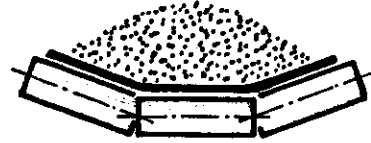# BELTCON 1

BELT CONVEYORS - DESIGN, OPERATION AND OPTIMIZATION

PAPER A5

THE DESIGN OF CONVEYOR PULLEYS

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CONVEYOR PULLEY DESIGN

INTRODUCTION

Up until a few years ago , when conveyors in South Africa were rather
small compared with their counterpart in other countries overseas , the
need for engineered systems was limited.

With the advent of the oil crisis this situation changed over night for
two reasons , i.e. diesel fuel vehicles became almost too expensive to
use and secondly the demand for coal as a fuel became increasingly
important. In order to mine the coal economically , it became necessary
to increase the tonnage mined , as well as to improve the method of
transporting the coal that was mined.

The obvious answer to the problem was to use conveyor belts in conjunction
with unit trains and sophisticated loading and unloading stations.

Coal mining was not the only type of mine effected by the rise in cost
of oil based fuels , but also the base minerals , such as iron ore , copper
and uranium.

It was the base mineral mines , which at the time of the oil crisis , that
were using the open cast method with electro haulpacks for transporting the
ore and overburden more so than the Collieries.

Unfortunately or fortunately for South Africa , the change from small or
medium size conveyors to large installations happened virtually overnight
and the technology to undergo the change was not readily available , hence
people in the conveyor field used their limited existing knowledge to try
and make the larger and more sophisticated conveyors.

This knowledge was however insufficient and hence resulted in many failures.
The most noticable failures being that of terminal Pulleys and belting.
The idler manfucturers seemed however , to have escaped the lime light ,
either because they were more prepared for the change and sufficient
technology was available from their overseas parent company , or the time
and the cost to replace idlers was not sufficient to warrant an investigation.

With the terminal Pulleys , a different situation arose. Just about every
mining house and other conveyor orientated companies saw the need to
improve the standard and the design of Pulleys. However , unlike Germany
and other countries , each individual concern tried to develope
standard and design on their own , without consulting each other.

The end result of these standards can only be classed as disasterous
and unnecessary waste of money and time , as none of the standards are
competable with each other , except for one thing , and that is the
rigidity of the expected standard of workmanship and the amount of
paper required to prove the soundness of the equipment.

By the amount of paper required , I mean , this so called quality
assurance procedures , whereby it has now become necessary to prove
that even the smallest Pulley , which is just an oversize idler , passes full
pressure vessel tests.

It is not that I am opposed to quality assurance , but just that I feel
that the relationship between quality control and purpose of use should
be equated so as to build a unit fit for service at the right price.

This bring me to the definition of an engineer , which goes as follows:-
" An Engineer is someone who can build for a penny , what any old fool
   can build for a pound ".

When our company went into the manufacture and design of Pulleys like
everyone else, we went our own way to establish a standard , but with a
difference. We first tried to find out how people overseas were tackling
the problem. During the course of this investigation we found out that
Germany , Australia and some parts of the United States had all had their
problems when they changed over from small to big conveyor belts. Germany
in particular , where most of the large material handling plants are designed,
seemed to have spent many years in developing their technology in conveying
with the emphasis on conveyor Pulleys. At Hannover University two doctorates
were awarded for research and development in conveyor Pulleys.

By dealing with the right people , we were able to get hold of copies of
both these doctorates , as well as many other papers written about conveyor
Pulleys. By tabulating this information , we arrrived at the bases of
our design.

DISC DESIGN
―――――――

The most complicated part of a Pulley and the area most prone to failure
is the end disc.

The origin of disc design :
―――――――――――

Formula for stress and strain of disc subject to loads acting at the centre ,
are to be found in many technical hand books .

At present we are basing our calculation on formula for disc stress , which are obtainable from book data out of " Roarks " handbook and a thesis written by " W. Schmoltz ".

The disc can either be regarded as subject to bending moment , while the outer periphery is imagined to be fixed rigidly (see condition A) , or on the other hand , the disc can be regarded as being suspended in such a way , where it cannot transmit any bending moment (as in condition B) .
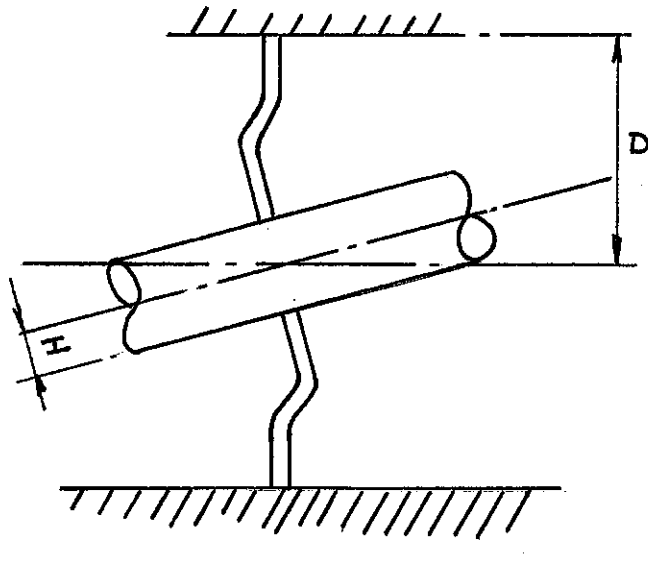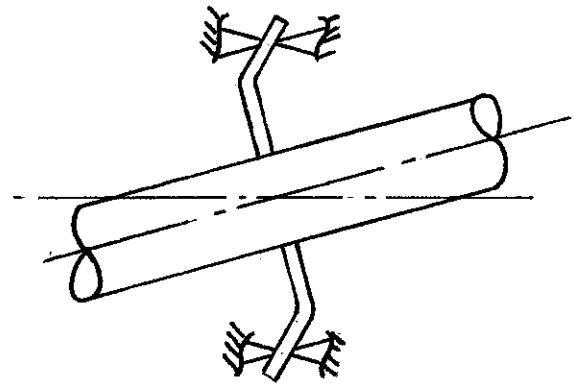
Condition A   Condition B

In both cases the inner diameter is taken to be rigidly connected to the shaft. Both conditions of fixing have to be considered as extreme cases for a Pulley disc.

