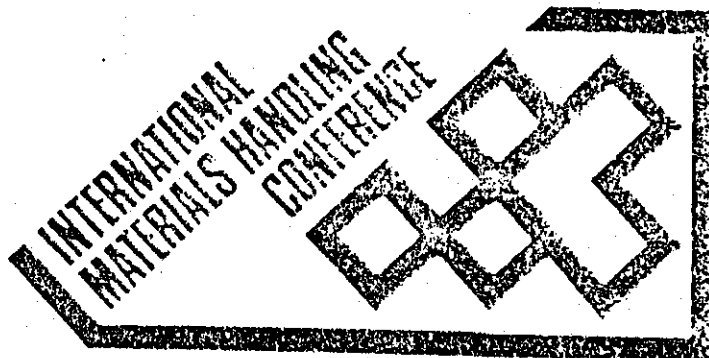


The Organising Committee  
International Materials Handling  
Conference  
P O Box 95194  
Grantpark  
Johannesburg  
2051  
SOUTH AFRICA

Telephone (011) 728-4562

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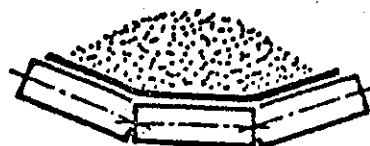


BELT CONVEYOR PULLEY DESIGN -  
WHY THE FAILURES

T J KING  
BOSWORTH STEEL STRUCTURES  
SOUTH AFRICA

18 - 19 MAY, 1983  
MILNER PARK  
JOHANNESBURG

**BELTCON 2**



BELT CONVEYORS - DESIGN, OPERATION AND OPTIMIZATION

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BELT CONVEYOR PULLEY DESIGN - WHY THE FAILURES?

TERRY KING Pr.Eng.

B.Sc Mech Eng (Hons) M.S.A.I.M.E.  
A.S.S.A.I.M.M.

Design Engineer to the Bosworth  
Group of Companies

1. SUMMARY

A system for the design and dimensioning of conveyor belt pulleys, in a manner which permits use at drawing office or computational level, is laid out.

The theoretical model is used to explain the reason for some common failures and to place in context some of the pulley construction features seen in recent years.

Lastly, an account is given of the factors which limit the life of a pulley and a design is proposed for the next generation of long-life, low cost pulleys for the South African market.

## 2. INTRODUCTION

### 2.1 Historical Context

Great strides have been made in recent years in the provision of low mass belts with ever higher tension ratings. In many cases these low mass belts permit the use of small diameter pulleys, whilst the tension ratings demand similar or increased shaft diameters.

Often the conveyor designer need not restrict himself to the limits of the belt. However, in situations such as in-seam underground conveyors, where the transfer points, tension section and drive are all of restricted height, the optimum size is required.

Similarly, in the pulley construction field, wide acceptance has been earned for the keyless shaft connections originally introduced by Ringfeder. These have resulted in a new generation of pulley constructions being adopted, designed largely by the user company or mine. It has also led to a confusion of different pulley styles, backed by conflicting claims as to their value.

## 2.2 Design Standards

The situation of designs being produced at many places in South Africa and used to and in some cases beyond their limits has lead to the requirement of a practical design standard against which the limitations of a given pulley construction can be determined. Such information is available piecemeal but is not in a readily usable form.

It is the purpose of this paper to supply such a design system, the system being justified both analytically and by comparison with failures recorded in recent years. It is a specific aim of this paper to avoid analytical techniques unsuited for use at drawing office level. Where this has required simplifications it is noted in the text.

Clearly, not every aspect of design is stress-based. Where this is the case, notes on successes and failures are given for your guidance. Lastly, an attempt has been made to place in context some of the construction features that have been appearing and disappearing in recent years. This is aimed at improving consistency and providing a basis for standardisation and new design thinking.

### 3. ANALYTICAL SYSTEM

#### 3.1 Background

A number of authors (1, 2, 3, 12, 13) working in Germany between 1963 and 1973 have contributed much to the study of pulley stresses. Unfortunately, the dissertations resulting have only recently been translated and are at a level not conveniently used in the design office. They describe systems which differ from local practice in a number of important respects.

3.1.1 Shaft and drum end materials are of significantly higher tensile strengths.

3.1.2 Pulley proportions are different, drum diameters being in the range 1000 to 1750 mm, drum widths are similar to local practice.

3.1.3 Manufacturing methods are aimed at low mass. Locally, mass has remained secondary to manufacturing economy due to a lack of standardisation and the undemanding conditions our pulleys have worked in.

The works referred to have therefore been adapted to form much of the basis for the method which follows. The main value of this method being that each part of the pulley can be treated individually with minimal repetition required.

### 3.2 Common Constructions

Figures 1 (a to d) show some common pulley constructions used locally, all of which can be analysed by the system described. Variations such as assymetrical end plates are admissible and have been shown (3) not to affect the stress pattern significantly. The shading of pulley sections indicates the stress levels encountered, dark areas being highly stressed, and will be referred to in the text.

