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INCLINE CONVEYORS: THE IMPORTANCE OF T2 TAKE-UP TENSION TO STOP RUNBACK

by

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SYNOPSIS

This paper puts forward 4 different actual incidents of conveyor runback and their case studies. These incidents occurred on four different mines over the last 3 to 4 years.

By presenting this paper, we hope to benefit the users of conveyors in order that they prevent similar incidents from happening by informing them of the possible causes.

The paper will also be beneficial to conveyor designers, manufacturers and equipment suppliers.

CONTENTS

1. SUMMARY

Runback on incline conveyors due to no load sharing or failure of T2 take-up tension can cause immense damage, downtime, fatalities and expensive replacements.

2. INTRODUCTION

Holding back or backstopping of belt conveyors is just as important a function as the starting of the belt conveyor. Using information from four very similar incidents of conveyor runback, actual facts will be revealed to substantiate the cause.

These facts are accurate to the best of our knowledge. The Mines and equipment manufacturers concerned have given us their consent to present said facts.

The justifications of the failures are purely speculative and not necessarily accurate. We require your input during the discussion of any other similar incidents which may have occurred, your input will be greatly appreciated.

3. INCIDENT NO. 1 EASTERN PLATINUM MINE

3.1 Case Study

The accident occurred on the 14th August 1992, on the no. 2 incline belt.(refer to figure no. 1).

The following sequence of events were reported:

- The Mine personnel tried three times to start the conveyor belt, with the overload on the motor tripping out on each attempt. The drive was inspected. The guard around the fluid coupling and drum brake was removed and the brake pads were adjusted away from the brake drum. The personnel were convinced that the brake was not releasing, causing the motor to overload and trip.

- The brake was adjusted on quarter of a turn to ensure that the brake was opening fully, which it was. The motor was again started, and continued to pick up speed until full speed was reached. An excessively high pitched whining noise was heard. Eventually the fluid coupling broke up, spraying oil and parts 20 metres away, pushing the motor backwards 3 metres, (with the power cable still attached). Three mine personnel were injured, one very seriously after he was hit by a flying piece of metal.

3.2 Observation

- An employee had noticed that the belt ran backwards just before the coupling exploded.

- The take-up mass (1200 kg) was flung up to the top of its limit. The tail pulley is fitted on to a moving carriage. The conveyor had only ± 2 tons of ore loaded on the incline belt.

- The coupling had dented the base plate as it dislodged itself. This was on the opposite side of normal rotation - proving that when it dislodged, the coupling was turning in reverse.

- After removal and inspection of the high speed backstop (fitted to the gearbox secondary shaft), there was no evidence of any scarring on the outer shell. However, our facts revealed that because the coupling was driven in reverse by the gearbox, the holdback had indeed failed.

3.3 Possible Causes

We assume that initially the belt had for some reason become jammed. Everytime the motor was started, approximately 150% peak torque was introduced into the belt, (depending on the running time before the overload tripped), thereby tensioning the belt more and more after each start.

On the fifth and final start, the belt dislodged from its jammed position (possibly at the tail take-up pulley or at the carriage). With the incredible amount of the kinetic energy stored in the belt, (the forces in the top section of the conveyor) the belt broke away. The conveyor now travelling in the opposite direction drove the fluid coupling in reverse. The overspeed situation resulted in the coupling been driven to beyond its critical speed limit. Centrifugal forces imposed too great a tensile force on the couplings aluminium casing causing it to disintergrate. Further imbalance caused the coupling to break out of its position between the motor and gearbox shafts. The 160 kW motor was torn off its holding bolts (4 off M24 high tensile bolts) and thrown almost 3 metres away. At 3000 RPM the outer shell's peripheral speed is about 880 km/h. The imbalance forces at this speed are very high.

The exact cause of this accident has not yet conclusively been established.

3.4 Conclusion

Due to the high speed holdback failure, the belt drove the gearbox through the pulley, to very high speeds, turning the fluid coupling in reverse.

Due to the imbalancing, the fluid coupling actually dislodged itself in the reverse direction, colliding with the base plate.

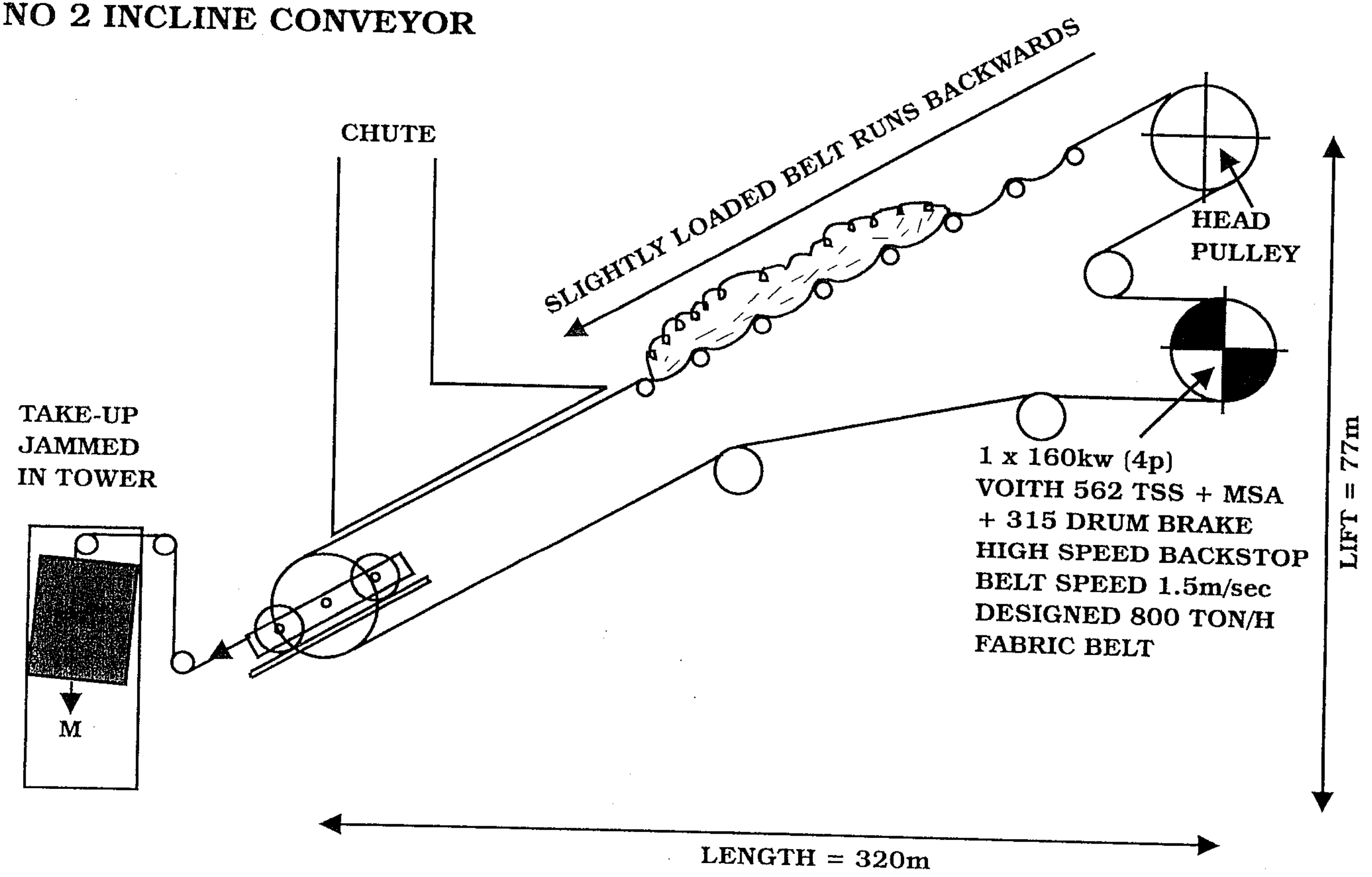
Comprehensive reports were submitted by the CSIR and Voith Turbo Germany, these reports are available for viewing.

Calculations regarding the tensile strength of the aluminium, based on a speed of 1485 RPM, shows the safety factor to be approximately 3,6, i.e. A maximum speed of over 3000 RPM can be achieved before disintegration.

Voith designs and manufactures fluid couplings for 3000 RPM applications, as is the case for dewatering centrifuges. However, although the same aluminium alloy "silumin" is used the material must first be specially quenched and tempered for more strength.

FIGURE 1

EASTERN PLATINUM
NO 2 INCLINE CONVEYOR



4. INCIDENT NO. 2
IMPALA PLATINUM MINE near Rustenburg
SHAFT NO. 1 - CONVEYOR NO. 3

Refer to figure 2 for the drive specification and dimensions.

4.1 Case Study

The accident occurred in November 1993.

The conveyor belt was fully loaded. The belt was started. After a short period, the start was aborted. The belt ran backwards, causing the drive pulley to drive the gearbox in the opposite direction. This in turn drove the aluminium fluid coupling in reverse, reaching approximately 3000 RPM, causing the coupling to break which resulted in an imbalance. The coupling broke out of its position, tearing the motor from its pedestal. The debris was found on the opposite side of normal rotation, proving that the coupling was accelerated in reverse.

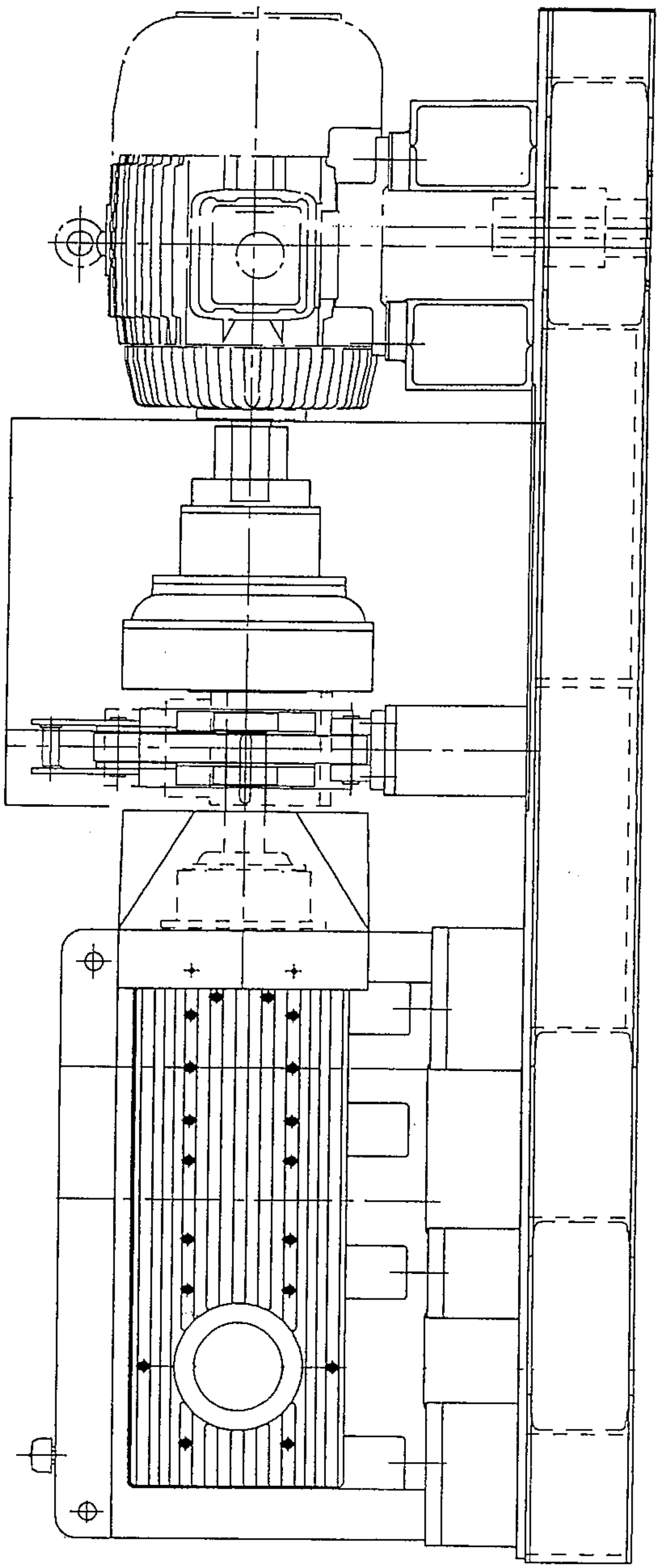
4.2 Observation

At the time, a slow speed holdback was installed at the head pulley. No holdback or brake was fitted to the drive. The Mine insisted that the holdback had malfunctioned. The holdback was stripped under supervision which showed no scarring marks or any evidence that the holdback had failed or run in the opposite direction.