The actual loading will be somewhere in between condition "A" and "B" . Consequently it should be necessary to calculate the stress in the inner diameter of the disc for condition "A" as well as condition "B" .

In Pulley design we will be simulating condition "A" to a greater degree than condition "B" and the stresses as per condition "A" result in higher stress than for condition "B" .

For the abovementioned two reasons only condition "A" will be considered in this paper.

Considering now the bending moment in the shaft , disregarding the
restraining action of the disc , the shaft would develope the following
bending moment diagram :
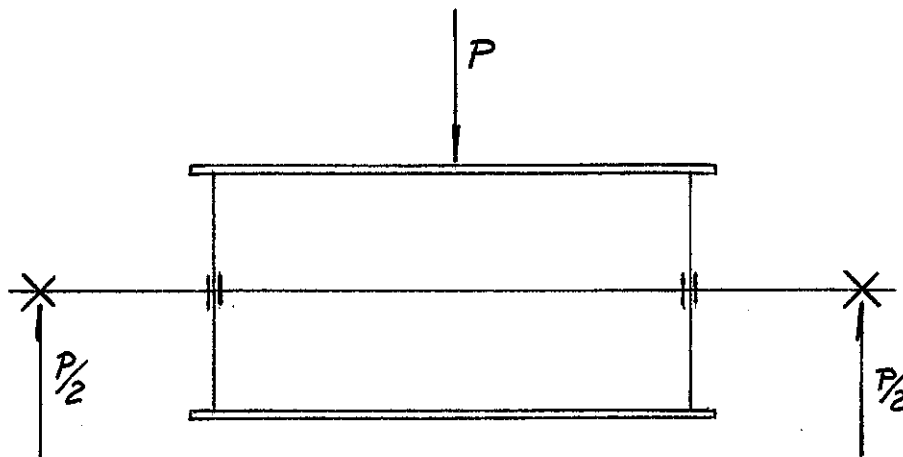
In corporating the resistance of the disc and shell to restrain the deflection
of the shaft , the actual bending moment looks therefore like shown below :
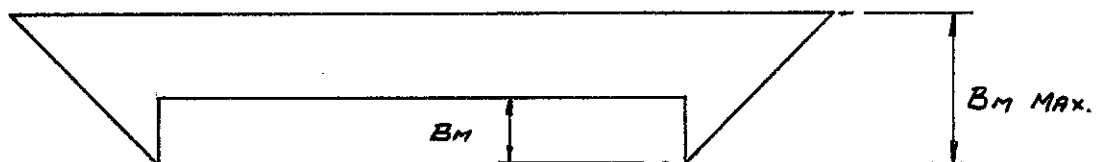
$$Bm \ disc = \frac{P \ (B-L) \ L}{8I \ (\frac{1}{2}L + \frac{1}{\alpha t^3})}$$

(Bm disc) is the portion of shaft bending moment which sets up the stress in
the disc at the connection with the hub , which so often has resulted in hub
to disc failure if a hub is used.

" Simple solution : Don't use hubs " .

The question then arises , if hubs are not used , how do you fix the disc to shaft . (This aspect is covered in the next section headed : LOCKING ELEMENTS).

FACTORS WHICH AFFECT BENDING MOMENT AND STRESS IN DISC

From the above it will be obvious that by reducing the deflection of the shaft , one immediately reduces the amount of disc deflection as well as counteracting disc restraint:

Thus   1.   Bm and deflection of the disc can be readily reduced
             by increasing the shaft diameter (expensive method) .

         2.   Bm disc can be readily reduced by decreasing the
             thickness of the disc .

A steel Pulley with a thin disc allows the shaft to deflect as it naturally would by virtue of the two loads applied to it through the disc. A Pulley made with thick discs that do not bend appreciably as a result of the bending moment applied to them causes another type of shaft deflection , namely , from the disc outwards and for all practical purposes , the shaft could just as well be cut off between the disc.

Between these two extremes lies a range of thickness that causes high disc stresses , unnecessary high bending moment on the inner diameter of the disc and eventually causes Pulley breakage.

Using basic data , a formula has been developed to calculate the disc stress:

$$SB = \frac{\beta P\ (B-L)\ L}{4Dt^2 I\ \frac{(L\ +\ 1)}{2I\ \propto t^3}} \qquad k8/cm^2$$

P = Total resultant belt pull on the Pulley = k8

B = Bearing centre distance   = cm

L = Centre to centre distance between disc = cm

D = Disc outer diameter = cm

t = Disc thickness = cm

I = Moment of inertia of shaft = cm^4

$\propto$ = Disc constant

β = Disc constants

The factors $\propto$ and β are selected after calculating the ratio :

Inside diameter of disc
Outside diameter of disc

For derivation of formula see Appendix "A" .