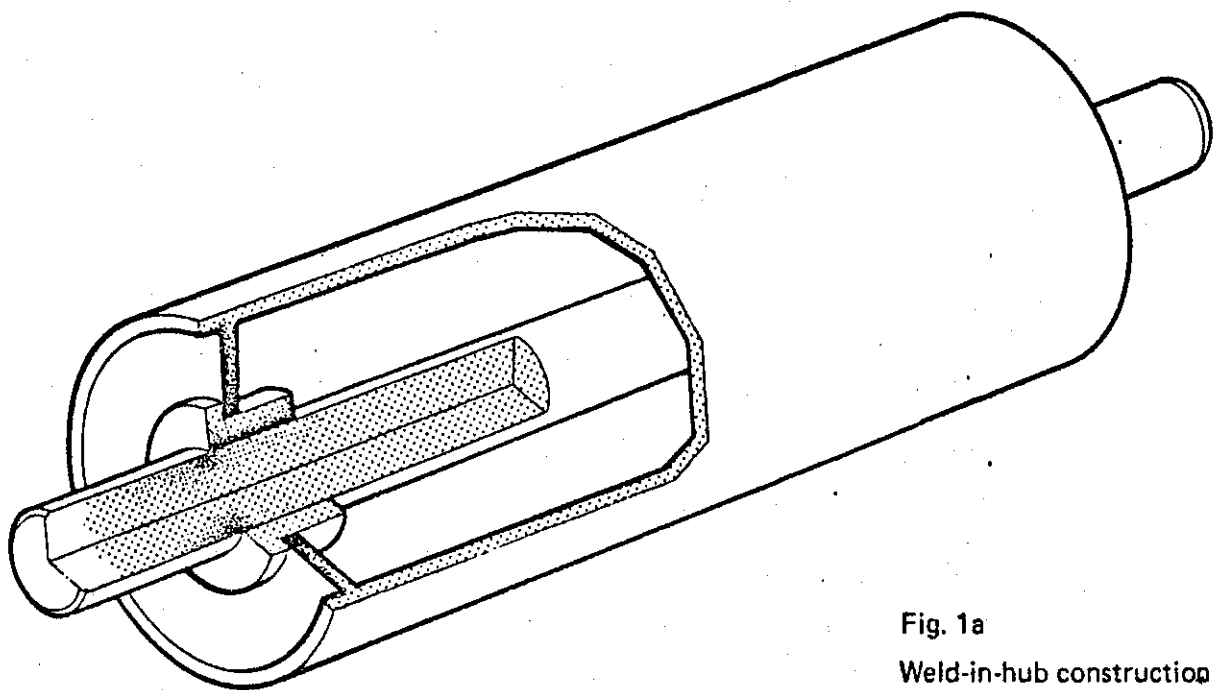


Fig. 1a  
Weld-in-hub construction

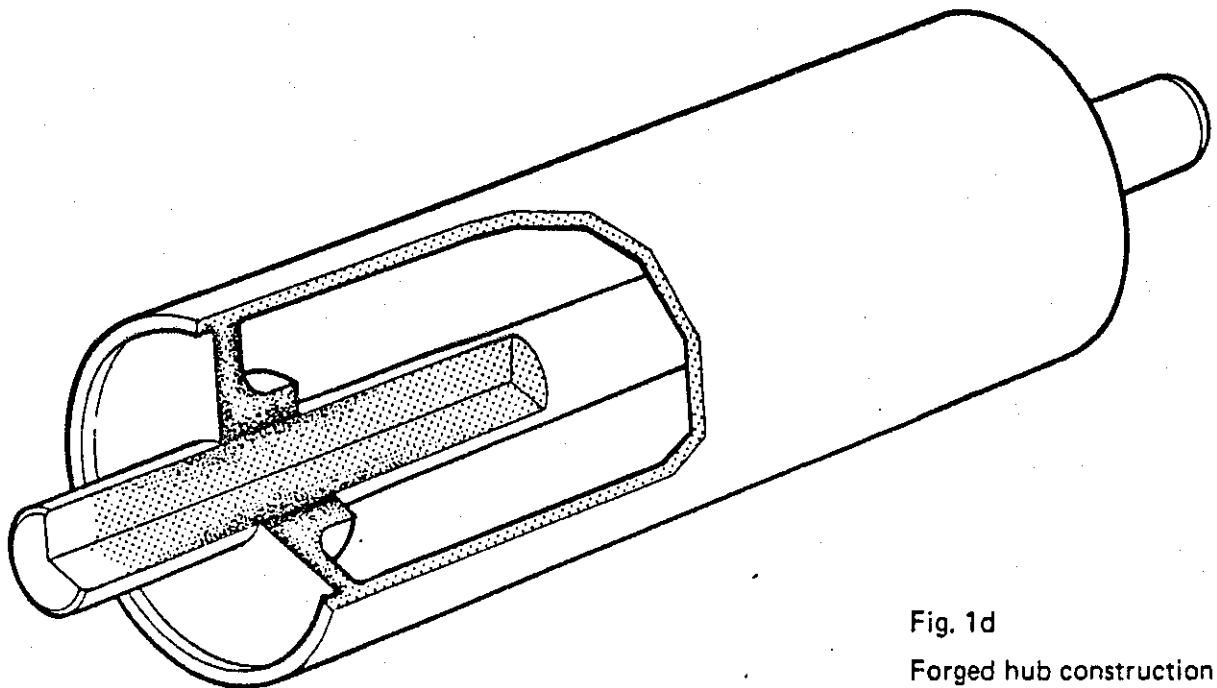


Fig. 1d  
Forged hub construction



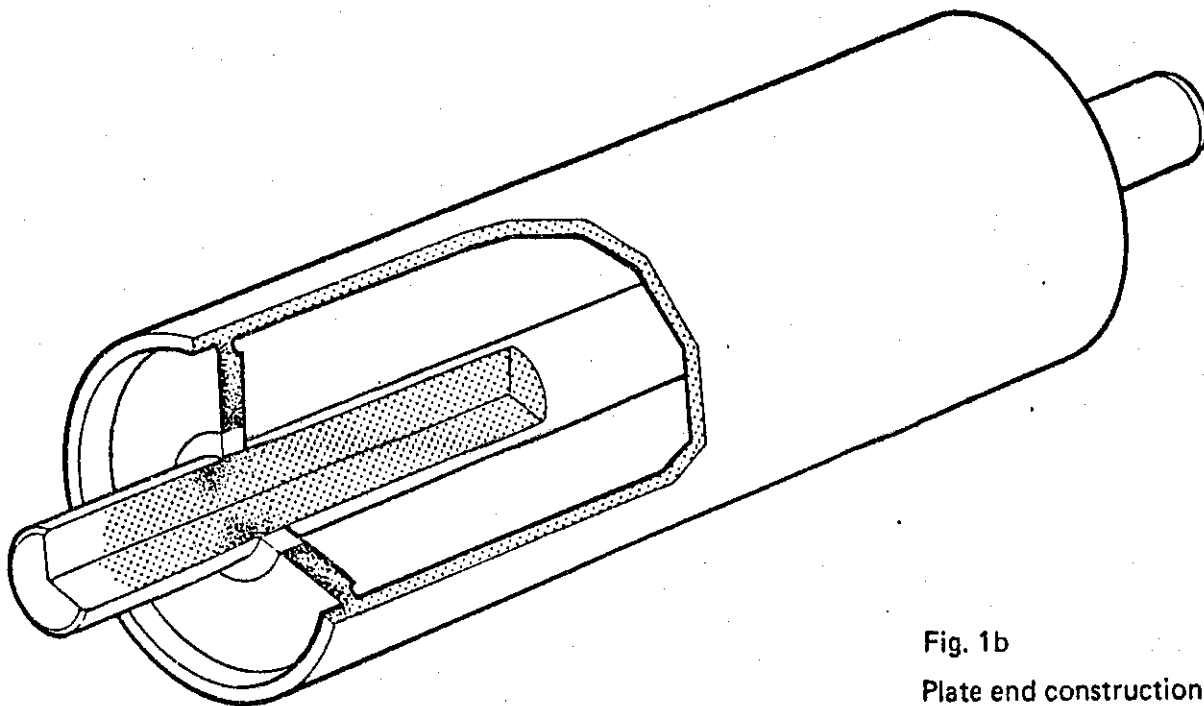


Fig. 1b  
Plate end construction

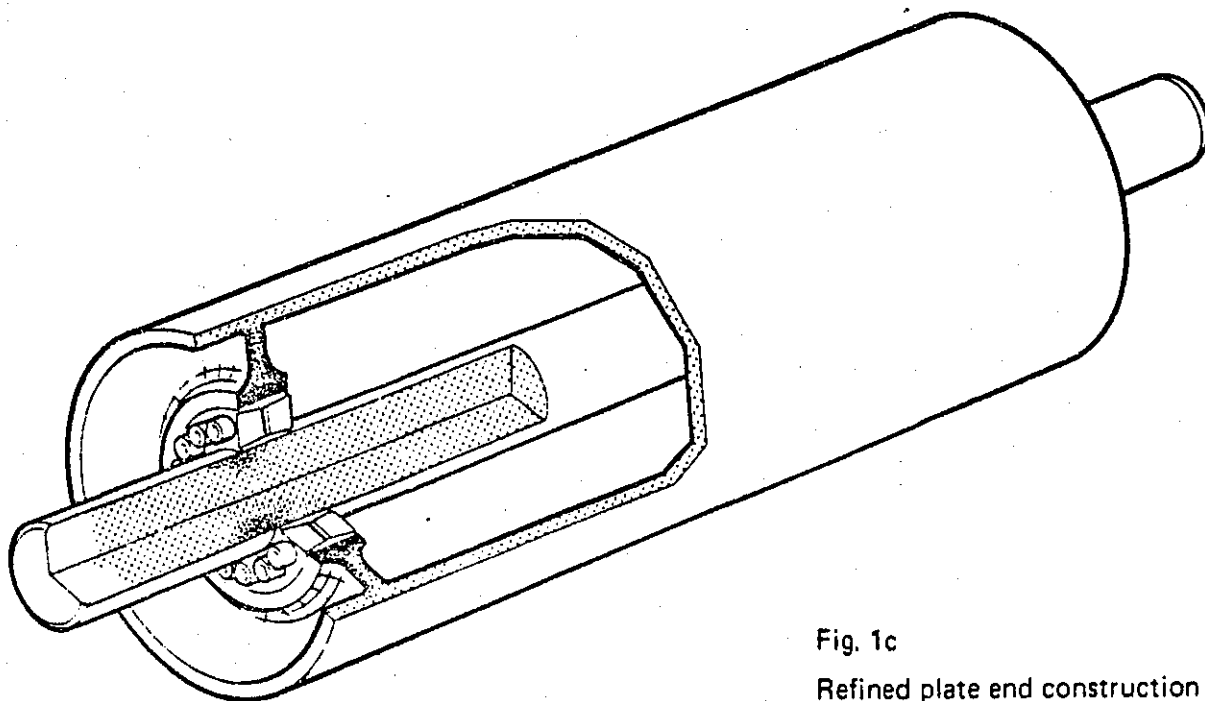


Fig. 1c  
Refined plate end construction

### 3.3 System of Analysis

#### 3.3.1 Shaft Sizing

The shaft must be sized both for the stresses at its point of entry to the hub and for its deflection.

Find Drive Torque T, if any, and shaft bending moment M in the usual way making allowance for duty factors.

From these find the following:

Combined Torsion Moment  $T_e = T^2 + M^2$  1.

Combined Bending Moment  $M_e = \frac{1}{2}(M+T_e)$  2.

Using these moments in the Torsion and Bending Equations gives the first two possible diameters (4).

Torsion Based Diameter  $d_r = \sqrt[3]{\frac{T_e \times 16000}{\tau_s \times \pi}}$  3.

Bending Based Diameter  $d_b = \sqrt[3]{\frac{M_e \times 32000}{\sigma_s \times \pi}}$  4.

#### Note

$\sigma_s$  and  $\tau_s$  are taken to be equal since the allowable direct principle stress  $\sigma_s$  is a fatigue case and the shear stress  $\tau_s$  is not.

Values for the principle stresses can be found from BS 153 Parts IIIb and IV : 1972 or any similar authority. Otherwise values in the range 41,5 to 45 Mpa(5) have been found satisfactory for BS 970 : 1972 070M20 (EN3A) steel, the most common shaft material.

