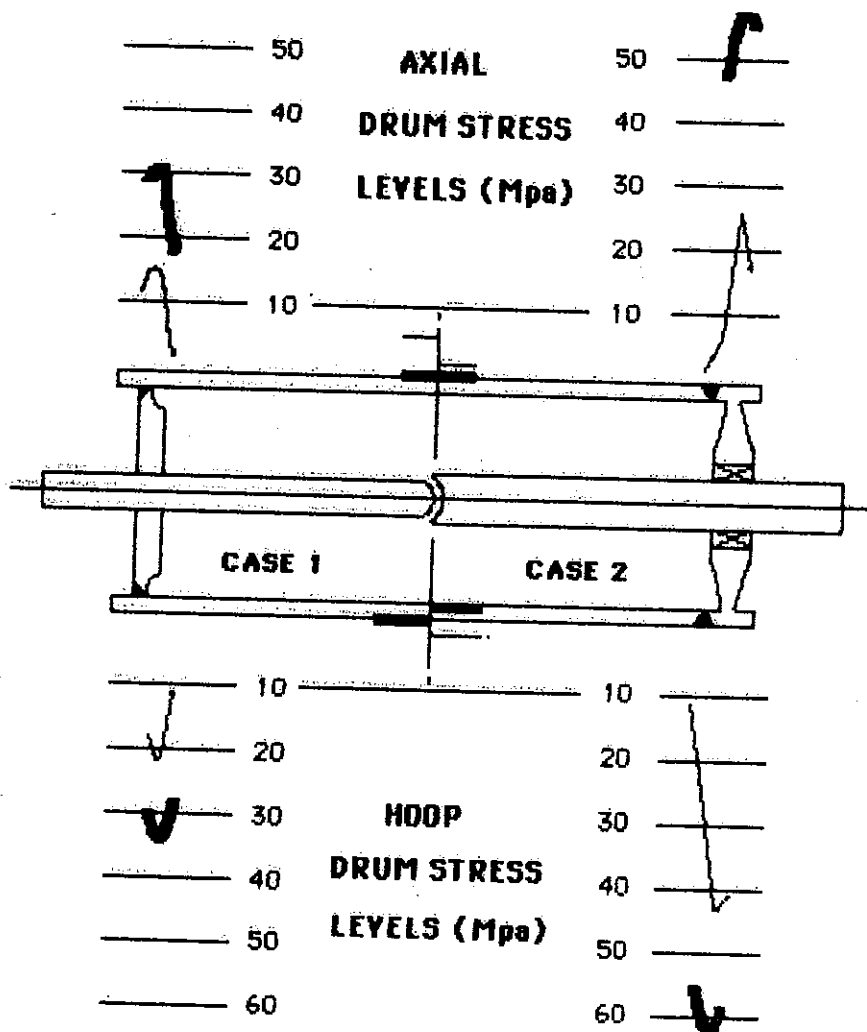


FIG 18

**COMPARISON OF MODEL
AND FINITE ELEMENT
SOLUTIONS FOR TWO
TYPICAL PULLEYS**

MODEL RESULT **—**
FE RESULT **—**

DATA	CASE 1	CASE 2
P1	84.8	192.5
P2	37	117.6
WRAP	210	210
LOAD FACTOR	1.5	2.0
DIAMETER	1000	1000
SHAFT	200	260
FACE	2300	2300
BRG CRS	2800	2800

CONCLUSION

We hope to have demonstrated that the shell as well as other pulley components can be designed to last the life of the plant. With all that this means in spares philosophy and life cycle costing.

The conveyor designer has an important role to play in respect of

- * Allocating dimensional standards
- * Avoiding pulleys of diameters less than three times shaft diameter

Variables such as stress relieving, lagging of non-drive pulleys, choice of shaft connection, inner stiffeners and material thicknesses should be discussed with the pulley designer for best results on any contract.

A new construction - the 'L' bottom pulley (Fig 14) can provide a great measure of protection at low cost against an increasingly common mode of failure.

Other constructions require great care in their design, assembly, welding and quality monitoring to prevent repeated failures of the pulley shell adjacent to the end.

Even the mechanism of wear on pulleys can be controlled by careful design. An increasingly important factor in the face of diminishing resources.

A C K N O W L E D G E M E N T

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