The actual disc stress versus thickness for a typical small diameter
Pulley is illustrated below :

Chosen Pulley 450 Dia X 1100 Face X 90 Shaft

Pulley loading 27.25 kN resultant load

Bearing centres = 1520

Disc centres = 1020mm
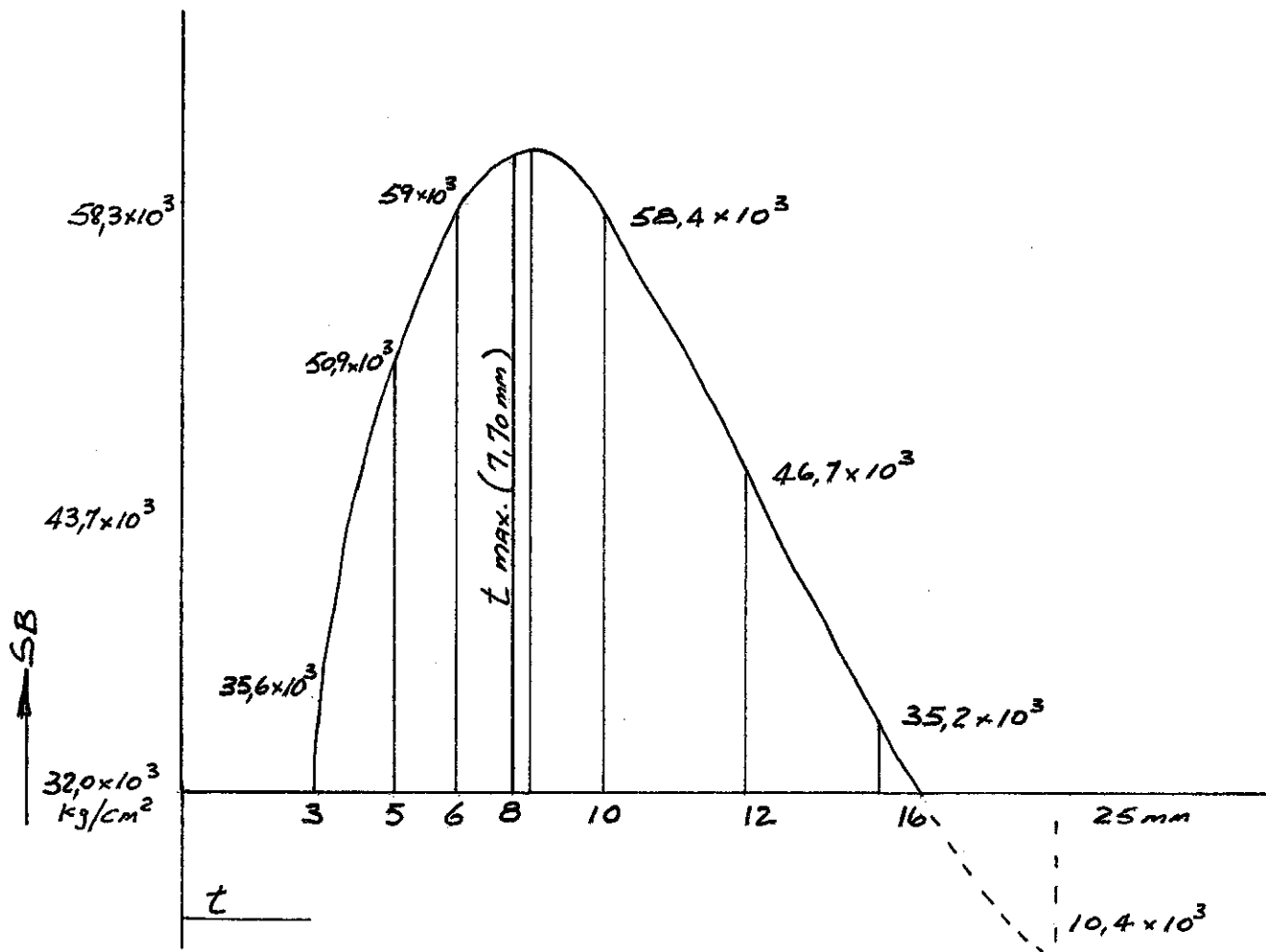
Disc outer diameter = 430   (10mm rim)

Disc inner diameter = 185

In this example :

α = 6.6

β = 1.65

After calculating the disc stresses for various disc thickness it will be
seen that the stresses increase from 3mm to 8mm and beyond this the stresses
reduced as the disc thickness increased. Please refer to the graph below :

Thus for an allowable working fatigue stress there are two disc thicknesses that can be utilised. One a thin disc and the other a thick disc.

Between these two limits a high stress is inevitable. This represents one of the few cases in design , where increasing material thickness results in higher , rather than lower stress in the part.

The disadvantage of employing thick disc to reduce stress is that the bending moment on the connection , from the shaft to the disc is greatly increased , and if this connection is not securely made the bore of the disc wears out and the Pulley moves or what is known as "walks" on the shaft.

The second item effecting end disc design is the positioning of the disc relative to the bearing centres. It is quite obvious that there are again two positions to limit the stresses if close together in the middle of the Pulley or alternatively as far out as possible on the face , will result in the least stress , while in between will result in high stress regions.

We have chosen the outer position for an obvious reason , and that is that the conveyor belt does not always strain perfectly , and if the disc was centrally situated and the belt runs off to one side , the disc would be subjected to additional bending stresses , due to the excentric load .
When using stress formula , it has been found that the large diameter Pulleys with the low set loading are seldom problemsome as they can be designed with the thin disc , while relatively small Pulleys with big shafts become difficult , because the disc becomes impractically thin or the direct stresses are too great to use thin disc . Thus forcing the manfucturer to use the thick disc design.