A third possible diameter should always be determined from the "free" shaft deflection as below:

$$\text{Deflection Based Diameter } d_d = \sqrt[4]{\frac{W \cdot a \cdot L \cdot 1600}{E \cdot \pi \cdot \alpha}} \quad 5.$$

Where W = Nett Tension without duty factor KN

a = Bearing centre to hub distance mm

L = Hub spacing mm

E = Youngs modulus for shaft N/mm<sup>2</sup>

$\alpha$  = Allowable deflection radians

Convenient deflection limits are in the region of 0,001 radians for shaft connection hubs and 0,01 radians (8) for shrink fit hubs.

The largest of the three possible diameters should now be chosen or a reasonable compromise reached. If it is a drive pulley the shaft connection can be selected using the drive torque. For non-drive pulleys any appropriate connection can be chosen using a suitable guide (11).

### 3.3.2 Hub Diameter

In this system the hub is first sized by maximising the circumferential stress at the inner edge, using Lamés equations (10) suitably transposed as follows:

$$\text{Hub Outside Diameter } d_o = d_i \sqrt[2]{\frac{f_c + q}{f_c - q}} \text{ mm} \quad 6.$$

Where:

$d_i$  = hub inside (connection outside) diameter mm

$f_c$  = max hub stress allowable (eg 80% of yield) N/mm<sup>2</sup>

$q$  = connection to hub pressure from catalogue N/mm<sup>2</sup>

This establishes the minimum hub size  $d_o$ . It is now necessary to check the circumferential stress at the outside of the hub to ensure that excessive stress is not applied to the next component, whether it is a weld or a diaphragm. For this, apply the equation:

External circumferential hub stress =  $f_c^1$

$$f_c^1 = 2 q d_i^2 / (d_o^2 - d_i^2) \quad \text{N/mm}^2 \quad 7.$$

Now one of several situations will occur.

- \* The hub will already be too large to fit the pulley. In this case use one of the special low pressure connections (see (11)) and start 3.3.2 again or use a larger pulley.
- \* The hub will fit in the pulley but the outer edge of the hub approaches the drum attachment weld. This is a good reason for adapting a construction such as Fig 1b. Note that the allowable external tangential stress must be reduced to allow for weld imperfections. A low pressure connection can be advisable to assist in this. Note also that for this construction, the hub stresses are additional to the diaphragm stresses dealt with below.
- \* The hub is an acceptable proportion of the pulley diameter and a construction such as 1a, 1c or 1d is appropriate.

### 3.3.3 Construction Choice

Choice between 1a, 1c and 1d depends largely upon shaft attachment method. Schemes 1a and 1d have extended hubs in order to distribute the very high radial attachment forces from shrink

fits along the shaft without local material collapse. These are unnecessary when using shaft connections, purely serving to add cost. In this case 1c is indicated or 1b for economy. If shrink fits are retained, as they should be for maximum economy, then a construction such as 1a is indicated and for heavier pulleys 1d, the major value of 1d is to eliminate the diaphragm to hub weld from a very highly stressed area.

#### 3.3.4 Shrink Fit Hubs

It will be noted that no attempt has been made to size shrink fit hubs, this topic could be a paper in itself. Shrink fits are extremely sensitive to manufacturing tolerances and cannot practicably be dealt with without the collaboration of the manufacturer. It is suggested therefore that the identical process described in 3.3.2 be used to size these hubs or that a particular manufacturers data be adopted.

#### 3.3.5 Hub Width

Chosen, in the case of shaft connectors, as the minimum hub width that will secure the element. This is sufficient.

If the pulley is of a construction having no separate hub such as Figure 1b then ignore this section and proceed to 3.3.6.

#### 3.3.6 Diaphragm Stress Components

The diaphragm of a pulley is subject to a number of loads. These can be summarised as follows:

- \* Radial Bending Moments due to shaft deflection.
- \* Radial direct stress due to pulley load.
- \* Tangential shear stress due to drive torque.

In addition there may be a radial direct stress due to shaft connection loads.

The first simplification is to ignore the Tangential shear component. These have been found to be negligible for the material thicknesses involved, as in Figure 2a.

Secondly, the approach suggested by some authors for radial direct stress (12,3) was adopted after checking against experimental results (3). It equates to a simple projected area approach, thus;

$$\text{Radial Direct Stress } f_d = \frac{W}{2d_o t} \quad \text{N/mm}^2 \quad 8.$$

Where:  $t$  = diaphragm thickness.

This stress is equal compressive and tensile on opposite sides of the pulley (3).

Radial Bend Stress is derived as follows:

Firstly relative stiffnesses of shaft and drum are established.

$$\text{Shaft Stiffness Constant } K_6 = \frac{\pi \times d^4}{32(L-200)} \quad \text{mm}^3 \quad 9.$$

Where:  $d$  = shaft diameter.

Diaphragm Stiffness Constant = K5

$$K5 = \frac{2.73}{\pi} \left( \frac{1 - R^2}{1 + R^2} + \log(R) \right) \text{ mm}^3 \quad 10.$$

Where: R = ratio of diaphragm diameters =  $\frac{d_i}{d_o}$

Then, a suitable thickness is chosen by setting radial direct stress to one third of the allowable fatigue stress and finding an estimated t value from Equation 8. Next the actual diaphragm stiffness is calculated, for this t value as below:

$$\text{Diaphragm Stiffness } K7 = \frac{t^3}{K5} \quad 11.$$

If the drum is assumed infinitely stiff compared with shaft and diaphragm (reasonable since stiffness is the fourth power of diameter, as supported by experimental results (1)). Then the Bending Moment is distributed in pulley end and shaft in proportion to stiffness, as follows:

$$\text{Diaphragm Bending Moment } M_d = \frac{K7 \cdot M}{K7 + K6} \quad \text{Nm} \quad 12.$$

Now the pulley end can be treated as a flat plate having rigidly supported edges (3,2,13) with good accuracy.

$$\text{Flat plate constant } K8 = \frac{3}{\pi \cdot R} \left( \frac{1 - R^2}{1 + R^2} \right) \quad 13.$$

and finally

$$\text{Diaphragm radial bend stress } f_b = \frac{M_d \cdot K8 \cdot 2000}{d_o t^2} \quad 14.$$

Maximum fluctuating stress occurs on the pulley end at the diaphragm/hub joint, here the peak stress is as follows:

$$\text{Peak Stress } f_{\max} = f_r + f_b + f_c$$

Where:  $f_r$  = shaft connector pressure, if appropriate.

$$\text{Minimum Stress } f_{\min} = f_r - f_b - f_d \quad (+ve \text{ or } -ve)$$

From these stresses and the fatigue tables the thickness used can be assessed and equations 8. and 11. to 14. revised if needed.

The thickness of the diaphragm may be increased at will, since the effect of diaphragm thickness on stress level is shown in Figure 2a worked in the manner above for a pulley of typical proportions. The so-called "flexible" end pulley does not exist except in drums of the proportion of mine-winders, e.g. Figure 2b where a significant low stress at low thickness region is visible.

The refined profile shown in Figure 1c is derived by using the calculation system above to find a minimum thickness at the hub and then reducing this in the ratio of the diameters outward toward the drum. Together with large blend radii at the hub diameter, this achieves the most even stress distribution feasible.