Note that the holdback in question was rated at 142 kNm - The maximum runback torque calculated would be approximately 50 kNm.

The gravity take-up was found to be jammed at the top of the tower. The belt had sagged between the idlers. The sagging had caused the belt to slow down considerably. The reason for the runback, could possibly have been due to the gravity take-up jamming on its railings in the tower after the aborted start.

The material had come to a stop approximately 10 metres from the head pulley.



4.3 Possible Causes

Calculations show that at a runback torque of 50 kNm and a total mass of the system of 87 tons, the accelerating force could speed up the coupling from zero to 3000 RPM in 2.2 seconds. From standstill to a belt speed of 2.78 m/s. Runback distance calculated is only 6M.

Fortunately no injuries occurred in this incident. The unit has since been replaced and is running satisfactory. We have been informed by the mine engineer that at some stage this year, it was reported that after stopping the conveyor, the wet belt moved over the head pulley, even with rubber diamond lagging fitted. Runback was approximately 4 metres, but in this case it was not sufficient enough to accelerate the coupling to its limit. This indicates that it is possible that this scenario exists and occurs on a regular basis to the majority of conveyors. We recommend that conveyor users should somehow monitor this if possible.

4.4 Conclusion

Due to the take-up mass jamming in the tower, there was no T2 take-up tension, to ensure tension over the head pulley where the backstop was fitted. The belt slipped over the pulley, allowing the mass of material to force the belt backwards down the 13 Deg. incline, accelerating the drive pulley and the fluid couplings to the excessive speed of destruction. Neither the holdback nor the fluid coupling were the causes of this accident.

4.5 Recommendation

A great deal of consideration should be put into tower take-up design, especially travel guides or rods, alternatively the distance between the mass and frame, ensuring that the hanging mass can never be restrained from moving up and down freely.

A brake can be fitted either to the high speed or low speed shaft. Remember that a brake cannot take the place of a holdback, as the brakes always need to be adjusted and maintained.
The other reason being that when brakes are fitted to an incline belt which is fully loaded. At start up, the motors are energized and the fail safe brake is released, momentarily allowing the conveyor to run backwards before the fluid coupling torque rises above break away. This can be very dangerous.

Svendborg, a Danish based company make hydraulic fail safe disc brakes, incorporating an adjustable time delay feature. This would be safe for use in this type of application. The cost of this type of brake is approximately half that of a holdback for similar torques. However, as the holdback is a mechanical one way clutch, its availability is a 100%.

5. INCIDENT NO. 3 "PRECIOUS" MINERAL MINE near Pretoria

5.1 Case Study

The accident occurred in June 1993.

The underground 3 "Winze" incline belt no. 33 ran backwards, hence driving the fluid coupling fitted to the secondary drive in the opposite direction to an excessive speed, causing it to break up.

A low speed holdback was fitted to the primary drive. The take-up was done by a winch (using an Eddy current type clutch to control the belt tension during starting or running stopping). (Refer to figure 3).

5.2 Observation

The conveyor was fully loaded. It ran back approximately 10 metres and stopped due to the sag between the idlers.

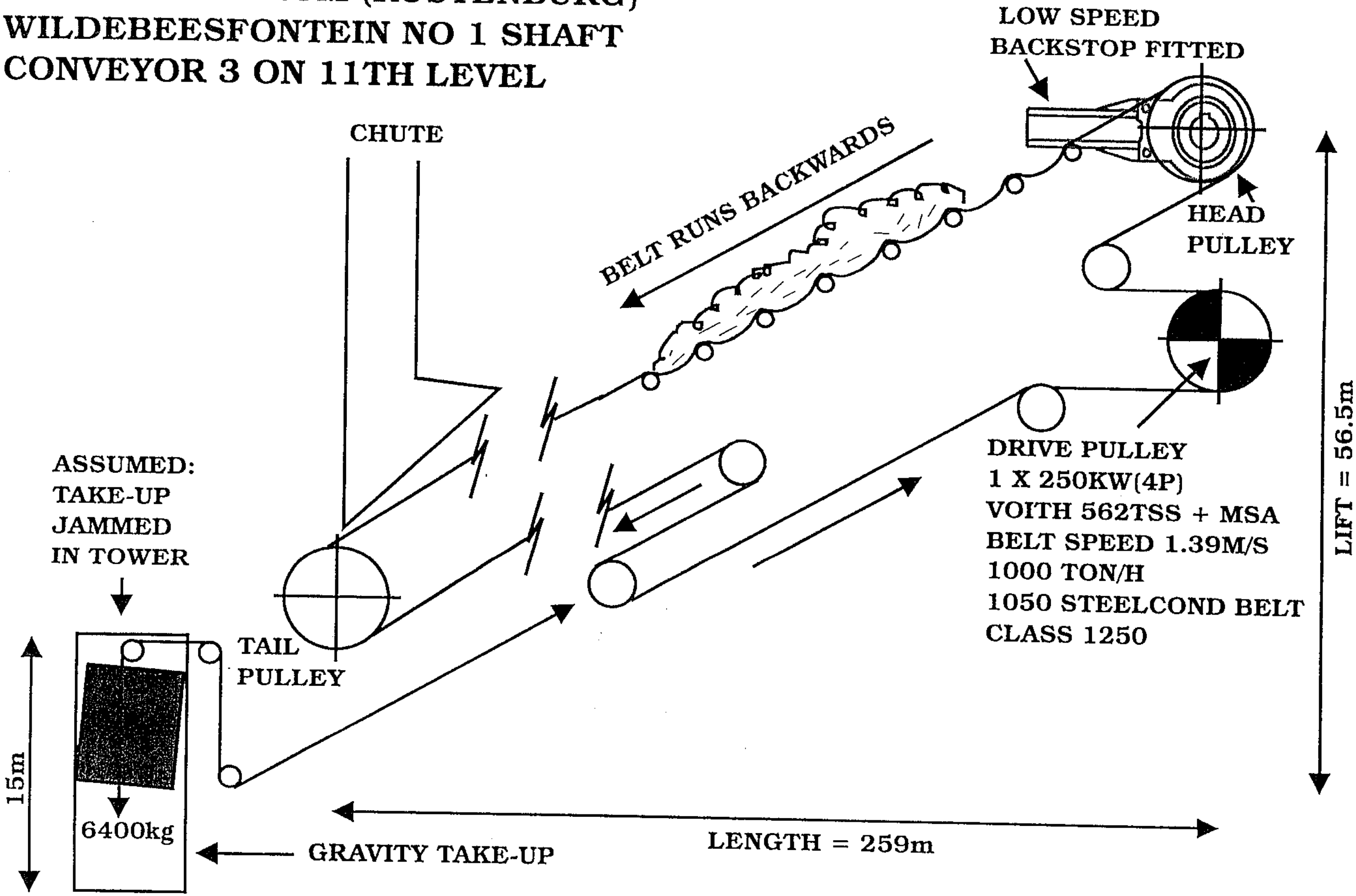
After visiting the site personnally, the take-up and the holdback were checked. A staff member confirmed that they had experienced problems with the Eddy Current winch. The tension monitored by a loadcell to the PLC was checked, and the computer printout showed that the setting was below the normal required tension, possibly the main reason for the runback. Again, reduced take-up tension over the drive pulley.
Again, the holdback was checked and found to be unscarred. A new fluid coupling was installed. Two days later the same incident occurred.

5.3 Conclusion

An additional holdback has been installed on the drive, to ensure maximum belt wrap, and to stop the conveyor from running backwards. (Refer to figure no. 4). Even this is not the solution, as the conveyor can still easily run back if the take-up tension system fails. However, as the pulley locks due to the holdback fitted, there is no chance of the coupling being destroyed under high speed reverse conditions. Fortunately there were no injuries in this incident, although it turned out to be a very expensive learning experience.

FIGURE 2

**IMPALA PLATINUM (RUSTENBURG)
WILDEBEESFONTEIN NO 1 SHAFT
CONVEYOR 3 ON 11TH LEVEL**

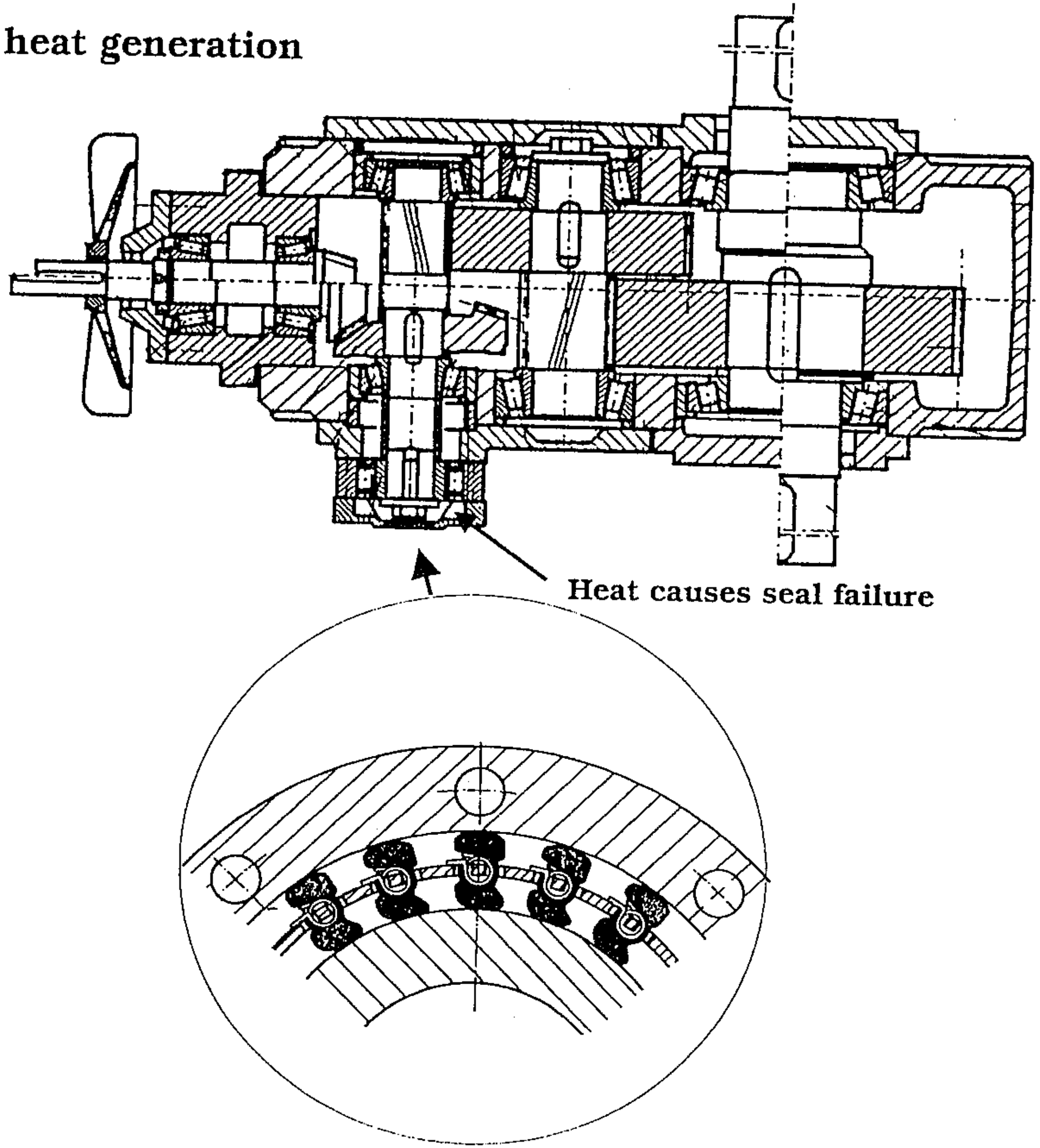


5.4 Recommendation

If one is using a winch, ensure that there is a load cell fitted with some alarm setting for tensions that are above or below the maximum or minimum limit required, which should either shut down the conveyor automatically or it should not allow starting if limits have been exceeded.

The winch fitted with an Eddy Current clutch works well to simulate gravity but at the cost of running a 55 kW motor continuously. Remember gravity is "Free" and a PLC is not required.