Typical examples of this type of Pulley is those associated with steel cord belts , where the required diameter of the Pulley is far smaller than the equivalent required for a fabric belt.

When using the thick disc design , there are two important areas to consider :
namely , the fixing of the disc to the shaft (Discussed under the section of Locking Elements) , and the disc to rim fixing .

On extremely high loaded Pulleys , it is often an advantage not to weld in the inter-face between rim and disc,but to move the weld inboard of the disc and make a but weld on the rim. This type of disc is known as a T bottom type disc.
See sketch below :
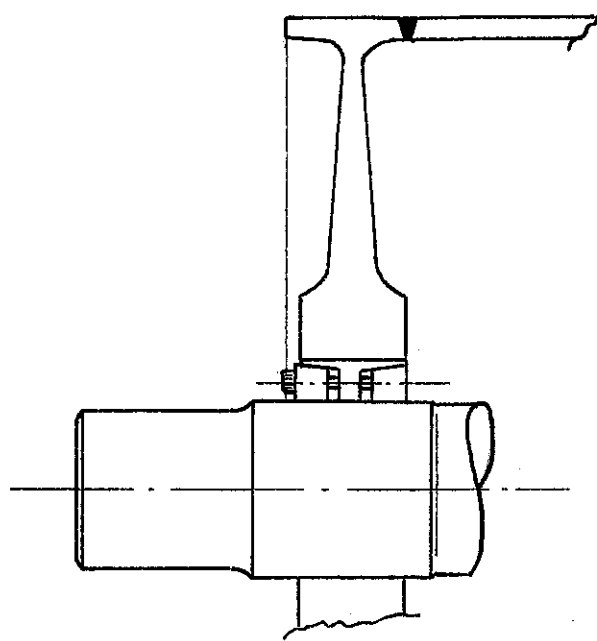
The advantage of this is that it has substantial radius both inside and
outside the high stress zone and to limit stress concentration it is
always better to use radii rather than sharp corners. The other advantage
is that the weld as positioned is firstly out of the high stress zone and
secondly is in a position where it is far easier to make a perfect weld and
the thickness of material to be welded is thinner. The question arises
"Why are not all Pulleys made like this?" The answer is simply cost.
Remember our definition of an Engineer.
When looking at the sketch of the T bottom , you will notice that the disc
is tapered and not parallel. There are two reasons for this ;

1. Although we are using the thick disc design there is still merit in
    giving the Pulley certain flexibility in order to lower the stress at
    the shaft to disc connection.

2. By profiling the end disc , it ensures that the stresses throughout
    the disc are constant and hence limit stress concentration areas.

As mentioned before not all Pulleys are T bottom , for the simple fact
that they cost far more than other designs and hence we come to the
second type of end disc , namely a Turbine End.
In our terminology the T bottom is our heavy duty range and the Turbine
and semi-Turbine our medium duty range.

The Turbine Pulley is similar in characteristic to our T bottom design
but has two distinct differences:
a) We use a different type of Locking Element
b) The welding is performed at the inter- face of disc to rim

Although we emphasized the advantage of removing the weld from this position
when we discussed the T bottom type Pulley , we must clearly state that this
type of Pulley we are now discussing , is one that does not have to with-
stand the same tentions as the T bottom , and hence the stress level is far
lower. When performing this weld between rim and disc it is important to
ensure that this weld is of a sound nature. With full penetration and no lack
of side-wall fusion. To insure full penetration and to limit stress
concentration on the inner face , it is good practise to perform an inner
fillet weld if possible.

See sketch :

INNER FILLET
WELD.
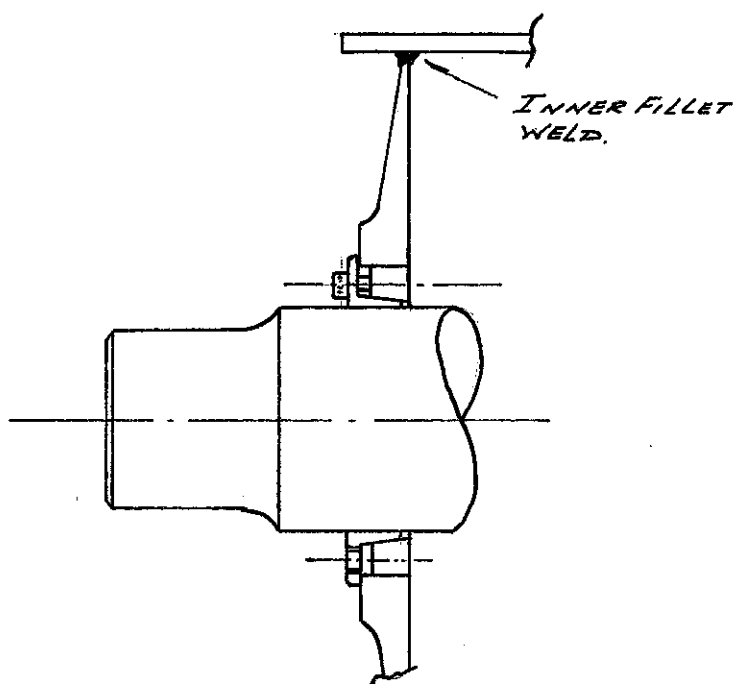
The last type of disc we are going to consider is a semi-turbine end or flat bottom design.