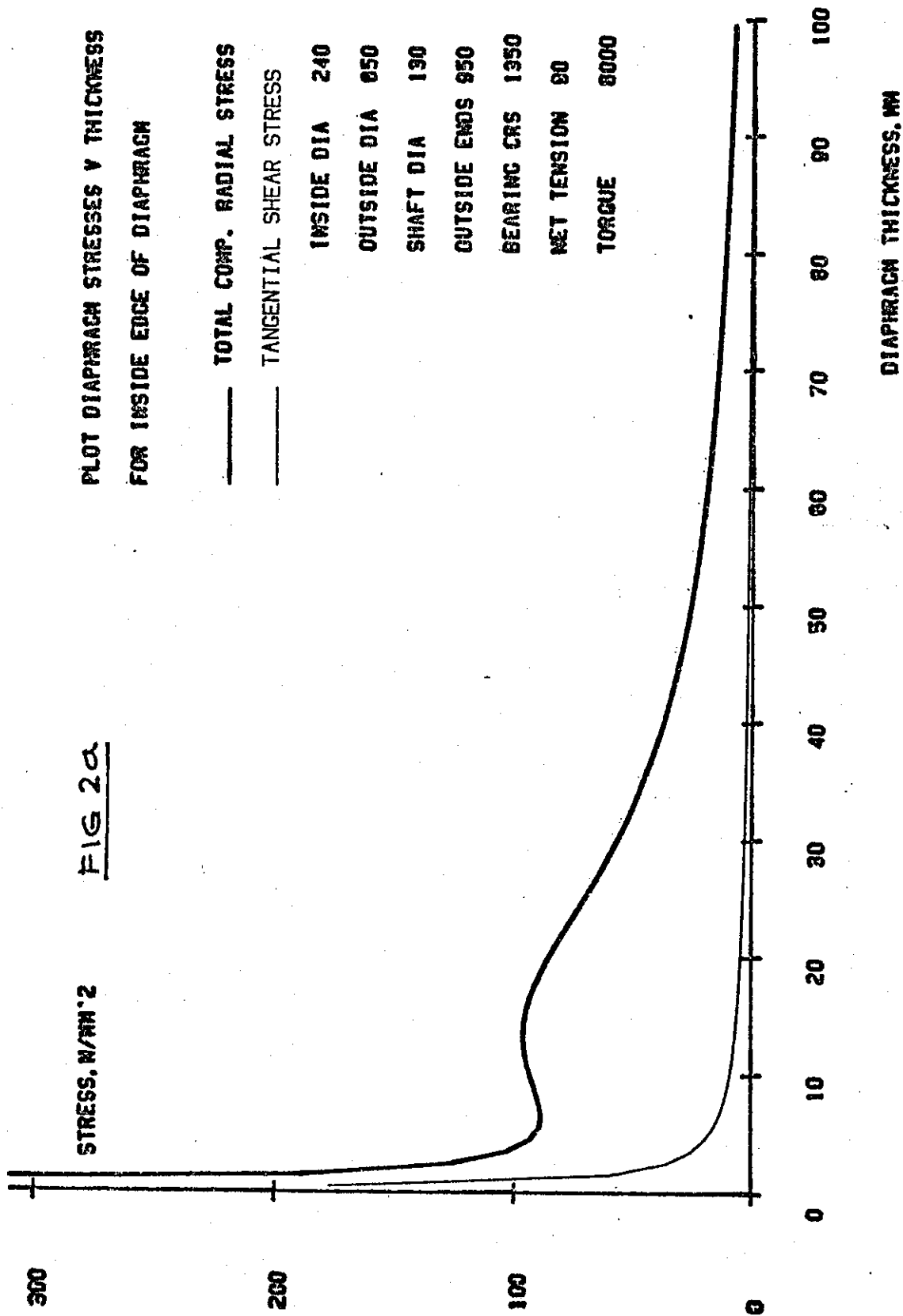
### 3.3.7 Pulley Drums

Several authorities (1,3,10) have devised systems for assessing the stress level in the drum surface. However, the data provided when extended for local pulley proportions, results in drum thicknesses that are impractically thin from the wear point of view. For example, 4mm plate for an 1100 mm wide pulley with 304 KN resultant tension. Both test results and observations of pulleys after service confirm that the theory is correct.



It is recommended that drum thickness is therefore considered carefully together with lagging and inner diaphragms in the light of the points made in 4.3

**FIG 2a**



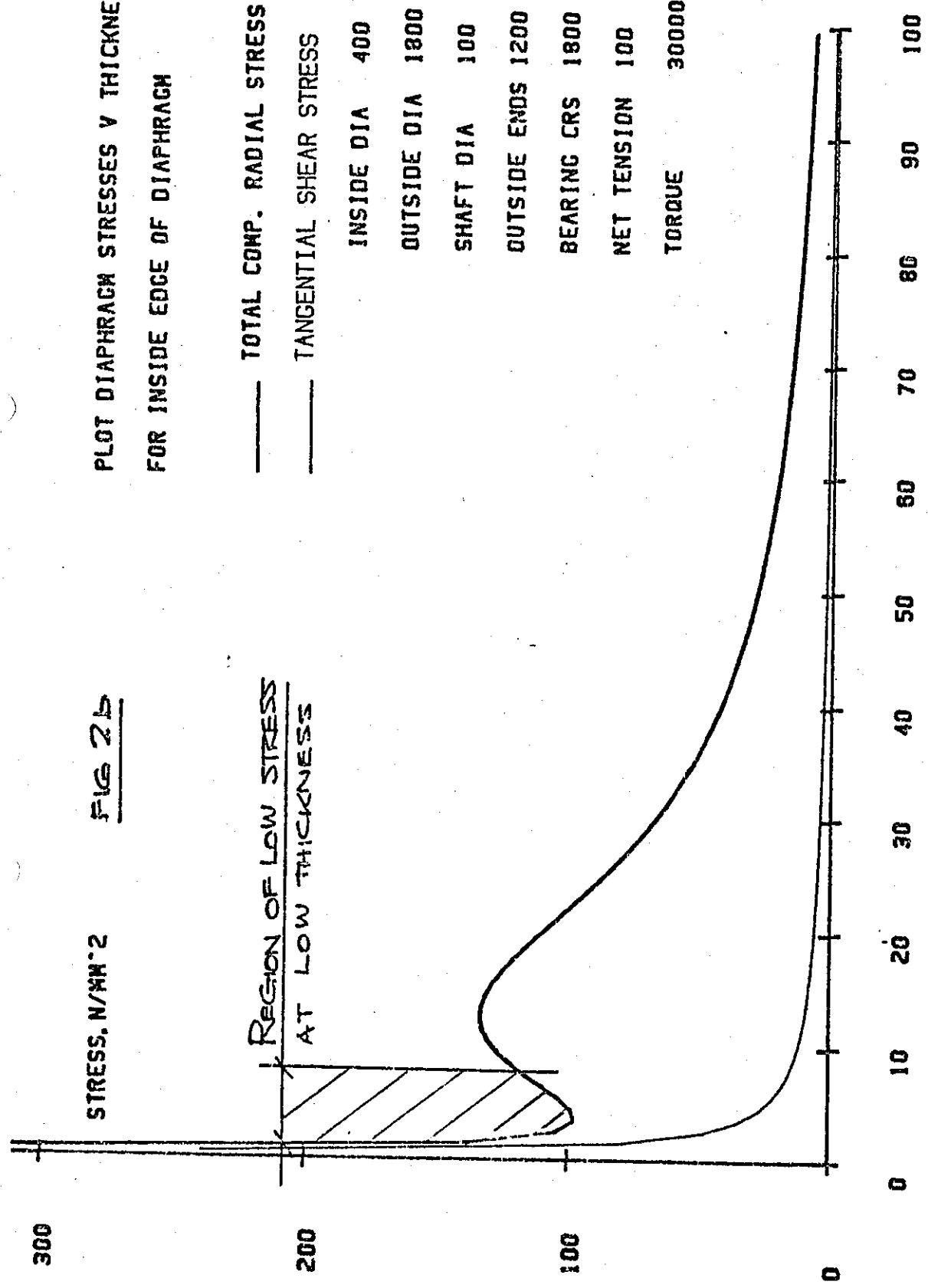
STRESS, N/MM<sup>2</sup>

FIG 2b

PLOT DIAPHRAGM STRESSES V THICKNESS  
FOR INSIDE EDGE OF DIAPHRAGM

- TOTAL COMP. RADIAL STRESS
- TANGENTIAL SHEAR STRESS
- INSIDE DIA 400
- OUTSIDE DIA 1800
- SHAFT DIA 100
- OUTSIDE ENDS 1200
- BEARING CRS 1800
- NET TENSION 100
- TORQUE 30000

REGION OF LOW STRESS  
AT LOW THICKNESS



DIAPHRAGM THICKNESS, MM

## 4.1 Shaft and Shaft Connection Related Failures

### 4.1.1 Shafts

Many authorities are tending to frown on use of reduced journal diameters to the shaft end. The practice of "journalling" arose from the significant bearing housing cost savings which can be achieved on large shafts. These savings are still available at no cost in bearing life. The argument against journalling is one of practice not principle, the following faults still arising frequently:

- \* Journals placed too close to the hub, so that the reduced diameter and increased stress as described by Peterson (7), apply in Equations 3. and 4.
- \* Radius and surface finish required are not specified by the designer.
- \* Manufacturing leaves significant stress raisers on the stepdown radius.
- \* Step downs are used on shafts whose bearing costs are negligible.

Figures 3 a and b demonstrate good and bad practice in this field. We suggest that journalling may be not only an economical practice but also a possible means of standardising pulley mountings across a conveyor scheme without excessive capital cost and are therefore a useful practice.

### 4.1.2 Shaft Connections

Pulley shaft connections are often hailed as the answer to a maidens prayer.

#### 4. FAILURES AND SUCCESSES

During 1982/83 all pulleys returned to Bosworth Steel Structures for refurbishing and replacement were examined for mode of failure. Some typical examples are given below in the light of the foregoing analytical system, with comments on construction trends in the industry.



FIG 3a A WELL RADIUSSED SHAFT AND HUB



FIG 3b A FAILED SHAFT WITH SMALL RADIUS AT THE STEP-DOWN

This is justified provided the following points are born in mind:

- \* Shaft connections depend totally on friction for their effect. They demand the same respect as bearings in their application. Machining tolerance, surface finish and contamination are all potential causes of failure.
- \* Shaft removal after service is often possible (11) but the shaft and element are likely to require renewing, since corrosion severely affects the operation of the element.
- \* Shaft sizes must often be increased up to 12% due to the deflection limitation. Specification of high strength steel, does not improve this. (See equation 5).
- \* Small size connections, up to 120 dia. have high shaft to connector pressure as appropriate to the European shaft materials. Compared to EN3A these pressures of up to 250 Mpa(16) before pulley loading are excessive and increase the risk of damage as in Figure 4 when fitting the shaft.
- \* Connections are far too loosely specified without regard to their properties.