High speed Holdback
Fitted to 1st stage shaft
Speed 750RPM
Friction causes heat generation



6. INCIDENT NO. 4 BOSJESSPRUIT COLLIERY : SASOL 2 INCLINE SHAFT BELT NO. 2

6.1 Case Study

The accident occurred on the 1 August 1994. (For conveyor profile refer to figure 5).

The sequence of events occurred as follows:

The incline conveyor "fully" loaded stopped for some unknown reason. The conveyor ran backwards driving the primary pulleys and the Voith 750 TVV fluid coupling in reverse, again to approximately 3000 RPM. The coupling disintergrated and a few seconds later the secondary drive fluid coupling also disintergrated. The conveyor ran back approximately 30 metres and then stopped.

6.2 Observation

The low speed backstop fitted to the primary pulley had failed, causing the full load to be taken by the secondary pulley which in turn also failed, allowing the fully loaded belt to run backwards.

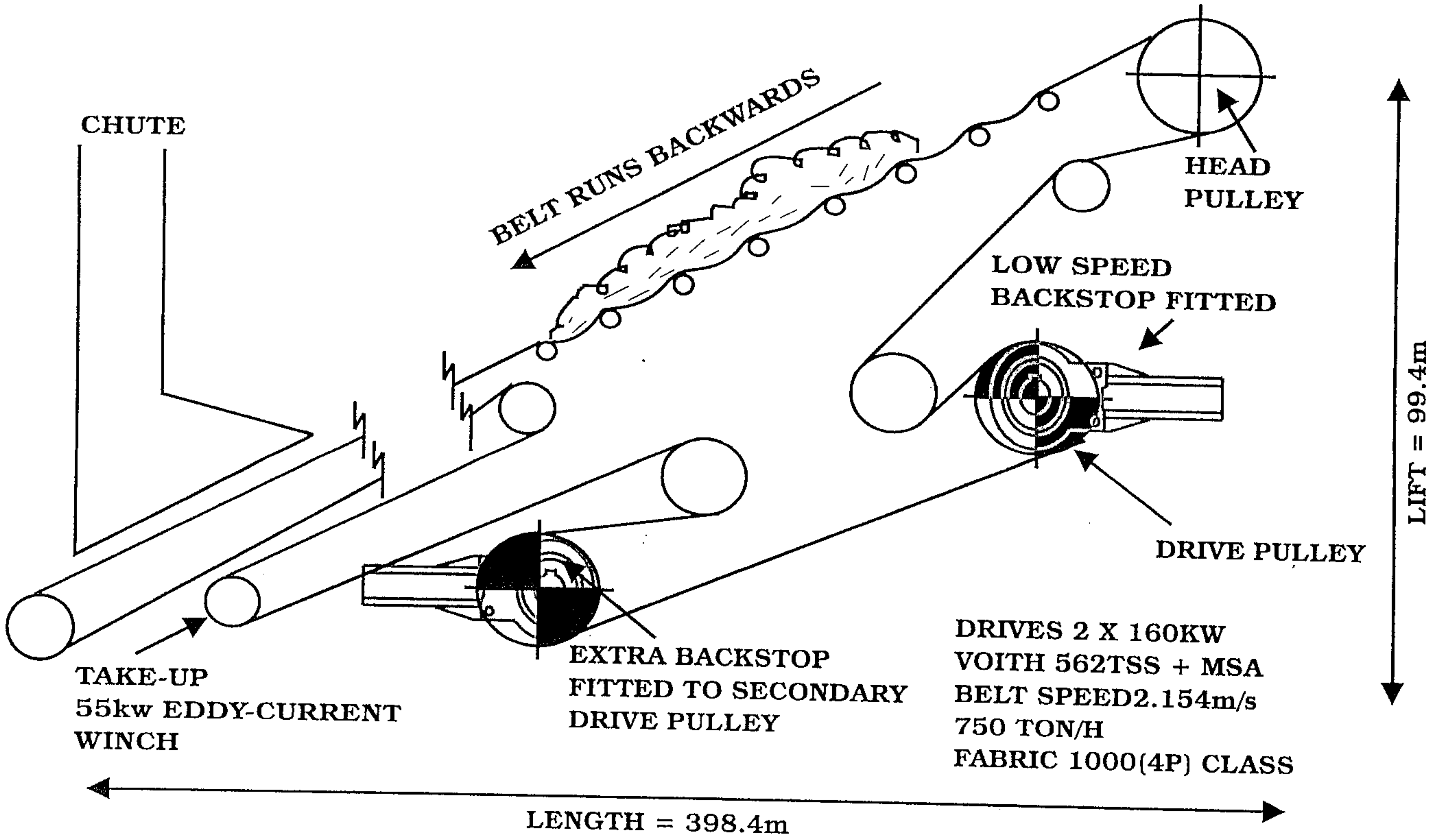
6.3 Possible Causes

a. The conveyor was overloaded with approximately 3200 Tons/hour, (the actual tonnage is still unconfirmed).

b. The primary pulley holdback, obviously taking all the load (the secondary primary holdback no load), eventually became overstressed and failed. With the accelerating force, the runback in turn over-stressed the holdback on the secondary pulley resulting in its failure as well.

FIGURE 4

**PRECIOUS MINERAL MINE (PRETORIA)
WINZE CONVEYOR NO 33
AFTER TWO RUNBACKS**



6.4 Conclusion

The main cause was due to the malfunctioning of the primary pulley holdback because there was no load sharing between the two holdbacks.

Subsequently the two holdbacks were replaced. The suppliers of the holdback insisted on installing a load sharing system which was designed and supplied by themselves, to ensure that at all times there is continuous load sharing between the two holdbacks. Linked hydraulic cylinders with an accumulator were used in this instance. (Refer to figure 6). Alternatively, Ringfeder type cupped springs can be used as installed at Phalaborwa's P.M.C. big in-pit incline belt.

6.5 Recommendations

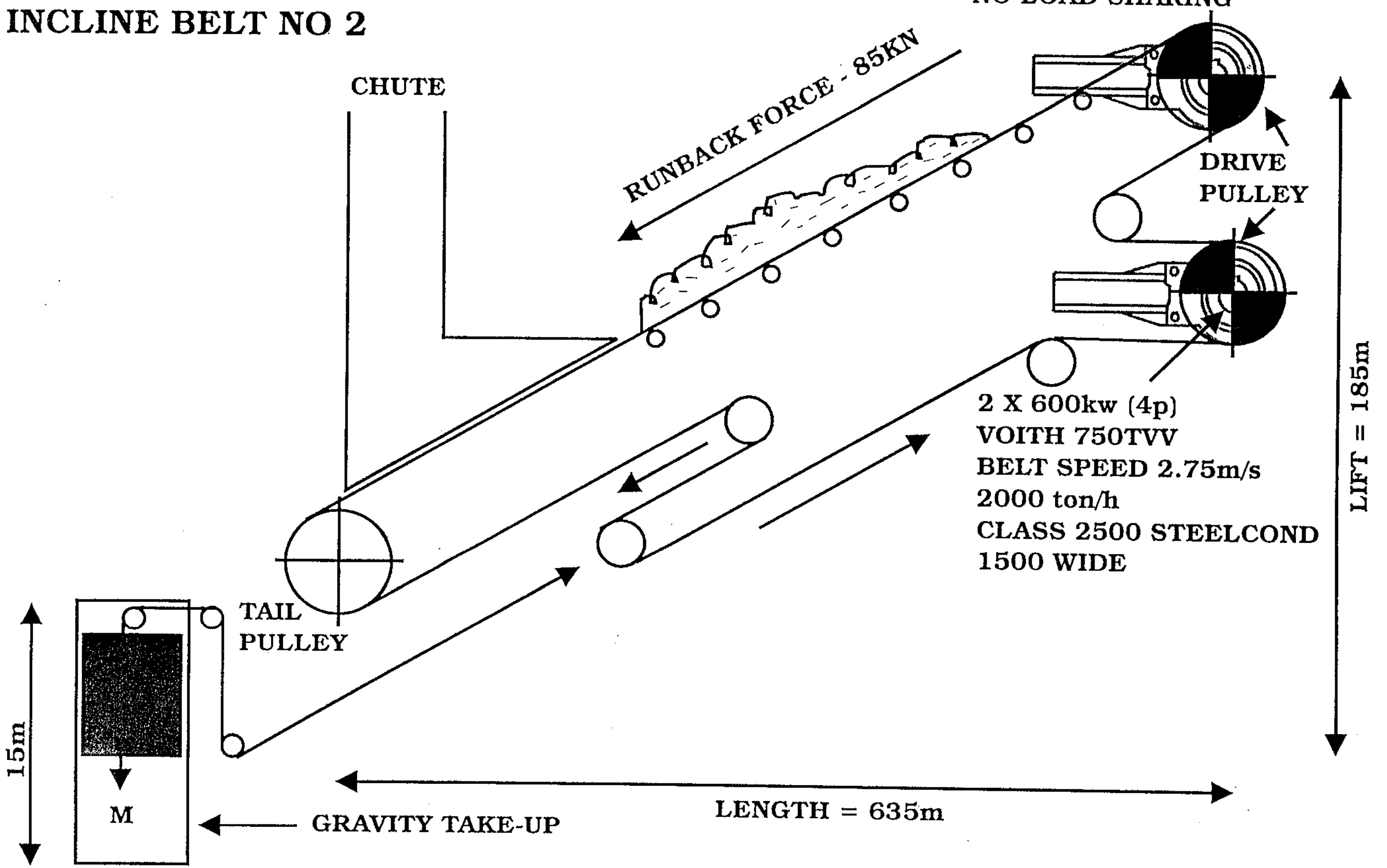
The fitting of high speed holdbacks to gearboxes, on multiple drives is unacceptable, due to the load sharing problems. No two gearboxes have exactly the same ratios, belt stretch and pulley lagging are not similar. (Leads to a major problem when the pulley lagging wears). It is impossible to obtain 100% load sharing on two holdbacks, even if they are fitted to one drive pulley.

When using two backstops, whether high speed or low speed, one must ensure that there is some device that will allow each holdback to take half the load.

Some high speed holdbacks are now fitted with a slip type clutch to help load sharing. One unit will rotate until the others engages. This is a good idea, but the setting can be tampered with and the clutch material wears. With this in mind, one should consider using a drum or disc brake at less than half the price of these units.

FIGURE 5

SASOL 2 + 3 BOSJESSPRUIT COLLIERY INCLINE BELT NO 2



7. INCLINES RUNBACK DUE TO BELT BREAKAGE

Some mines are experiencing major problems due to belts snapping, causing the conveyor to run away with the material. Holdbacks in this instance cannot assist, as there is no T2 tension around the pulley and therefore the belt will run to the bottom of the shaft with the material.

If the belt is going to break, then it will most probably break at the highest point of tension. This point is just before the drive pulley, (which is always positioned near the top of the incline).