This Pulley for the majority of conveyors, is the ideal type of Pulley. It is relatively in-expensive due to the easy method of machining the disc. It has most of the advantages of the turbine end but with the exception of the two zones where stress concentration occurs, but taking suitable stress concentration factors into consideration these stresses are well below normal exceptable levels. Again using suitable welding procedures to perform the disc to rim weld to ensure a sound weld, will result in a Pulley which is fit for the duty.

See sketch:
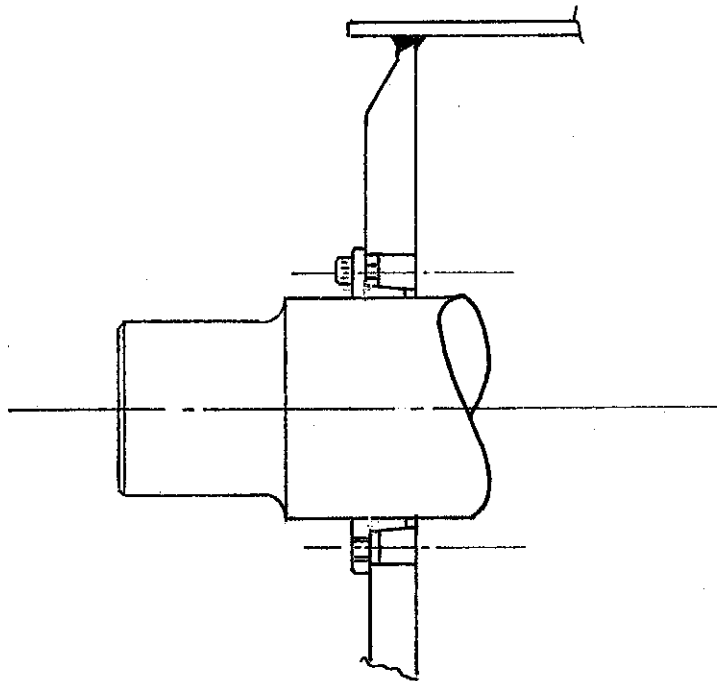
Many people are of the opinion that the reason for using a locking element for fixing the disc to the shaft is so that the shaft can be removed if necessary. This in itself is a good reason for in – stalling a locking element, but by no means the only reason for fitting a locking element.

Below are three sketches of suitable locking elements for Pulley construction and we will discuss the advantages of these elements collectively and thereafter individually.

TYPE A   TYPE B   TYPE C
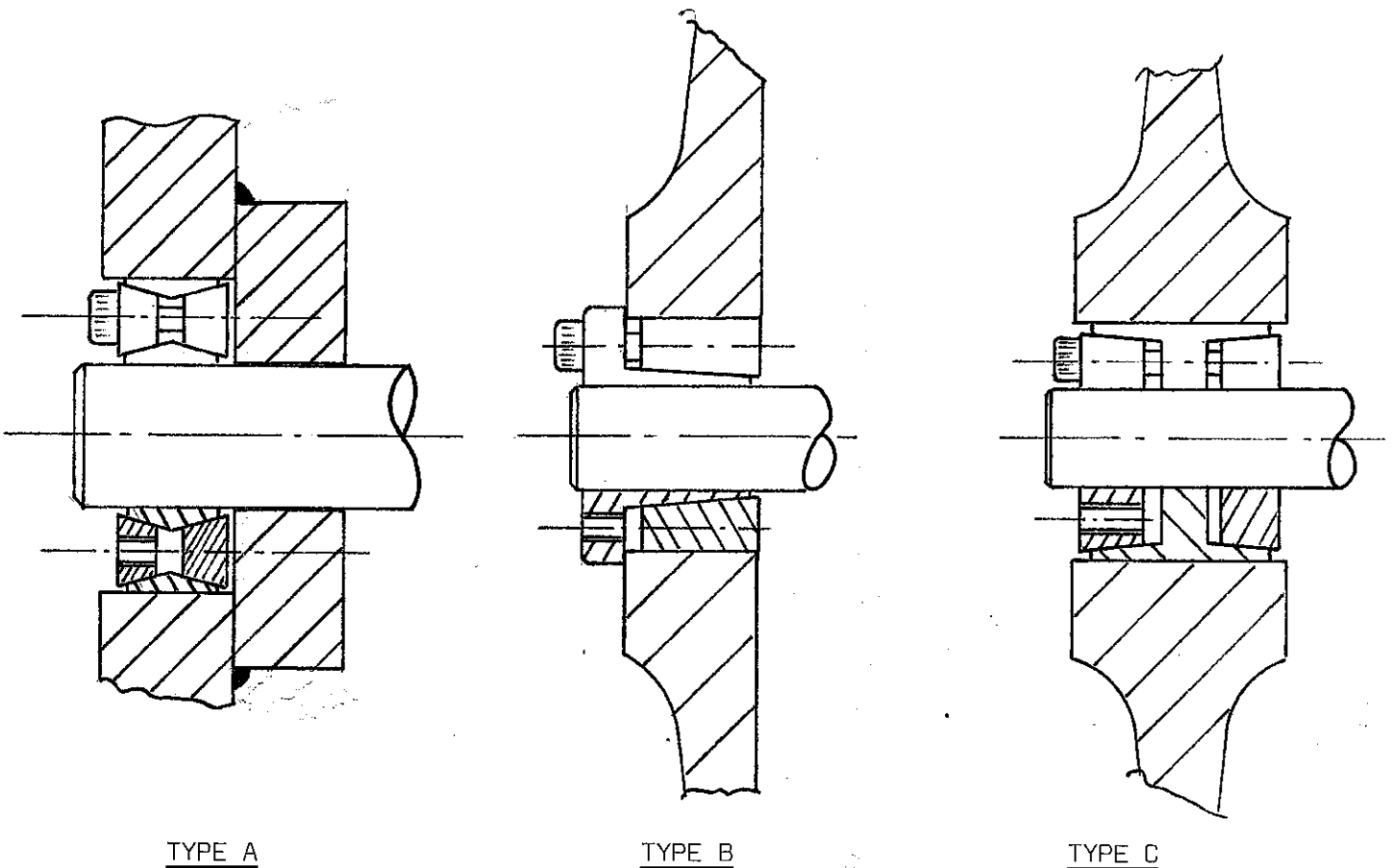
The three elements share the following advantage over all other known
hub to shaft connections:

a.) They by virtue of the torque applied to their bolts, have a
controlled pressure on both, hub and shaft. By simple mathe-
matics, this pressure can be calculated and hence eliminating
the hit and miss factor which is inherent in a "shrink fit"
thus making it possible to calculate, by using thick wall
cylinder formula, the size of the hub required to with-stand
the pressure fairly accurately and thus eliminating unnecessary
hub material.

b.) Their torque, which they can transmit, is also a factor controlled
by the torque applied to their bolts, thus again making it unnece-
ssary to fit additional safe guards, such as keys and keyways, to
ensure that they do not slip in the direction of the applied torque.
By the elimination of these keys and keyways the stress concentration
associated with the above is naturally eliminated.

c.) All three locking elements can withstand great axial thrust thus
virtually eliminating the possibility of walking on the shaft, and
in the unlike event of this happening, it is relatively simple to
rectify, as it is usually only necessary to place the drum back to
its original position and re-torque the bolts in the locking element.
If a similar situation occurs with a shrink fit, the bore has inevit-
ably been worn away and the entire Pulley becomes redundant.

LOCKING ELEMENT TYPE A

Locking element type A has been on the South African market for approxi-
mately ten years and is probably the most common locking element used in
Conveyor Pulleys. In the unloaded condition it has undoubtably the best
stress pattern of the three locking elements. However, when load is app-
lied to the drum and the shaft deflects, the stress pattern changes.
See comparison sketch between locking element "A" and "B".
In locking element "A" because of the angle of the tapered segments, it
is neither self centering nor self locking, hence at all times the
securing bolts are   in tension, thus it becomes necessary to use a
centralising ring as shown on the sketch, as well as having to limit the
deflection of the shaft, to ensure that the bolts do not exceed their
elastic limit.
The acceptable level of deflection for this type of locking element is
in order of 1/2500 bearing centres or if prefered, in radial units 0.05
degrees or 0.0008 radians, although this deflection restriction is not
totally unfavourable, it often forces the designer to use a bigger shaft
than he normally would, to satisfy acceptable stress levels of torque to
be transmitted.

The most unfavourable characteristic of this type of locking element
is that it is virtually impossible to control the mating of the sur-
faces of the segments. This can result in high point loads initially
and also make it necessary to re-torque the bolts after initial
running in period, after the segments have settled.

LOCKING ELEMENT TYPE B

Locking element type B is the type we are currently using in our medium
duty Pulleys, i.e. the turbine or flat bottom design. The reason for
using this type of locking element and not type A, is that it has the
following advanatges:

a.) Because of the angle of the tapers on the segments, these
    elements are both, self-centering and self-locking, there-
    fore do not require a centralising step behind the disc.
    Because of the elimination of this step, it is always
    possible to withdraw the shaft from the drum, because the
    inevitable build-up of rust between centrallising step and
    shaft does not occur.

b.) By having self-locking tapers, the bolts are not in tension
    to the same extent as type A, hence greater shaft deflections
    are possible. The permissible deflection being in the order
    of 1/1800 to 1/2000 bearing centres or expressed radially $\pm$
    0.06 degrees or $\pm$ 0.001 radians.

c.) Unlike locking element type A it is not so critical that the
    locking element bolts are tightened in sequence as due to the
    tapers it is impossible to tighten this element unsemetrically.

d.) The major disadvantage of this type of locking element, is that
    it cannot transmit the same torque as type A. However, on con-
    ventional Pulleys this seldom causes a problem as the torque to
    be transmitted by the shaft is usually well within the capabi-
    lities of the chosen element.

LOCKING ELEMENT TYPE C

Locking element type C was specially designed for conveyor Pulleys.
It has all the advantages of type A and B , with far lower surface
pressures than either , while at the same time being able to transmit
between 2 and 3 times the Torque of either A or B.

This element is used in all our T bottom Pulleys , with the shaft sizes
up to 400mm in diameter. The Rhein Brown Coal A.G. in Germany have
been using them with shaft sizes up to 600mm successfully.

## PRECAUTIONS WHEN USING LOCKING ELEMENTS IN CONVEYOR PULLEYS

Many people think that by specifying a Pulley fitted with locking
elements , they will automatically get a better Pulley than a shrink
fit type Pulley. This is often not the case , as the type of hub that
is used is as important as the type of locking element chosen. Often ,
if the hub design does not suit the locking element , the hub cracks
under pressure from the locking element. or alternatively the hub
diameter is increased , so as to accomodate the locking element , and
this hub is too large , and when using a thin disc often limits the flexing
of the disc , hence cracking the weld around the hub.