If all this is taken into account failures such as that in Figure 5 can be avoided. It is suggested that the shrink-fit not be abandoned too readily, particularly for light pulleys, as this is still the most economical connection method and considerable experience has been built-up in its use. As one authority (17) says "For highly stressed shaft-hub connections the shrink-fit is unsurpassable ..... regarding fatigue strength under alternating torsional stress".

#### 4.2 Drum End Influences

The theory advanced by some commentators (14) that a thin "flexible" diaphragm is appropriate, results from taking the radial bending stress out of context. Figure 2a demonstrates that a region where stress increases with thickness does exist, but is insignificantly small except in very large diameter pulleys with extraordinary bearing centres (Figure 2b). This is born-out in practice by Figures 6 and 7a which show two typical cases of very light diaphragm pulleys, failure having occurred at the inside and outside diaphragm welds respectively, due to radial fatigue stresses. Failure in Figure 7a may have been hastened by shrinkage stresses in the weld. Shrinkage stress is known to be a hazard particularly in small diameter pulleys. Stress relieving is recommended to off-set this problem in critical applications.

The opposite end of the scale is shown by Figure 7b, one of a range of pulleys with intermediate diaphragm thicknesses. We are not aware of a single reported failure of such a pulley.



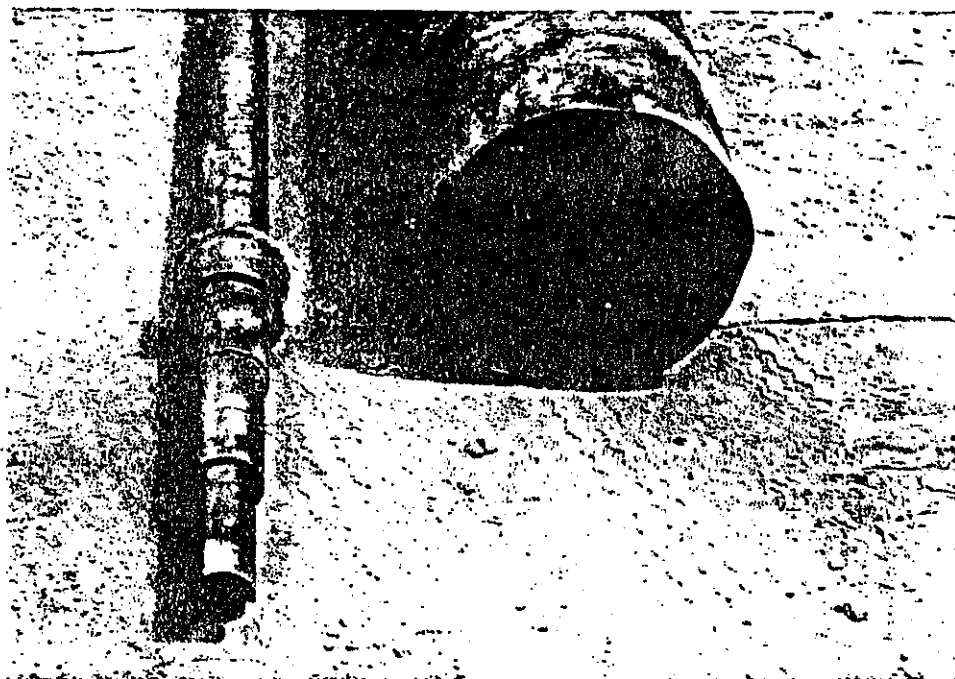


FIG 6 FAILED HUB/DIAPHRAGM WELD IN THIN DIAPHRAGM PULLEY



FIG 7a FAILED DIAPHRAGM/SHELL WELD IN THIN DIAPHRAGM PULLEY  
(Arrow shows line of weld fracture)

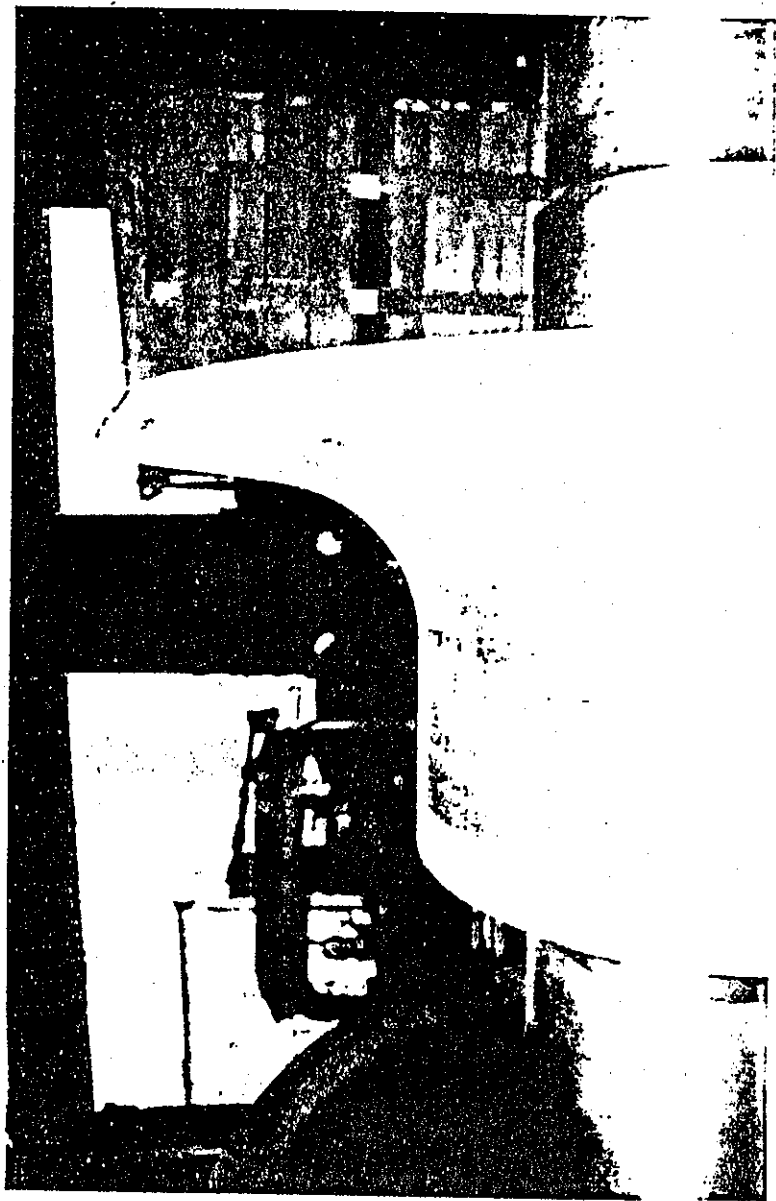


FIG 7b A MEDIUM-HEAVY DIAPHRAGM DURING PULLEY ASSEMBLY

#### 4.3 Drum Surface Phenomenon

When talking of drum surface wear, "failure" is a contradiction in terms. The time at which a drum surface becomes unusable due to wear legitimately determines the useful life of the pulley. If we assume that the remaining structure of a pulley can be made sound, then it becomes very important to understand the way the drum behaves in order to improve pulley performance. Sadly our understanding is very incomplete.

A few general points can however be made as outlined below:

- 4.3.1 The very low level of stress mentioned in 3.3.7 is a real phenomenon. This has been demonstrated experimentally (1, 3, 18) and is demonstrated practically in Figure 9. To the best of our knowledge, this pulley was replaced due to bearing failure, both ends.

The theory also tells us that the drum fatigue stress undergoes three cycles per revolution, not one. This is shown in Figure 8 reproduced from (3).

- 4.3.2 Inner diaphragms have a marked effect on pulley wear, Figures 9 and 10 and 11a are pulleys with none, one and two inner diaphragms respectively. In each instance the "saddleback" between diaphragms can be clearly seen. In our attempt to determine why a stiffener should affect the rate of wear we dissected some used drums. Figure 11a and 12a show two of these drums, between seven and ten years old. On measuring the dimensional changes of the drum we produced the results shown in Figures 11b and 12b.