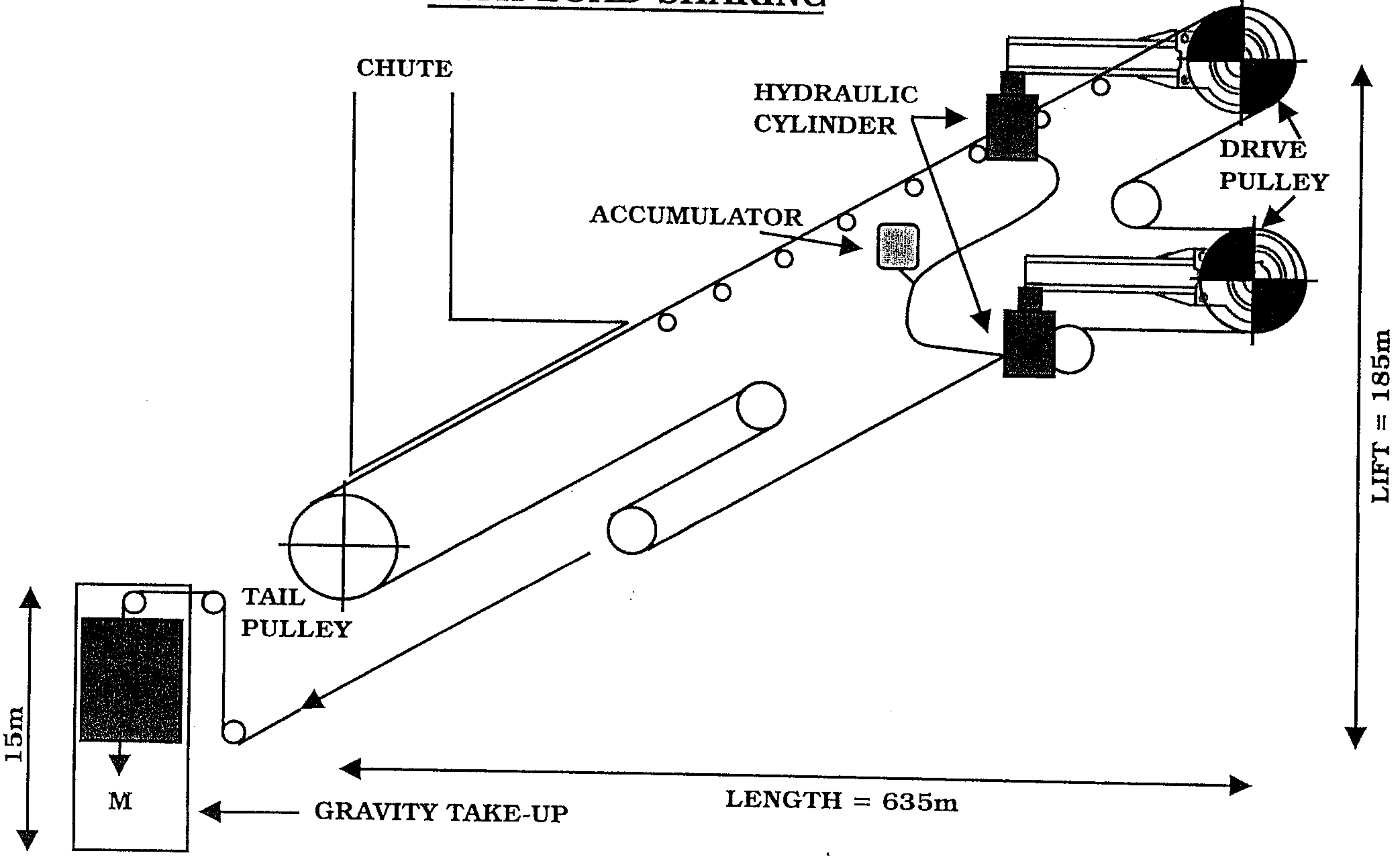
There is a device available which is fitted 10 metres away from the head pulley. It is actuated by a reverse switch fitted to the belt, which in turn operates a hydraulic ram, which grasps the belt when it starts running backwards. Unfortunately one would require at least 5 or more of these units stationed along the whole length of the conveyor to ensure 100% operation and safety.

Another possible solution to solve this broken belt scenario, is to somehow brake the idlers. The belt will slow down due to the friction between the idler face and the belt covers (irrespective of the conveyor being full or not). It is possible to fit a holdback/bearing in place of a standard idler bearing. These should be placed on the wing rolls and spaced evenly at every station.

This is an expensive exercise which will not actually hold the belt, but will no doubt ensure the belt will not accelerate on the incline, thus preventing damage to the steel work or belt, thereby paying for itself after one incident.

FI URE 6

SASOL 2 & 3 BOSJESSPRUIT COLLIERY INCLINE BELT NO 2. WITH LOAD SHARING



8. CONCLUSION

8.1 Size selection of the holdback/s

Do not compromise. The following factors must be taken into consideration.

a) Normal runback force mgh, due to the material mass, less the friction due to the idlers multiplied by the radius of the pulley will give a torque.

Runback force (N) = (mgh - Idler resistance force)
M = Material mass (kg/m)
g = Gravity 9.81 m/sec^2
h = Vertical lift (m)

b) The maximum holdback torque one can experience is during an aborted start. Peak starting torques of 150% of Te, which the fluid coupling can reach during startup. Consequently the belt is now over tensioned to accelerate the load. By stopping during the starting sequence, the excess tension between the drive pulley (fitted with a backstop) and up to where the load starts on the belt, will remain under the starting tension of 150% of normal running tension Te. This torque must be considered.

c) The shaft diameters must be checked to the maximum bore of the holdback and if the shaft can transmit the torque, due to an aborted start.

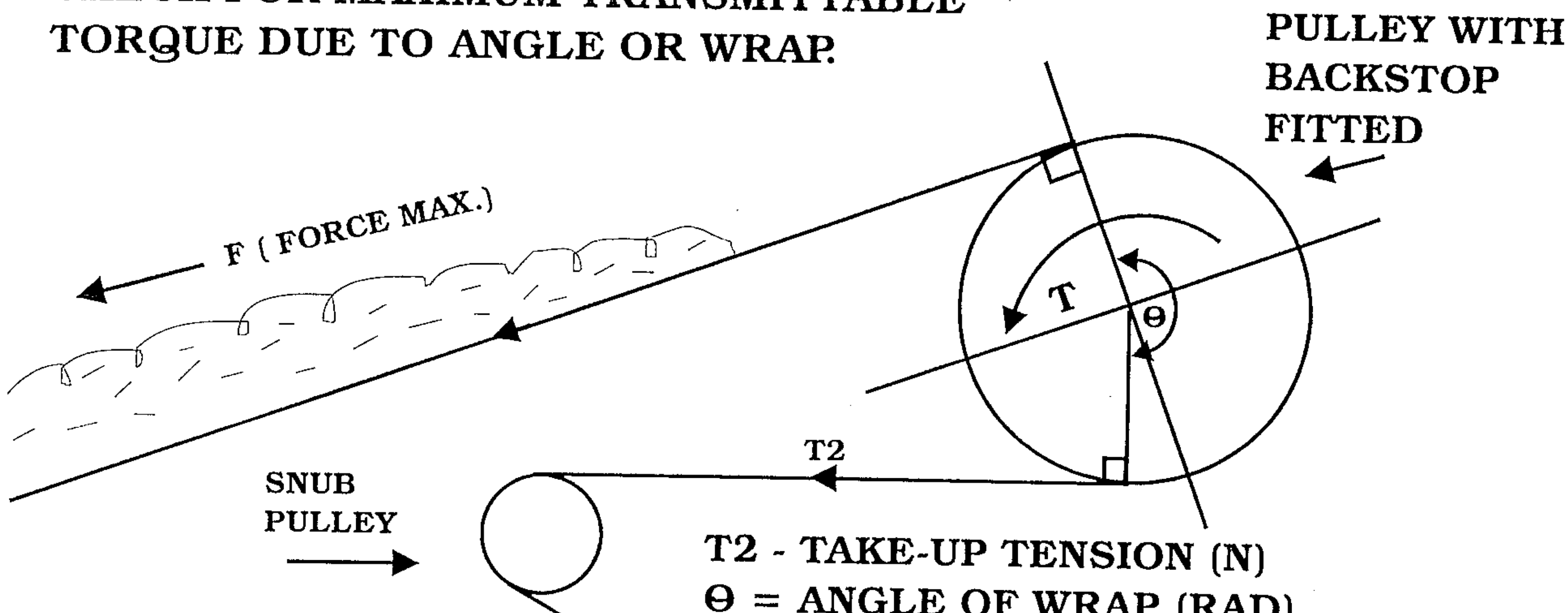
8.2 Position of holdback:

It is said:- "If two pulleys are required to accelerate the belt, two holdbacks will be required to stop the belt from running backwards".

It is recommended that if using only one holdback, the safest position for the holdback is at the primary pulley or secondary pulley, for maximum angle of wrap.

<u>FIGURE 7</u>

CHECK FOR MAXIMUM TRANSMITTABLE TORQUE DUE TO ANGLE OR WRAP.



T2 - TAKE-UP TENSION (N)
Θ = ANGLE OF WRAP (RAD)
μ = FRICTION FACTOR

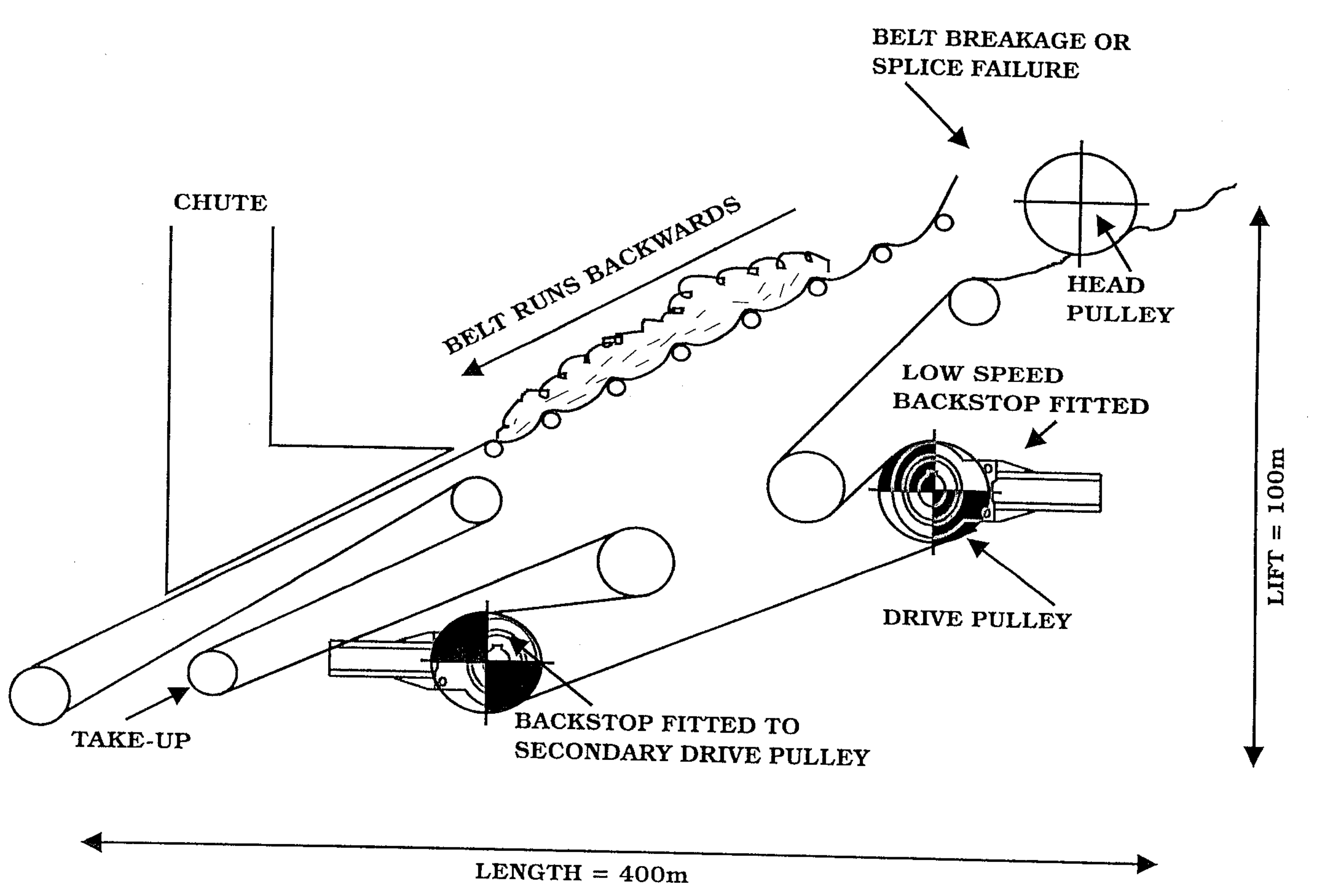
Operating conditions	Pulley surface			
	Bright steel pulley (smooth)	Polyurethane friction lining (herringbone grooves)	Rubber friction lining (herringbone grooves)	Ceramic friction lining (porous, herringbone grooves)
dry	0.35 to 0.4	0.35 to 0.4	0.4 to 0.45	0.4 to 0.45
wet (clean water)	0.1	0.35	0.35	0.35 to 0.4
wet (contaminated with loam/ clay)	0.05 to 0.1	0.2	0.25 to 0.3	0.35

R = RADIUS (m)

<u>FORMULA</u>

F (MAX) = T2 ($e^{\mu\theta}$ -1)
MAXIMUM TRANSMITTABLE TORQUE T = F.R .(Nm)

FIGURE 9



BELT BREAKAGE OR
SPLICE FAILURE
CHUTE
BELT RUNS BACKWARDS
HEAD
PULLEY
LOW SPEED
BACKSTOP FITTED
DRIVE PULLEY
TAKE-UP
BACKSTOP FITTED TO
SECONDARY DRIVE PULLEY
LIFT = 100m
LENGTH = 400m

One holdback can be fitted when there are two drives installed. However, one must consider the most important factor - how much torque can the pulley transmit, due to T2 on the one side and the runback force on the other side. (For calculations refer to figure 7).