TOP HAT FORGED HUBS

A design has been introduced into South Africa which is quite unique.
I have not seen anything quite like it any where in the world , and
it is because of its uniqueness I feel some kind of mention should be
made to try and analise it.
Firstly its origin during the change from light duty conveyors to
heavy duty conveyors. The most noticable area of failure was the two
fillet welds at the hub sketch A.
It then became apparent that if the hub was extended so that a double
butt weld could be used , this would obviously be a far better type of
hub sketch B than its predecessor A. Design B was used for a few years
and was later replaced by design C , which instead of machined out of
solid material , could easily be forged by using a single sided dye.
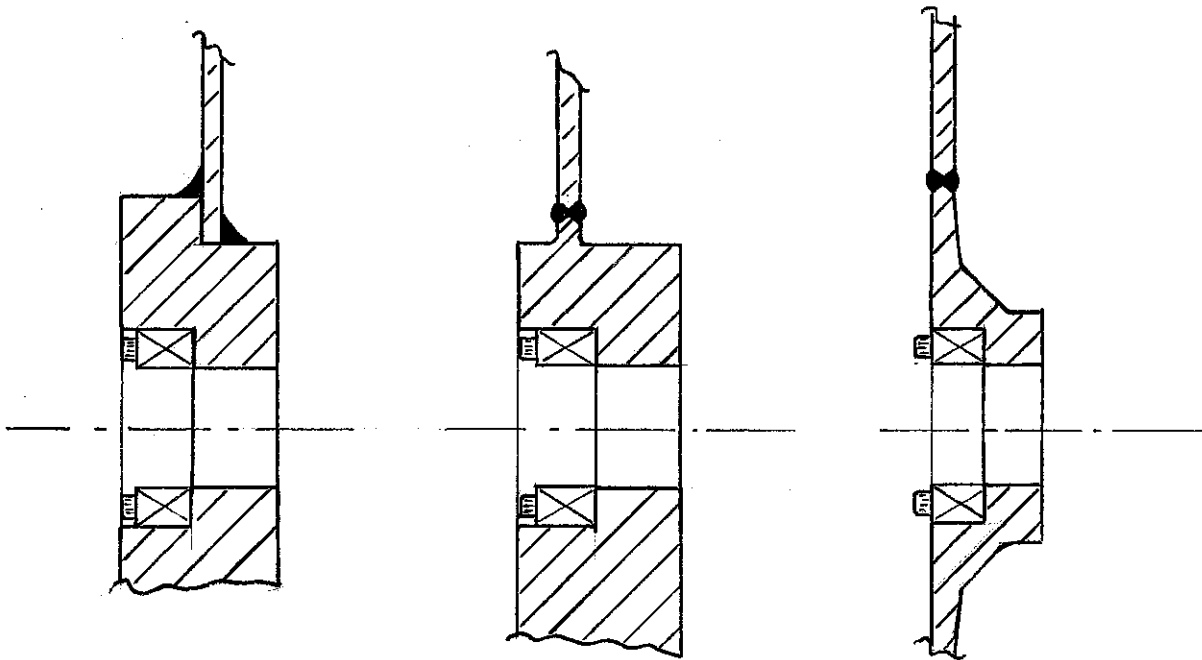
This hub had one great disadvantage and that is that in order to utilate
the potential of its simplicity , it has to suit various shaft diameters ,
drum diameters and disc thickness. This not being a suffiently complicated
excercise , this hub was further modified to be able to accept various
shaft to hub fixing , such as , shrink fits , taper locks and locking
element type A . The result of the excercise resulted in a large chunck
of useless material which only added weight and cost to the Pulley, no
wonder it is unique to South Africa.
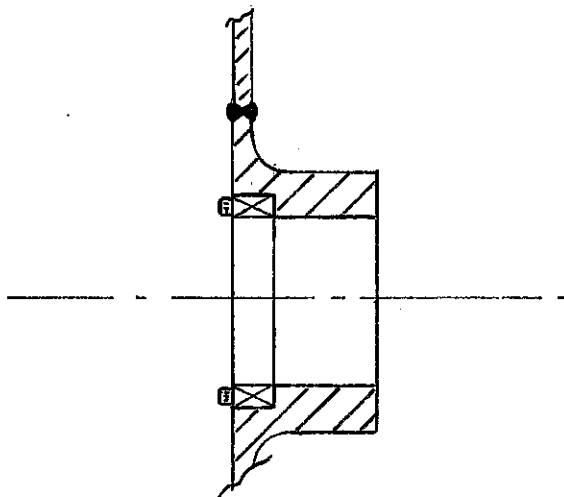
SHAFT DESIGN

During the course of the paper many times reference has been made about
deflection and its limits. A simple and common formula is used to
calculate this deflection :

$$\frac{PL2a}{8EI} \left(1 - \frac{4a^2}{3L^2} \right)$$

Where : L = Bearing centres

a = Lever arm

E = Modulus of elasticity

After calculating the deflection , the next step is to calculate the stress
level , either for straight bending in the case of non drive Pulley ,
or the combination of bending and torsion for drive Pulley.

The last step is to check the locking elements Torque charateristic
against the applied Torque , in the case of a drive Pulley. Remembering
that during starting and breaking , this Torque increases substantially
and appropriate precautions must be taken into consideration to prevent
the locking element slipping.