We concluded as follows:

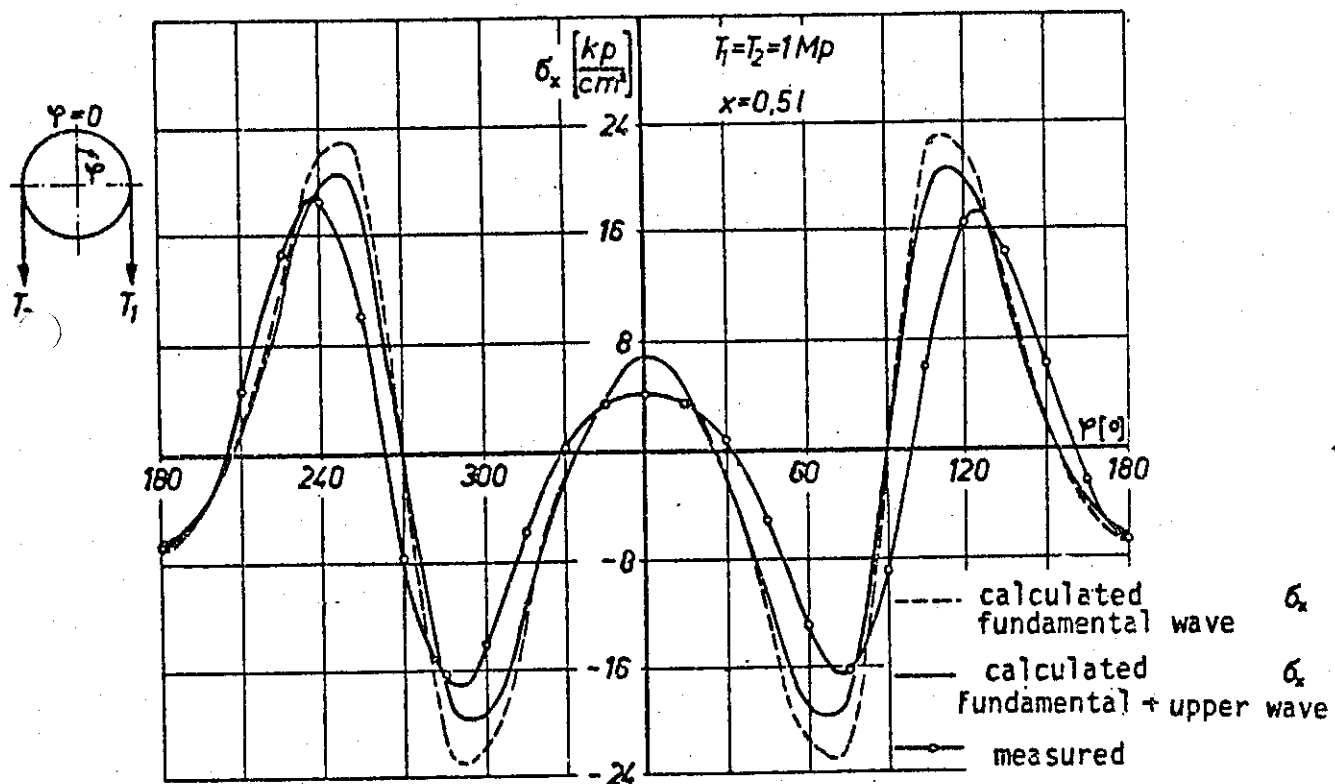


FIG. 8 : Axial stresses in the pulley shell

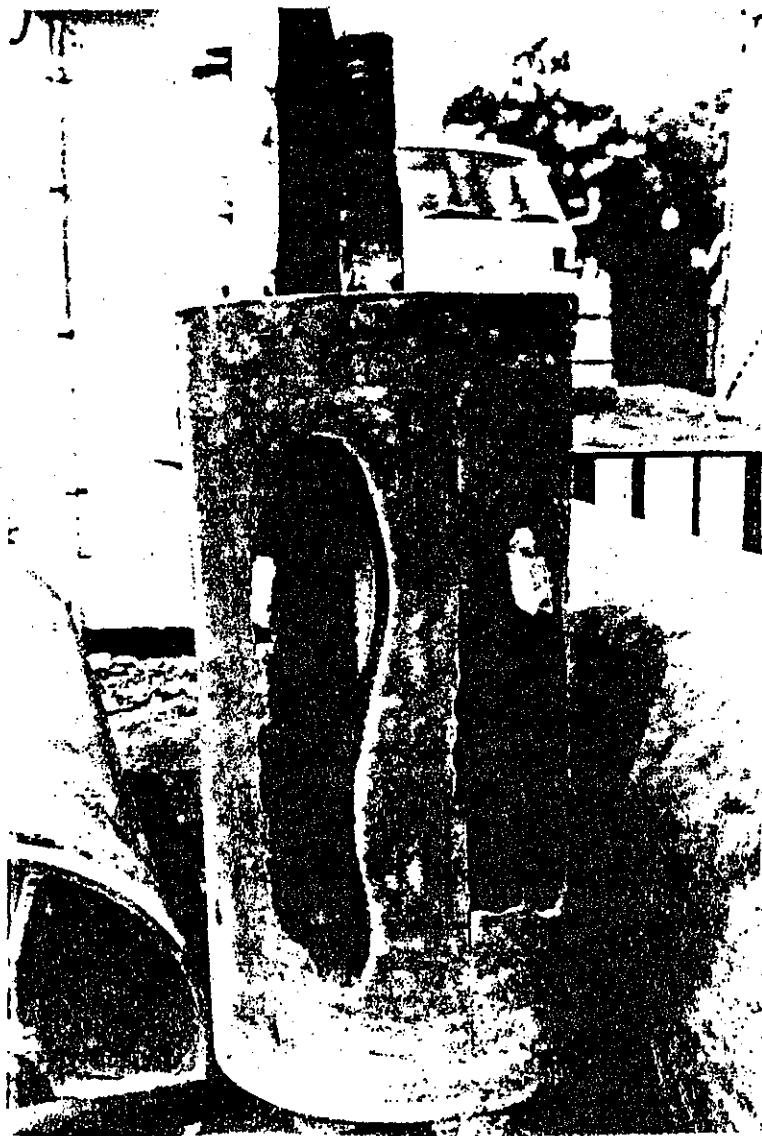


FIG 9 A WELL WORN DRUM SHOWING HOW LITTLE  
MATERIAL IS NEEDED IN A FUNCTIONING PULLEY

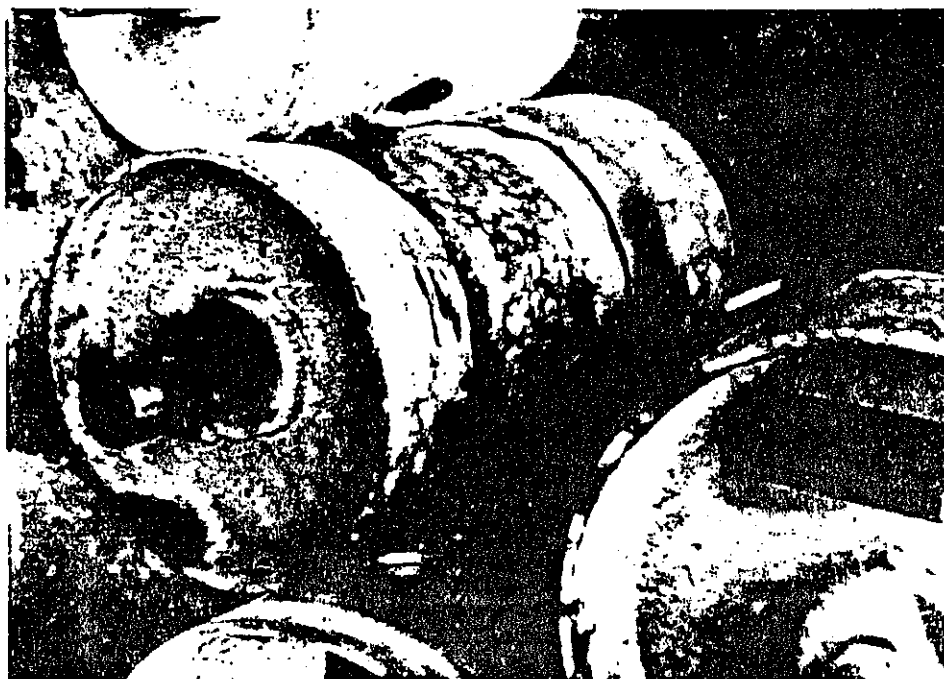