Again, if applying two holdbacks, it is recommended that some sort of load sharing system be installed.

9. RECOMMENDATIONS

9.1 VARIOUS TYPES OF LOW SPEED BACKSTOPS AND THEIR ASSOCIATED ADVANTAGES AND DISADVANTAGES

Metal Band Wrap-Down Clutch Type (Fig 3).

FIG. 3

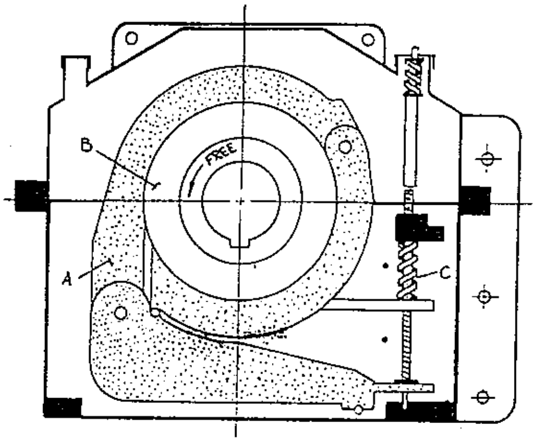


Fig 3

This type of backstop is basically a totally enclosed band brake which employs a metal band (a) and a boss or hub (b) which is keyed to the pully shaft. When free wheeling in direction of arrow, the spring (c) maintains light contact between the band and the rotating hub thereby minimizing any backlash when the unit reverts to its backstopping mode. When the conveyor stops and attempts to reverse, the band wraps down onto the hub preventing reverse rotation. The claimed advantage of this type of unit is its simplicity in design and construction. It has no roller bearings, only one moving part and employs split oil seals to facilitate easier replacement. However, these features are more likely to be a disadvantage as the absence of the roller bearings results in the rotating hub being supported only by the stationery casing, with the only relief from metal contact being the thin oil film. (See fig. 4).

Split oil seals are more susceptible to leakage and the single rotating hub, although lubricated, is subject to constant wear. In fact, the major disadvantage of this unit is that it requires periodic adjustment to compensate for such wear and therefore cannot be regarded as an automatic or failsafe backstop.

FIG. 4

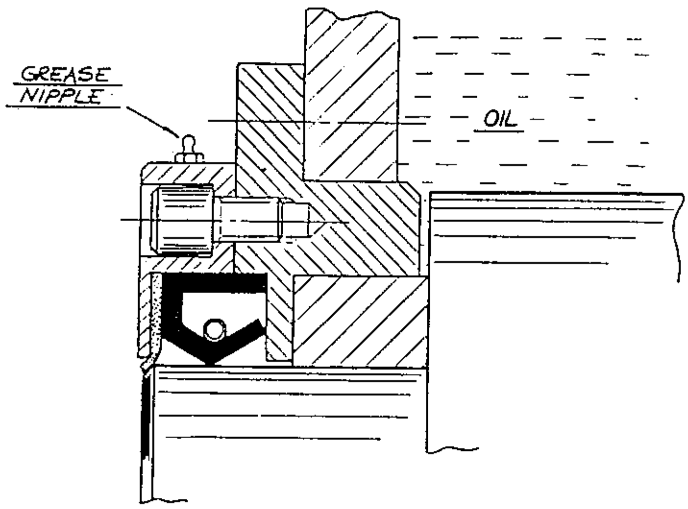


Fig 4

9.2 Roller and catcher plate type (Fig 5 A - C).

This is a somewhat complex backstop which claims the advantage of being lubrication free. A release system is available as on option in the form of a band type brake mounted to the unit. Such release systems should be avoided as they introduce a potential failure point to a backstop.

(A) When forward motion begins, the indexing rollers (8) and load pins (9) ride up the ramps on the stationary catchers (2) into the top of the slots in the control plate (5).

(B) When forward motion ceases, the indexing rollers locate on the ramps of the index rings (4).

(C) Reverse movement of the conveyor pulley drives the load pins down their angled slots until they locate with the ramps on the load rings (b) of the catcher and the opposing face of the wheel (6).
The disadvantage of this backstop type is that it suffers from excessive backlash during the transition from freewheel to backstopping modes.

FIG 5 A - C

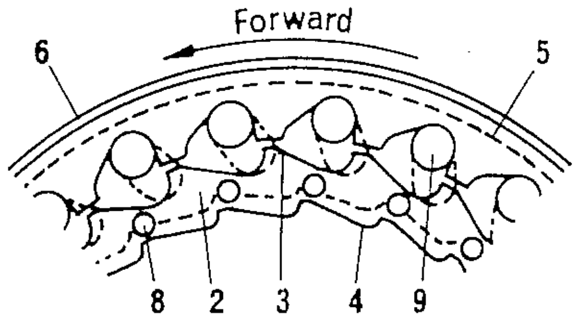


Fig 5A

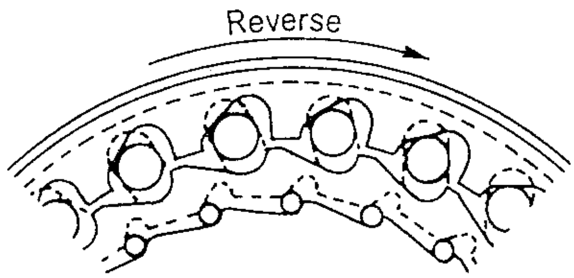


Fig 5B

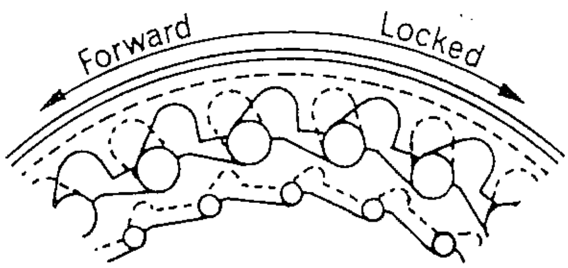


Fig 5C

9.3 Sprag type (Fig 6)

Several makes and configurations of this type of backstops exist but all are based on the same principal of operation. A series of irregularly shaped cams or sprags are positioned between two cylindrical races.

The inner race is connected to the pulley shaft and the outer race to the torque arm. During freewheel operation, the sprags are tilted to one side, enabling the inner cam to rotate freely beneath them. A spring connected to the sprags acts against the direction of tilt and pulls the sprags into contact with the outer race and rotating inner race, thereby eliminating backlash when the unit enters the backstopping mode. As the pulley stops and attempts to reverse, the compound curves on the upper and lower faces of the sprags cause a wedging action between the inner outer races, thus, preventing runback. The initial cost of this type of unit is usually lower than that of other backstop types since the inner and outer races are purely cylindrical and therefore simple to manufacture. On the other hand, the sprag elements are rather difficult to produce and reproduce constantly to the same precise dimensions. Irregularities can occur, resulting in uneven load distribution.

The greatest disadvantage of the sprag backstop is that the sprag elements are constantly rubbing against the inner and outer races. This constant wear can result in eventual roll over of the sprags which in turn means total failure of the backstop.

FIG. 6

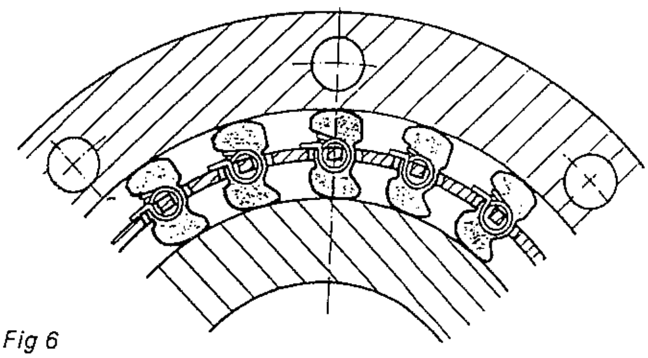


Fig 6

9.4 Roller-cam Clutch Type (Fig 7).

This type differs slightly from the normal sprag type, in that it employs a series of rollers between the cams which serve to support the outer race, thus eliminating the need for roller bearings. The cams operate on the same principle and are subject to the same wear pattern as the normal sprag type unit, therefore, it can be assumed that the same advantages and disadvantages apply to both types.

FIG. 7

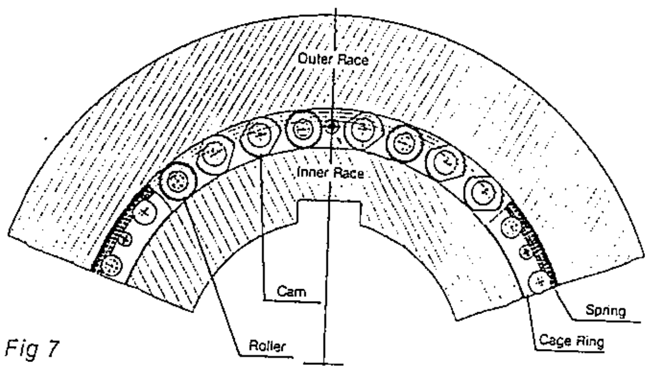


Fig 7

FIG. 8

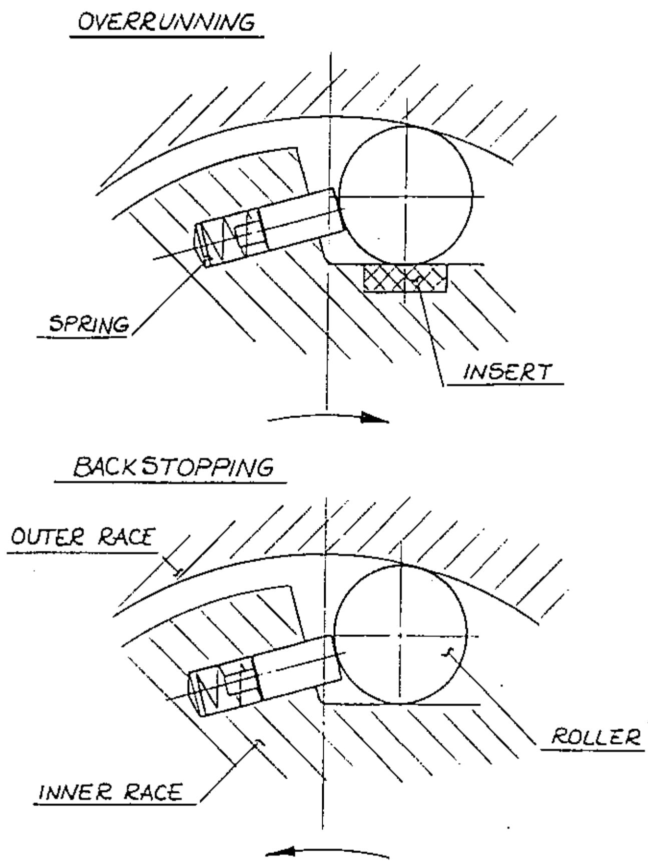


Fig 8

9.5 Individually spring loaded roller and ramp design (Fig 8).

This design of backstop consists of an inner race which is keyed to the pulley shaft and employs a series of ramps or cams machined around its perifery, a series of cylindrical rollers which are indiviually spring loaded and a cylindrical outer race which is connected to a torque arm. During freewheeling, the rollers rotate causing a skidding action on the outer race. The spring loaded pistons push the rollers outwards, maintaining contact between the rollers and the inner and outer races thereby eliminating any backlash. During backstopping, the rollers wedge between the inner and outer races preventing reverse rotation. The constant rubbing of the rollers against their respective pistons is likely to cause excessive wear at the contact points. The same applies to the contact points between the rollers and inner race.

Some manufacturers use hard metal inserts on the inner race ramp surfaces in an attempt to avoid the high cost of heat treating the inner cam. These inserts can be detrimental to the life of the backstop because the crushing force of the rollers between the inner and outer races could eventually cause the soft metal beneath these inserts to deform. To the extent that the inserts break loose causing certain backstop failure. It is also possible that this same deformation could cause the preceeding spring loaded piston to seize in its pocket.(See fig.9).