RIM DESIGN

Due to the complex nature of the calculations involved in rim design , we
will not be discussing it in detail in this paper.
For design purposes , the drum casing is regarded as a cylindrical shell
with a partially loaded surface. Particular attention must be paid to
the boundry conditions and the load distribution , if this approach
is used for conveyor Pulleys. The rim is then regarded as a simply supported
tube with no moments or axial forces. The belt forces at the edge are
transmitted in the direction of the plane of the drum only on the contact
surface between rim and conveyor belt. A pressure and shear stress
distribution , similar to operating conditions is achieved by super-imposed
sine functions. The total load of the element is obtained by summation
of the individual sinusoidal loads. This calculation is far too time
consuming without the use of a computer to be delt with in this paper
as they are usally unknown under eytewein boundry conditions. When the fatigue
strength of a rim is calculated , it is necessary to remember that three
load changes occur in each casing segment during a single drum rotation.
Rim design limiting factor is usually based on fatigue strength.

CENTRE STIFFENERS

Centre stiffeners in large conveyor Pulleys are again unique to
South Africa , and the only artical I have seen relative to stiffeners
was associated with pressure vessels titled " Callapse of Stiffened
Cylinders Under External Pressure " by S.B. Kendrick.
The only other association I have had with them is having seen samples
of failure which have occured in the vicinity of these centre stiffeners
on large Pulleys.

THEORY ANALYSIS - PULLEY DETAIL (DISC)
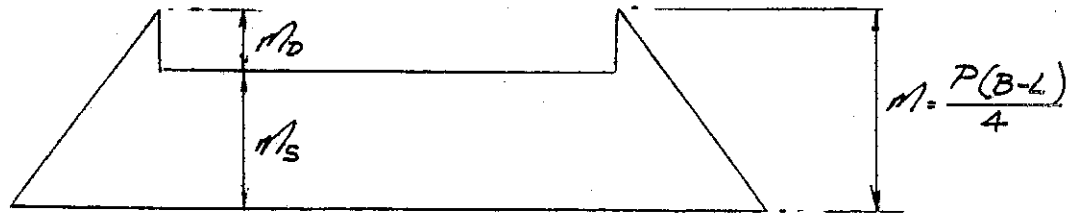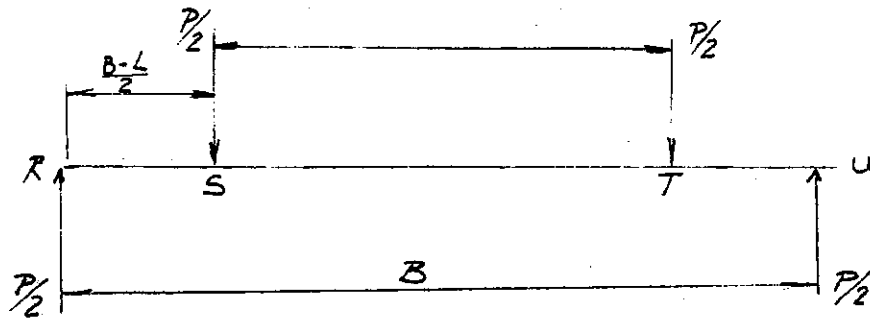
$$\frac{M}{I} = \frac{E}{R} \qquad BUT \quad R = \frac{\left\{1 + \left(\frac{dy}{dx}\right)^2\right\}^{3/2}}{\frac{d^2y}{dx^2}}$$

$$= \frac{1}{\frac{d^2y}{dx^2}} \quad WHEN \quad \frac{dy}{dx} \ IS \ SMALL.$$

$$\therefore \frac{d^2y}{dx^2} = \frac{M}{EI} \qquad \left(\begin{array}{l} BUT \ BETWEEN \ POINTS \ S \ AND \ T \ ON \ THE \ SHAFT \\ THE \ BENDING \ MOMENT \ M_S \ IS \ CONSTANT. \end{array}\right)$$

$$\therefore \frac{dy}{dx} = \frac{M_S}{EI} \int dx = \frac{M_S}{EI} x + C$$

BUT WHEN $x = \frac{L}{2} \rightarrow \frac{dy}{dx} = 0$

$$\therefore \frac{M_S}{EI} \times \frac{L}{2} + C = 0 \ \text{————} \ ①$$

THEN $C = -\frac{M_S L}{2EI}$ (SUBSTITUDE BACK INTO ①).

THEN $\frac{dy}{dx} = \frac{M_S}{EI} x + \frac{M_S L}{2EI}$ (AND $x = 0$)

$$\therefore \frac{dy}{dx} = -\frac{M_S L}{2EI} = \theta_S$$

BUT $M_D + M_S = M$ AND ALSO $\theta_S = \theta_D$

ie $M_D + M_S = \dfrac{P(B-L)}{4}$

$M_D \left(1 - \dfrac{2I}{\alpha t^3 L} \right) = \dfrac{P(B-L)}{4}$

$M_D = \dfrac{P(B-L)}{4\left(1 + \dfrac{2I}{\alpha t^3 L}\right)}$

$\dfrac{M_S L}{2EI} = \dfrac{M_D}{\alpha E t^3}$

$\therefore M_S = \dfrac{2I M_D}{\alpha t^3 L}$

BUT $S_D = \dfrac{2\beta M_D}{D t^2}$

$= \dfrac{2\beta P(B-L)}{4 D t^2 \left(1 + \dfrac{2I}{\alpha t^3 L}\right)}$

$\therefore \quad = \dfrac{2\beta P(B-L)}{4 D t^2 \, \dfrac{2I}{L}\left(\dfrac{L}{2L} + \dfrac{1}{\alpha t^3}\right)}$

$\therefore S_D = \dfrac{\beta P(B-L)\, L}{4 D t^2 I\left(\dfrac{L}{2L} + \dfrac{1}{\alpha t^3}\right)}$