FIG 10 WEAR PATTERN IN A SINGLE INNER DIAPHRAGM PULLEY

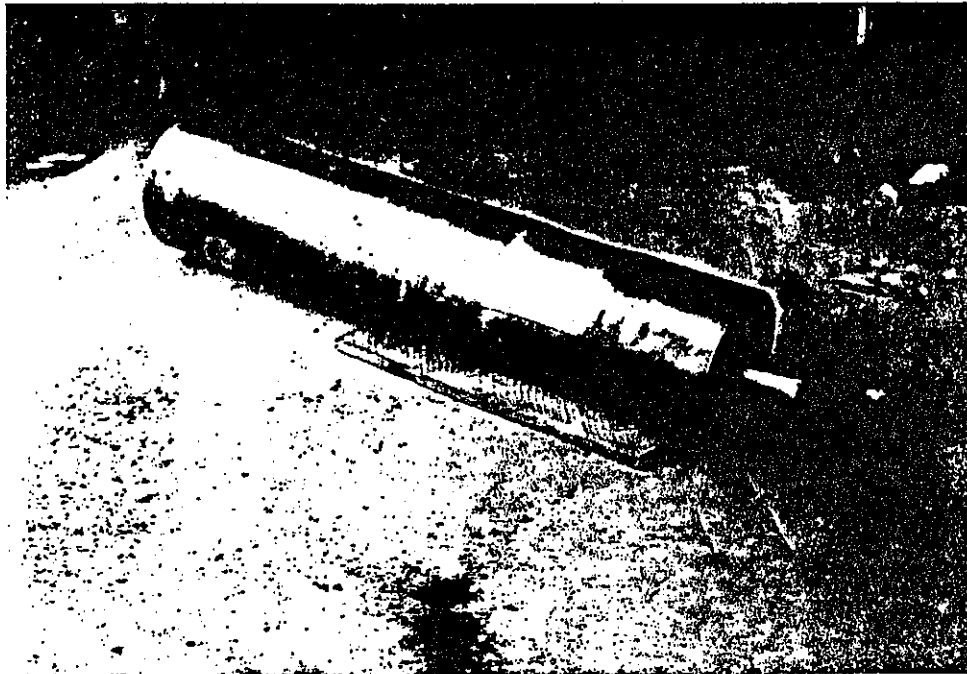


FIG 11a WEAR PATTERN ON A PULLEY SHELL (two inner diaphragms)

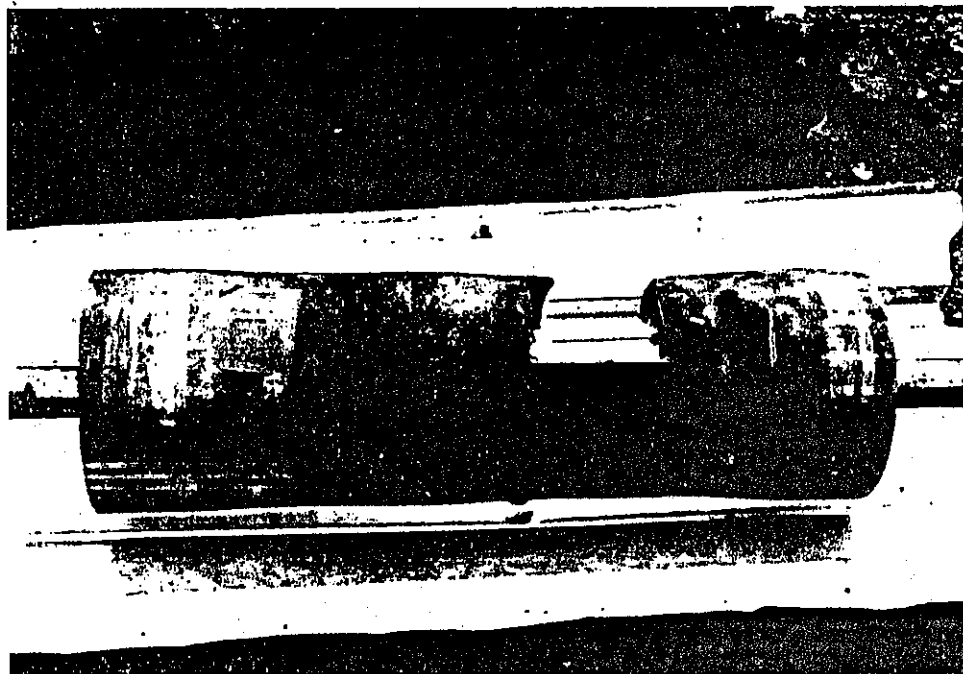
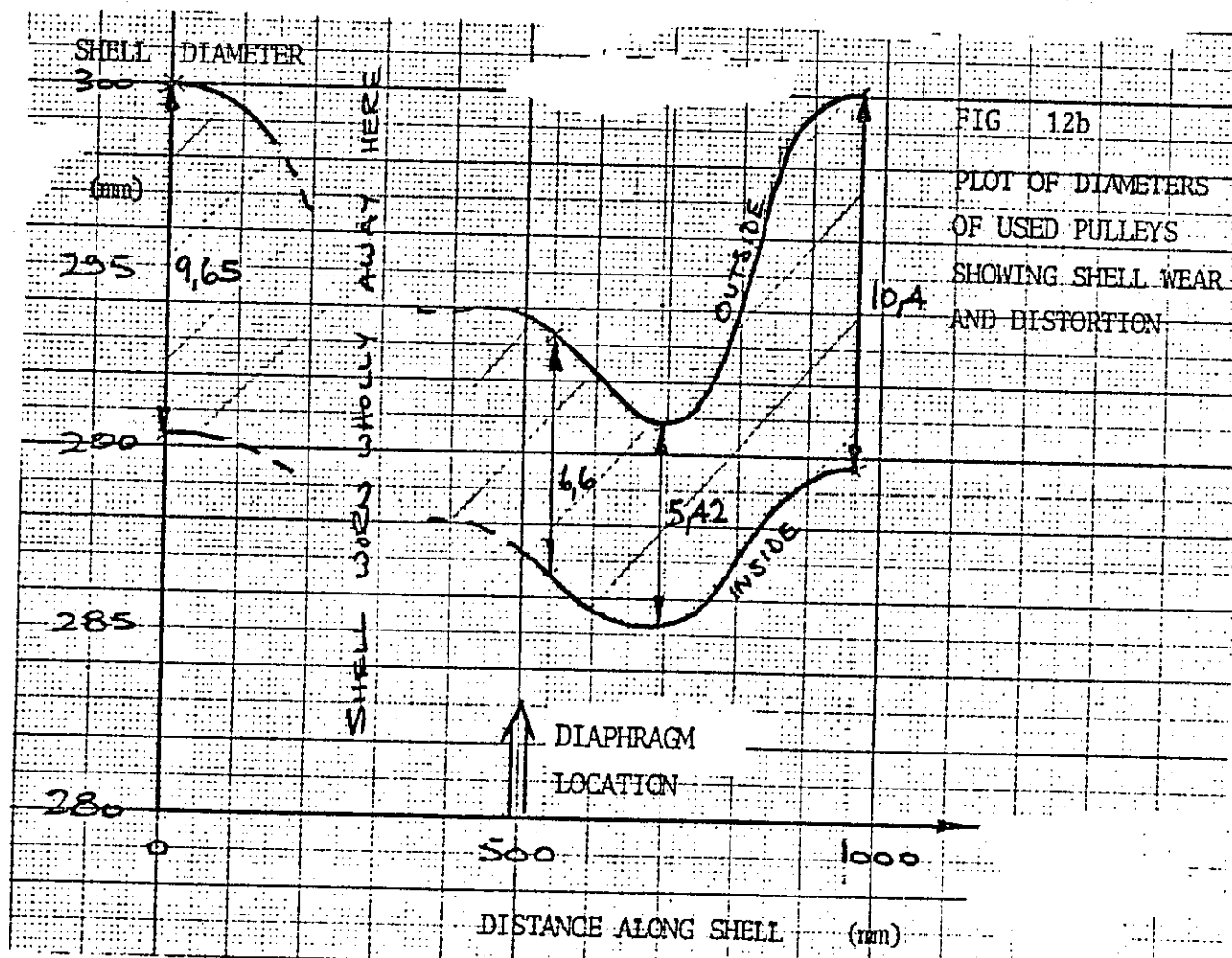
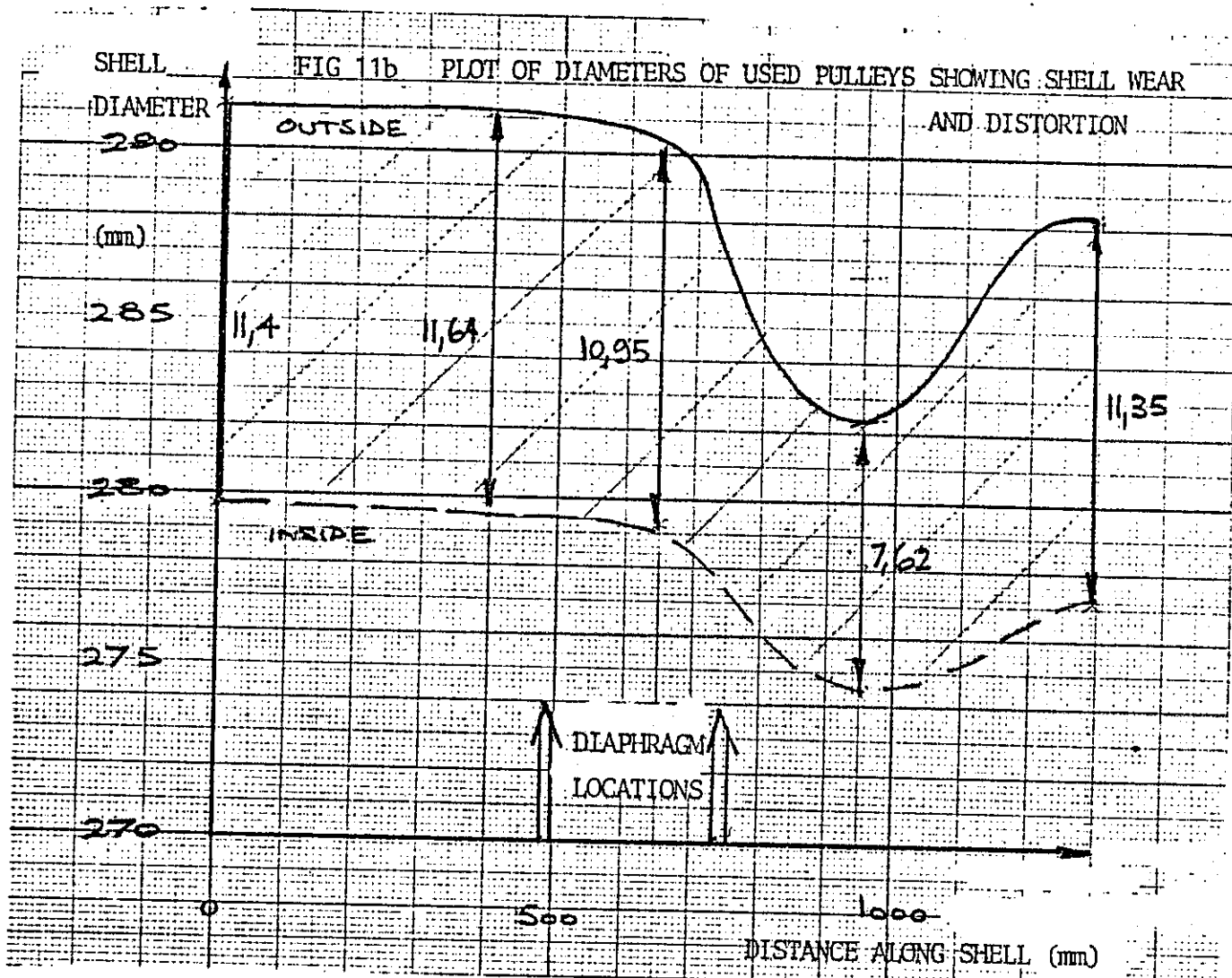