FIG. 9

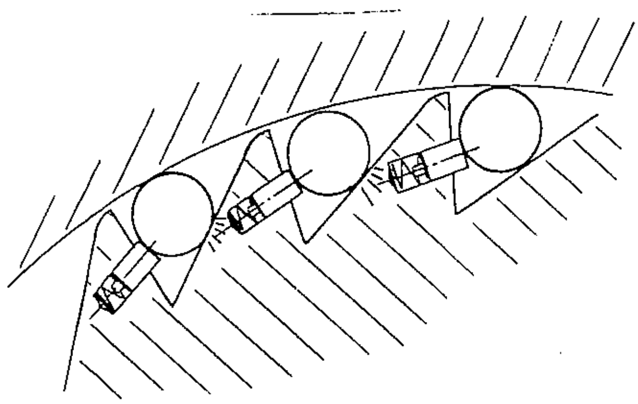


Fig 9

9.6 Roller cage and ramp design (Fig 10 A & B).

By far the most popular and successful design currently available, the roller cage type backstop is similar to the previous roller type. It uses the same basic principle of operation. However, the method used is somewhat different in that it avoids the use of numerous springs and pistons by employing a single spring loaded cage to position the rollers on the inner cam ramps.

FIG. 10A

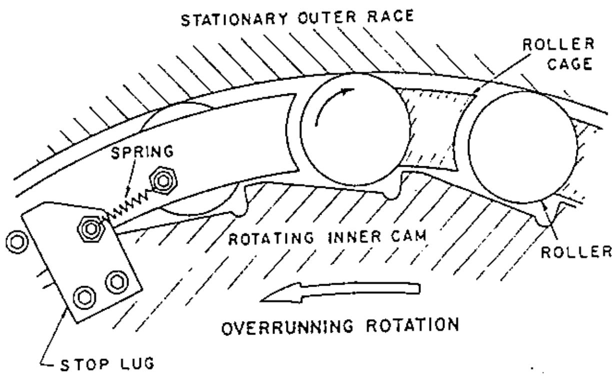


Fig 10A

Overrunning (Fig 10 A)

The rollers, roller cage and stop lugs rotate with the inner cam as a unit since they are connected by the energizing springs. The outer race does not rotate since it is bolted to the end covers, which are held by the backstop torque arm.

While overunning, the rollers roll on the outer race and slide on the inner cam ramps. Friction and centrifugal force tend to lift the rollers off the cam, minimizing contact and wear.

The energizing springs stretch during overunning to provide tension to the roller cage assembly. This tension keeps the rollers ready for instantaneous backstopping engagement and minimizes the relative rotation of the roller cage to the inner cam.

The stop lugs axially position the roller cage assembly on the inner cam. They also prevent the roller cage from rotating too far, which would cause the rollers to strike the upright side of the adjacent ramp

FIG. 10 B

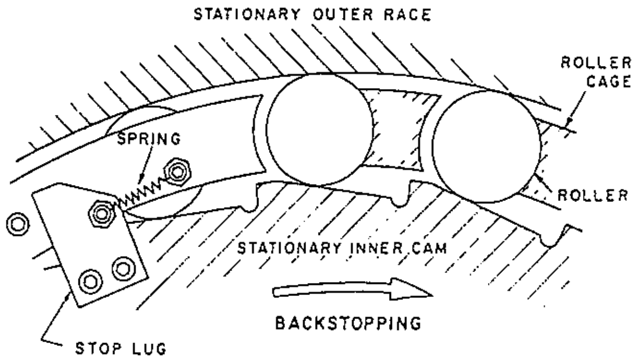


Fig 10B

Backstopping (Fig. 10B)

As the rotating shaft stops and attempts to reverse, the inner cam is instantly stopped by the wedging action of the rollers in the annular openings between the cam ramps and outer race.

From the outer race the backstopping torque is carried through the end covers to the torque arm and the adjoining superstructure.

All rollers are engaged simultaneously since they are positioned by the spring loaded roller cage.

Load division between the rollers is assured by machining accuracy of the inner cam ramps, rollers, roller cage and outer race.

As additional backstopping torque is applied to the inner cam, the rollers will tend to move deeper into into the wedging position, thereby increasing the resistance to slippage.

The torque capacity of the backstop is based on the tangential friction resistance force at the outer race developed by the compressive force between the inner cam rams, rollers and outer race.

The maximum torque capacity of the backstop is limited by the Hertzian contract stress at inner cam/roller and roller/outer race contact points, bending strength of torque arm, and hoop stress of outer race.

The fact that the rollers are positioned by a common spring loaded cage rather than individually spring elements means that load sharing between the rollers is not affected by spring failure or varying spring tension.

The only real disadvantage of this type of backstop is that of higher cost. Precise heat treatment of the inner cam and outer race is critical and the process is very costly. (Hard metal inserts are not used). The roller cage to is a complicated item to machine and tight tolerances must be maintained.

The other disadvantage found was that due to the mass of the roller, the one size would take all the load (see Fig 11 A). Now with the new tolerance compensation spring (as shown on Fig 11 B). The Gripper is now locally manufactured with this spring for best roller load sharing at all times.

These brief descriptions serve only as a indication of the basic types of backstops available and some of the more obvious advantages or disadvantages associated with each type. Factors such as material strengths and hoop stresses etc., all of which play an important role in good backstop design have not been dealt with.

2. AXIAL RETENTION

Low speed backstops are generally mounted to the pulley shaft by means of a slide fit in the hub with a side fitting key. The key alone is unlikely to be sufficient to prevent axial movement (creeping) of the backstop on the shaft, thus some other form of axial restraint must be used. Failure to retrain a backstop axially can result in undue forces being applied to the bearings and other internal components causing eventual failure.

3. ADEQUATE SEALING

It is essential that a backstop is adequately sealed against loss of lubricant and entry of foreign materials. Inadequate sealing and lack of axial restraint is the major cause of backstop failure.

The Gripper has been designed for the ultimate in sealing, even the high pressure water problem. (Note Fig. 11 C) back to back lip seals and two grease cavities.

"GRIPPER" HOLDBACK ROLLER POSITION

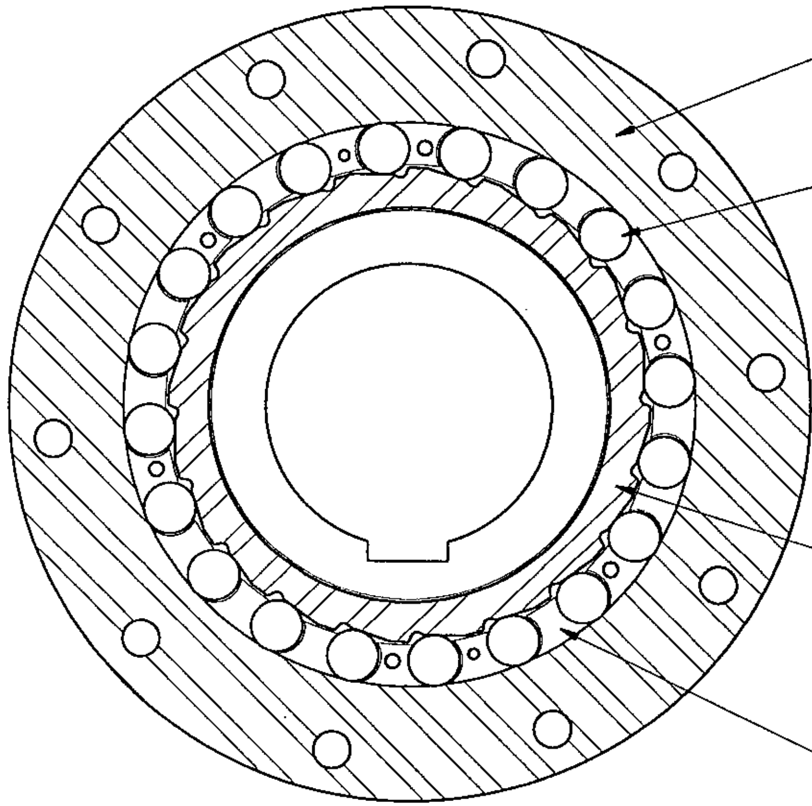


Fig. 11 A

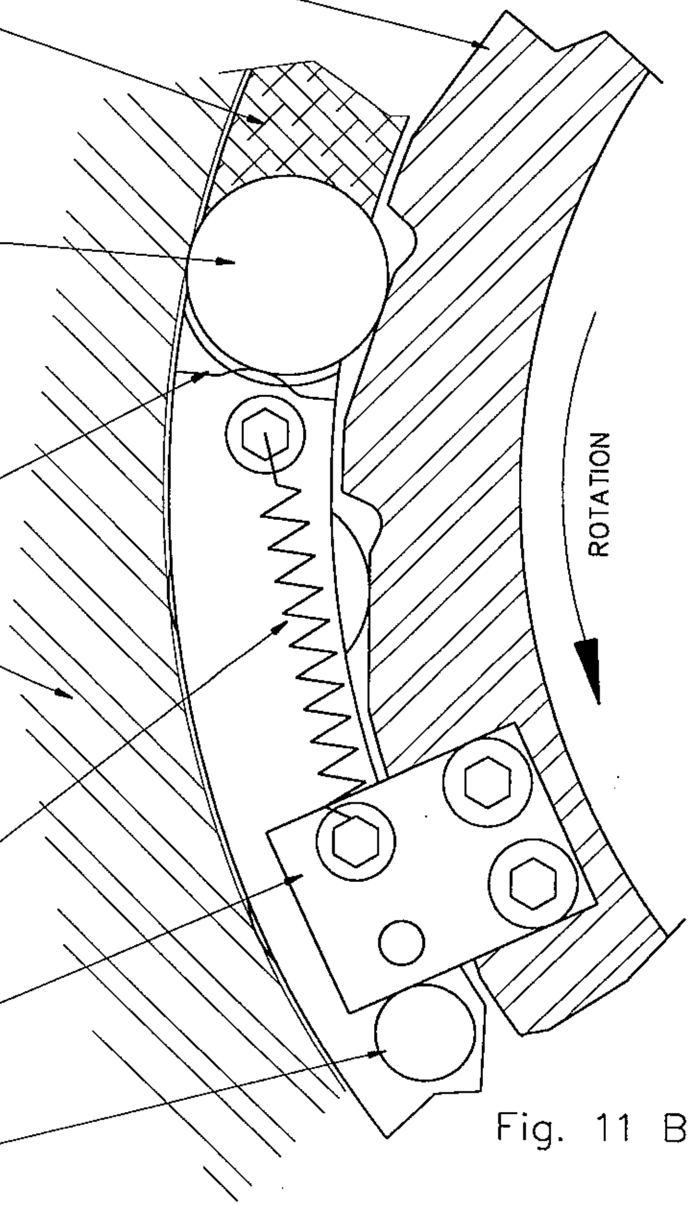


Fig. 11 B

HB14038A

10. ACTUAL EXAMPLE OF A CONVEYOR BELT

Surtees offer a design service to assist with the selection of the unit and the placing of the holdback.

Example of a conveyor design, namely SASOL 2 and 3 new plant feed conveyor.

Design no. 1 is the standard design using a T2 take-up of 23 kNm giving a class 800 belt and a take-up mass of 4795 kg. Note that by using 824 diameter pulleys the maximum torque that a head pulley can transmit due to the 180 deg. wrap angle is 20.13 kNm. The runback torque calculated is 35.34 kNm. Therefore, it will not be suitable to fit the holdback to the head pulley in this situation as the belt will slip over the head pulley and run backwards.

Unfortunately the holdback cannot be fitted to primary pulley or the secondary pulley as we have 4 shaft mounted drives. The holdback must be fitted to the head pulley. Our only option would be to either increase the diameter of the head pulley, or alternatively increase the take-up mass.

Note our design No. 2. In this case, in order to standardise, the end user insisted that the conveyor should be fitted with a steel cord belt class 1250 instead of using a class 800 purposes. Note that the T2 tension was increased to 100 kNm and the mass required therefore would be 20387 kg. The torque that the head pulley can now transmit is 83.24 kNm. This is with a wrap angle of 180 deg. If the wrap angle of 200, the transmittable torque would be 98.59 kNm. We are now able to fit one holdback for this system.

Unfortunately due to the high tension, the shaft diameters on the head pulley and drives will increase. Note page 4, that from design 1, was 229 mm calculated at the drive pulley and 245 mm at the head pulley and on design no. 2, drive pulley 256 mm versus head pulley 268 mm. This is a substantial extra cost for these larger shaft and bearing diameters.