FIG 12a WEAR PATTERN ON A PULLEY SHELL (one inner diaphragm)



- \* Wear rate is asymmetrical due to uneven belt tension.
- \* The visible distortion is due in approximately equal parts to wear (metal loss) and to plastic deformation of the surface.
- \* The latter item explains the effect inner diaphragms have. The distortion is due to fatigue at low stress.

4.3.3 We can now arrive at some guidelines. Firstly, poor belt tracking is a primary cause of rapid drum wear. Secondly, inner diaphragms do help to distribute drum wear evenly. Thirdly, since the purpose of inner diaphragms is one of support (stiffness) and not one of strength, thickness is unlikely to be important. A few thin diaphragms are thus better than one thick one. Fourthly, the success of your chosen drum thickness is dependant on wear rate, for your carried material. Not on any stressing technique now available.

It is strongly recommended that all drums are lagged in order to prevent metal wear. If this is done extremely thin shells with inner diaphragms to preserve their shape can be utilised safely. Remaining plastic deformation is not of any consequence on its own. This saving in the shell has a large effect on pulley cost and moment of inertia.

Lastly two cautions are appropriate:

- \* Heavy diaphragm welds, un-stress relieved, have been a cause of drum failure and should be avoided.
- \* If you still wish to abandon inner diaphragms contemplate the results of the loose piece in Figure 10 or 13 entering your belt system. An inner diaphragm would prevent this sort of occurrence.





FIG 13 A FAILED SHELL CREATING RISK OF SEVERE BELT DAMAGE.

## 5. CONCLUSION

If we in South Africa are to move ahead of the European pulley producers a number of items are required:

- \* Shaft connection methods better suited to the mild steels we - correctly - favour using.
- \* An improved understanding of the component which determines the life of a pulley. The Drum.
- \* Manufacturing techniques which are economical under our conditions.
- \* A more widespread appreciation of the cost of wear in our operations.
- \* Bearings of acceptable life-span.

Figure 14 shows a possible next-generation pulley which demonstrates most of these features.

- \* Shaft connection is by key-less close control shrink fit, allowing a 15% smaller shaft without risk of fretting corrosion in the assembly. Shaft material is still mild steel and the whole is well suited to small diameter large shaft pulleys since it is compact.
- \* Drum end is turned from plate with minimum wastage and giving excellent stress distribution.
- \* Pulley weld is made in a position of low stress, where automatic techniques and non-destructive testing can be applied reliably.
- \* Inner diaphragms are used to control plastic deformation caused by low stress high cycle fatigue.

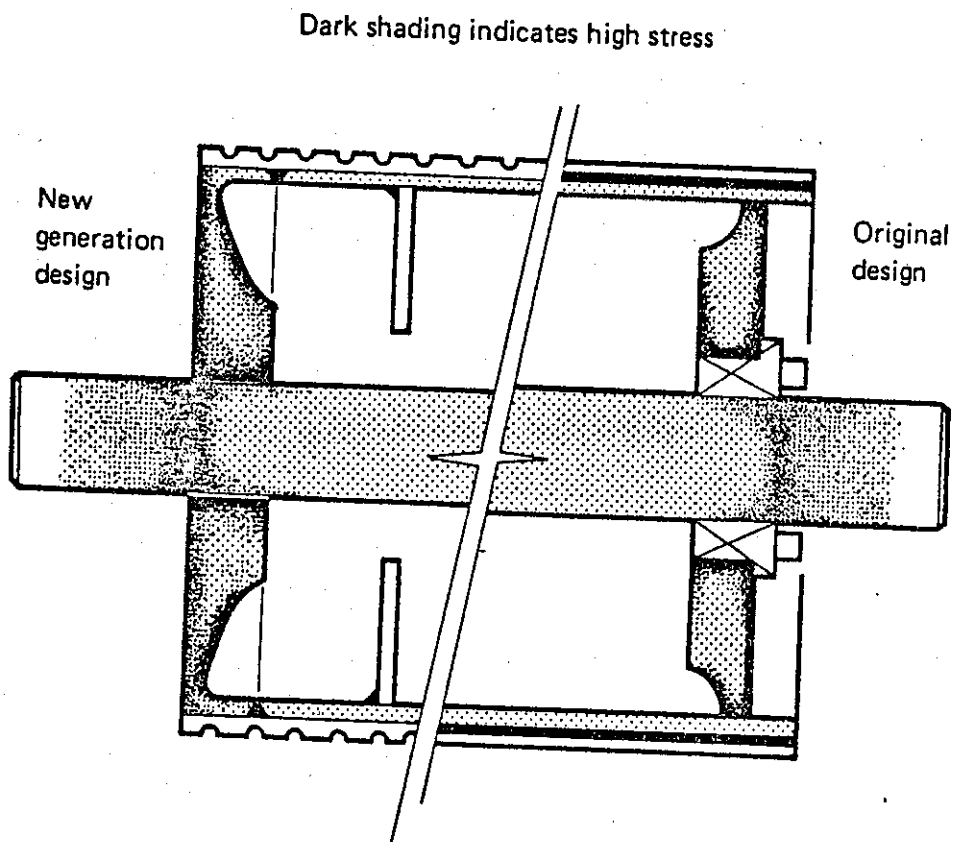


Fig. 14 NEW GENERATION BELT CONVEYOR PULLEY  
- COMPARED WITH CURRENT EQUIVALENT

H7 -  $\frac{3}{4}$  to 1 thou/inch

- \* All pulleys would be covered in a low cost high wear resistant two-colour lagging. Excessive wear is indicated by change of colour in time for re-lagging.

Such a pulley is between twenty and thirty percent cheaper than the current equivalent and would last indefinitely or until the bearings saw through the shaft.

In closing may we in the pulley industry make a plea in the interest of simplified manufacture. In one nine month period recently, one pulley company manufactured 47 different pulley diameters, 41 different face widths and 25 different shaft diameters. More than 10 different constructions were employed, making a total of more than 480,000 different potential pulley designs, not counting shaft variations.

When drawing up your next company standard may we ask that you talk to a manufacturer, or better still, use ISO 1536 (19) as a guide so that we can help you by making more economical pulleys, in a rational range of sizes.

#### ACKNOWLEDGEMENT

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