'GRIPPER' HOLDBACK SEALING ARRANGMENT

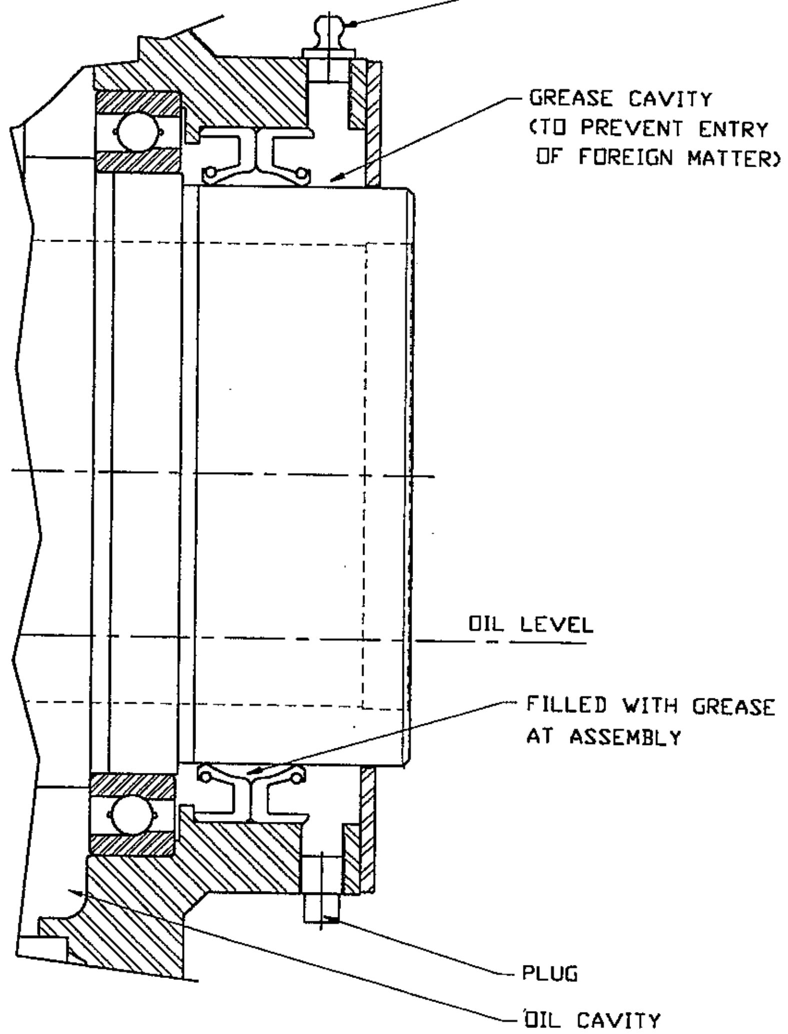
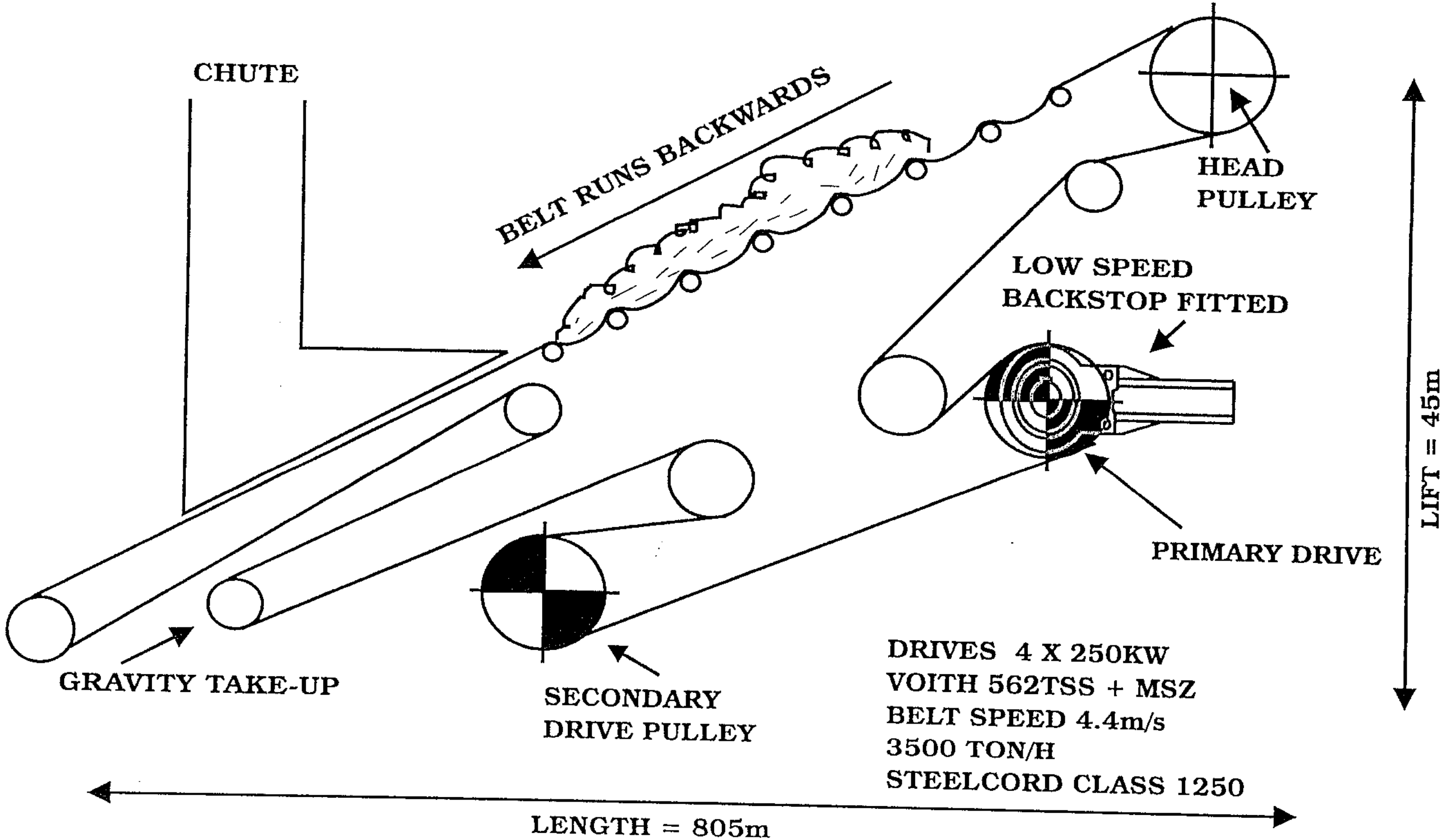


Fig. 11 C

FIGURE 10

SASOL 2+3 PLANT FEED (SECUNDA)



SURTEES AND SON (PTY) LTD

CONVEYOR 1

CONVEYOR START UP PROGRAM & GRIPPER HOLDBACK SELECTION PROCEDURE

CONVEYOR PARAMETERS

DATE:-	09/27/95	
CLIENT:- CONVEYOR NO:-	1.00	
SASOL 2 & 3 NEW FEED		
NUMBER OF DRIVES:	4	
NUMBER OF DRIVE PULLEYS:-	2	
LENGTH (L)	805.00	m
HEIGHT FROM TAIL TO HEAD PULLEY	45.00	m
FULL BELT SPEED (Vf)	4.40	m/sec
MATERIAL MASS IN TONS PER HOUR	3500.00	ton/h
BELT MASS (Mg')	35.60	kg/m
TROUGHING IDLER MASS IN Kg. (ENTER 0 IF UNKNOWN)	18.00	kg
TROUGHING IDLER SPACING	1.50	m
CALCULATED TROUGHING IDLER MASS per. METRE.	12.00	kg/m
RETURN IDLER MASS IN Kg.(ENTER 0 IF UNKNOWN).	16.00	kg
RETURN IDLER SPACING	3.00	m
CALCULAED RETURN IDLER MASS per. METRE.	5.33	kg/m
EST. TOTAL DRIVE/S MASSES ON BELT LINE(0 IF UNKNOWN)	1000	kg
EST. TOTAL PULLEY MASSES ON BELT LINE(0 IF UNKNOWN)	12000	kg

SELECT TYPE OF FLUID COUPLING:-TRACTION "T" USE 180% START FACTOR.
DELAY "TV" USE 150% START FACTOR.
DOUBLE DELAY "TVV" USE 140% START FACTOR.
TURBO SOFT START "TSS" USE 130% START FACTOR.
ACCELERATION CONTROL "TPE" USE 120% START FACTOR.

SELECTED STARTING FACTOR (%)	130	%
DRIVE PULLEY DIA (D1)	824	mm
HEAD PULLEY DIA (D2)	824	mm
PRIMARY OR HEAD DRIVE PULLEY ANGLE OF WRAP (0)	200	(DEG)
SECONDARY DRIVE ANGLE OF WRAP (ENTER 0 IF ONE DRIVE)	200	(DEG)
HEAD PULLEY ANGLE OF WRAP (0)	180	(DEG)
PULLEY FACE	1700	mm
PULLEY BEARING CENTRES	2220	mm
FRICTION FACTOR (U) (OLD=0.35,NEW=0.4)	0.35	
BELT WIDTH	1500	mm
TYPE OF BELT USED (STEELCORD=1,FABRIC=2 & COTTON=3)	1	
HEIGHT FROM HEAD TO DRIVE PULLEY_(0 IF DRIVE AT HEAD)	45.00	m
HORIZONTAL LENGTH FROM HEAD TO DRIVE(0 IF DRIVE ATHEAD)	152	m
HORIZONTAL LENGTH BEFORE RISE_(ENTER 0 IF INCLINE)	613	m
ENTER INSTALLED MOTOR POWER PER DRIVE.	250	Kw.
START UP TIME REQUIRED	25.00	sec.

CONVEYOR CALCULATIONS

SELECTED LENGTH CO-EFFICIENT	1.1100	
SELECTED f VALUE	0.0200	
DRIVE PULLEY R.P.M.	101.98	r.p.m.
CALCULATED POWER (ABSORBED) @ FULL LOAD:-		
FULL LOAD ABSORBED POWER REQUIRED AT MOTOR/S.	715.87	Kw
FULL LOAD ABSORBED POWER PER DRIVE.	178.97	Kw
FULL LOAD ESTIMATED STARTING TIME FROM BREAKAWAY.	25.12	Sec
CALCULATED EMPTY BELT:-		
EMPTY BELT ABSORBED POWER REQUIRED AT MOTOR/S Pa	74.01	Kw
EMPTY ESTIMATED STARTING TIME FROM BREAKAWAY (t)	4.86	Sec
CALCULATED STOPPING TIME & DISTANCE:-		
ESTIMATED COASTING TIME FOR LOADED BELT:-	7.32	Sec.
ESTIMATED COASTING DISTANCE FOR LOADED BELT:-	16.10	m
H.S.BRAKING TORQUE NEEDED (O IF NO BRAKE):-	0	Nm.
L.S.TAIL BRAKE TORQUE NEEDED FOR STOPPING TIME:-	0	Nm.

SELECTED TENSIONS:-

EFFECTIVE TENSION (TE)(G24)	153065	N
TENSION T2 (T2 TO ISO) T2=Te/(e^aθ)	16837	N
TENSION T2=T2*START FACTOR*eff.	21013	N
TENSION T2,Pins*eff/Vel*K.	23520	N
TENSION T2,(2% SAG AT TAIL)Ts-.15LMb+Mbgh:-	22468	N
SELECTED TENSION T2,MAX ABOVE	23520	N
TENSION (T1) STEADY OPERATION	176585	N
TENSION (T1) STARTING:-	222722	N
TENSION BETWEEN PRIMARY & SECONDARY DRIVES:-	100053	N
TENSION AT TAIL:-	27007	N
LIMITATION DUE TO BELT SAG (MAX 2% SAG.):-		
MINIMUM TENSILE FORCE NOT EXCEED T2 (ON CARRYING SIDE):-	23595	KN
SELECTED BELTING TYPE:-	STEELCORD	
SELECTED MINIMUM BELT TENSION (STEADY OPERATION)	788.75	KN/m
SELECTED MINIMUM BELT TENSION (STARTING)	816.65	KN/m
BELT CLASS.(CHECK BELTING CAT. FOR NO. OF PLIES & COVERS)	800	Class
DYNAMICS TEST FOR IDEAL STARTING & STOPPING:-		
MIN. TORQUE RAMP UP OR STOPPING TIME (AT T.REL=5):-	2.24	Sec.
TORQUE RAMP PLUS STARTING TIME FROM BREAKAWAY, LOADED:-	27.35	Sec.
MINIMUM TIME DELAY BETWEEN DRIVES,(CONSULT SURTEES):-	0.56	Sec.
MIN. VERTICAL & COVEX CURVE TO PREVENT BELT LIFT OFF, & EDGE TENSION.		
MIN RADIUS @ START FACTOR FOR COUPLING SLECTED:-	570	m
TAKE UP DETAILS.		
TAKE-UP MOVEMENT TOTAL L /2 FOR VERTICAL GRAVITY TYPE.L=	3.81	m
TAKE-UP MASS=T2*2/g. FOR VERTICAL GRAVITY TYPE.	4795	Kg

SELECTED EQUIPMENT SCHEDULE:-

HIGH SPEED FLUID COUPLING SELECTED:-		
SELECTED VOITH H.S.FLUID COUP. SIZE.(ON INSTALLED POWER):-	562	
SELECTED VOITH FLUID COUPLING TYPE:-	TSS-MSA	
LOW SPEED GRIPPER HOLDBACK IF REQUIRED:-		
GRIPPER HOLDBACK IF NECESSARY:-	REQUIRED	
LOW SPEED COUPLING SELECTION:-		
1.GRIDFLEX COUPLING FOR FOOT MOUNTED GEARBOXES:-		
SIZE SELECTED ON ABSORBED TORQUE:-	1130T10	
CATALOGUE TORQUE RATING:-	57000	Nm.
MAXIMUM BORE CAPACITY:-	165	mm
SIZE SELECTED ON DRIVE PULLEY SHAFT DIA.:-	1100T10	
CATALOGUE TORQUE RATING:-	5700	Nm.
MAXIMUM BORE CAPACITY:-	110	mm
2. RINGFEDER TYPE 30 RIGID FLANGE COUPLINGS FOR SHAFT MOUNTED GEARBOXES		
SIZE SELECTED ON ABSORBED TORQUE:-	SLE 400/115	
CATALOGUE TORQUE RATING:-	35200	Nm.
MAXIMUM BORE CAPACITY:-	115	mm
SIZE SELECTED ON DRIVE PULLEY SHAFT DIA.:-	SLE 400/115	
CATALOGUE TORQUE RATING:-	35200	Nm.
MAXIMUM BORE CAPACITY:-	115	mm
VOITH HIGH SPEED DRUM BRAKE:-		
1. SELECTED ON ENTERED BRAKE TORQUE ABOVE PER DRIVE:-	NO BRAKE	
DRUM DIAMETER:-	NO BRAKE	mm
RATED TORQUE:-	NO BRAKE	Nm.

SURTEES AND SON (PTY) LTD

CONVEYOR START UP PROGRAM & GRIPPER HOLDBACK SELECTION PROCEDURE

CONVEYOR PARAMETERS

DATE:- 08/22/95

CLIENT:-	CONVEYOR NO:-	2.00
SASOL 2 & 3 NEW FEED		
NUMBER OF DRIVES:	4	
NUMBER OF DRIVE PULLEYS:-	2	
LENGTH (L)	805.00	m
HEIGHT FROM TAIL TO HEAD PULLEY	45.00	m
FULL BELT SPEED (Vf)	4.40	m/sec
MATERIAL MASS IN TONS PER HOUR	3500.00	ton/h
BELT MASS (Mg')	35.60	kg/m
TROUGHING IDLER MASS IN Kg. (ENTER 0 IF UNKNOWN)	18.00	kg
TROUGHING IDLER SPACING	1.50	m
CALCULATED TROUGHING IDLER MASS per. METRE.	12.00	kg/m
RETURN IDLER MASS IN Kg.(ENTER 0 IF UNKNOWN).	16.00	kg
RETURN IDLER SPACING	3.00	m
CALCULAED RETURN IDLER MASS per. METRE.	5.33	kg/m
EST. TOTAL DRIVE/S MASSES ON BELT LINE(0 IF UNKNOWN)	1000	kg
EST. TOTAL PULLEY MASSES ON BELT LINE(0 IF UNKNOWN)	12000	kg

SELECT TYPE OF FLUID COUPLING:-TRACTION "T" USE 180% START FACTOR.
DELAY "TV" USE 150% START FACTOR.
DOUBLE DELAY "TVV" USE 140% START FACTOR.
TURBO SOFT START "TSS" USE 130% START FACTOR.
ACCELERATION CONTROL "TPE" USE 120% START FACTOR.

SELECTED STARTING FACTOR (%)	130	%
DRIVE PULLEY DIA (D1)	824	mm
HEAD PULLEY DIA (D2)	824	mm
PRIMARY OR HEAD DRIVE PULLEY ANGLE OF WRAP (0)	200	(DEG).
SECONDARY DRIVE ANGLE OF WRAP (ENTER 0 IF ONE DRIVE)	200	(DEG).
HEAD PULLEY ANGLE OF WRAP (0)	180	(DEG).
PULLEY FACE	1700	mm
PULLEY BEARING CENTRES	2220	mm
FRICTION FACTOR (U) (OLD=0.35,NEW=0.4)	0.35	
BELT WIDTH	1500	mm
TYPE OF BELT USED (STEELCORD=1,FABRIC=2 & COTTON=3)	1	
HEIGHT FROM HEAD TO DRIVE PULLEY_(0 IF DRIVE AT HEAD)	45.00	m
HORIZONTAL LENGTH FROM HEAD TO DRIVE(0 IF DRIVE ATHEAD)	152	m
HORIZONTAL LENGTH BEFORE RISE_(ENTER 0 IF INCLINE)	613	m
ENTER INSTALLED MOTOR POWER PER DRIVE.	250	Kw.
START UP TIME REQUIRED	25.00	sec.

CONVEYOR CALCULATIONS

SELECTED LENGTH CO-EFFICIENT	1.1100	
SELECTED f VALUE	0.0200	
DRIVE PULLEY R.P.M.	101.98	r.p.m.
CALCULATED POWER (ABSORBED) @ FULL LOAD:-		
FULL LOAD ABSORBED POWER REQUIRED AT MOTOR/S.	715.87	Kw
FULL LOAD ABSORBED POWER PER DRIVE.	178.97	Kw
FULL LOAD ESTIMATED STARTING TIME FROM BREAKAWAY.	25.12	Sec
CALCULATED EMPTY BELT:-		
EMPTY BELT ABSORBED POWER REQUIRED AT MOTOR/S Pa	74.01	Kw
EMPTY ESTIMATED STARTING TIME FROM BREAKAWAY (t)	4.86	Sec
CALCULATED STOPPING TIME & DISTANCE:-		
ESTIMATED COASTING TIME FOR LOADED BELT:-	7.32	Sec.
ESTIMATED COASTING DISTANCE FOR LOADED BELT:-	16.10	m
H.S.BRAKING TORQUE NEEDED (O IF NO BRAKE):-	0	Nm.
L.S.TAIL BRAKE TORQUE NEEDED FOR STOPPING TIME:-	0	Nm.

SELECTED TENSIONS:-

EFFECTIVE TENSION (TE)(G24)	153065	N
TENSION T2 (T2 TO ISO) T2=Te/(e^ua)	16837	N
TENSION T2=T2*START FACTOR*eff.	21013	N
TENSION T2,Pins*eff/Vel*K.	23520	N
TENSION T2,(2% SAG AT TAIL)Ts-.15LMb+Mbgh:-	22468	N
SELECTED TENSION T2,MAX ABOVE	100000	N
TENSION (T1) STEADY OPERATION	253065	N
TENSION (T1) STARTING:-	299202	N
TENSION BETWEEN PRIMARY & SECONDARY DRIVES:-	176533	N
TENSION AT TAIL:-	103487	N
LIMITATION DUE TO BELT SAG (MAX 2% SAG.):-		
MINIMUM TENSILE FORCE NOT EXCEED T2 (ON CARRYING SIDE):-	23595	KN
SELECTED BELTING TYPE:-	STEELCORD	
SELECTED MINIMUM BELT TENSION (STEADY OPERATION)	1130.36	KN/m
SELECTED MINIMUM BELT TENSION (STARTING)	1097.07	KN/m
BELT CLASS.(CHECK BELTING CAT. FOR NO. OF PLIES & COVERS)	1250	Class
DYNAMICS TEST FOR IDEAL STARTING & STOPPING:-		
MIN. TORQUE RAMP UP OR STOPPING TIME (AT T.REL=5):-	2.24	Sec.
TORQUE RAMP PLUS STARTING TIME FROM BREAKAWAY, LOADED:-	27.35	Sec.
MINIMUM TIME DELAY BETWEEN DRIVES,(CONSULT SURTEES):-	0.56	Sec.
MIN. VERTICAL & COVEX CURVE TO PREVENT BELT LIFT OFF, & EDGE TENSION.		
MIN RADIUS @ START FACTOR FOR COUPLING SLECTED:-	570	m
TAKE UP DETAILS.		
TAKE-UP MOVEMENT TOTAL L /2 FOR VERTICAL GRAVITY TYPE.L=	3.91	m
TAKE-UP MASS=T2*2/g. FOR VERTICAL GRAVITY TYPE.	20387	Kg

SELECTED EQUIPMENT SCHEDULE:-

HIGH SPEED FLUID COUPLING SELECTED:-		
SELECTED VOITH H.S.FLUID COUP. SIZE.(ON INSTALLED POWER):-	562	
SELECTED VOITH FLUID COUPLING TYPE:-	TSS-MSA	
LOW SPEED GRIPPER HOLDBACK IF REQUIRED:-		
GRIPPER HOLDBACK IF NECESSARY:-	REQUIRED	
LOW SPEED COUPLING SELECTION:-		
1.GRIDFLEX COUPLING FOR FOOT MOUNTED GEARBOXES:-		
SIZE SELECTED ON ABSORBED TORQUE:-	1130T10	
CATALOGUE TORQUE RATING:-	57000	Nm.
MAXIMUM BORE CAPACITY:-	165	mm
SIZE SELECTED ON DRIVE PULLEY SHAFT DIA.:-	1100T10	
CATALOGUE TORQUE RATING:-	5700	Nm.
MAXIMUM BORE CAPACITY:-	110	mm
2. RINGFEDER TYPE 30 RIGID FLANGE COUPLINGS FOR SHAFT MOUNTED GEARBOXES		
SIZE SELECTED ON ABSORBED TORQUE:-	SLE 400/115	
CATALOGUE TORQUE RATING:-	35200	Nm.
MAXIMUM BORE CAPACITY:-	115	mm
SIZE SELECTED ON DRIVE PULLEY SHAFT DIA.:-	SLE 400/115	
CATALOGUE TORQUE RATING:-	35200	Nm.
MAXIMUM BORE CAPACITY:-	115	mm
VOITH HIGH SPEED DRUM BRAKE:-		
1. SELECTED ON ENTERED BRAKE TORQUE ABOVE PER DRIVE:-	NO BRAKE	
DRUM DIAMETER:-	NO BRAKE	mm
RATED TORQUE:-	NO BRAKE	Nm.

HOLDBACK SELECTION STEPS :-

STEP 1:- SELECT HOLDBACK ON TOTAL SYSTEM ABS. POWER * START FACTOR, WHEN THE HOLDBACK IS FITTED TO HEAD PULLEY:-		
CALCULATED..,Ta*eff*%*No.drives.	81.98	KNm
STEP 2:- SELECT HOLDBACK ON TOTAL SYSTEM INSTALLED TORQUE, WHEN THE HOLDBACK IS FITTED TO HEAD PULLEY:-		
CALCULATED..,Ti*eff*No.drives.	93.64	KNm
STEP 3:- SELECTED ON RUNBACK TORQUE :-		
CALCULATED..,mgh - IDLER RESISTANCE FORCE ON INCLINE:-	35.34	KNm
STEP 4:- SELECTION ON MAX. TRANSMITTABLE TORQUE DUE TO PULLEY WRAP :-		
WHEN HOLDBACK IS FITTED TO HEAD PULLEY:-		
TENSION T2..,SELECTED T2 + BELT MASS HEAD TO DRIVE (mgh)	24.40	KN
CALCULATED..,T2(e^aθ):-	20.13	KNm
WHEN HOLDBACK IS FITTED TO SINGLE DRIVE PULLEY:-		
CALCULATED..,T2(e^aθ):-	23.19	KNm
WHEN HOLDBACK IS FITTED TO PRIMARY & SECONDARY DRIVE PULLEY:-		
CALCULATED ON PRIMARY...,T12(e^u0-1):-	23.19	KNm
CALCULATED ON SECONDARY...,T2(e^u0-1):-	23.19	KNm
CALCULATED USING ESCOM SPEC.(NWS 1556):-	75.53	KNm
CALCULATED USING ANGLO AMERICAN SPEC.(T2*K*Dd/2000):-	88.09	KNm
HOLDBACK SELECTION ON TORQUE USING ONLY ONE, IS:-		
SELECTION @ ABSORBED L.S.TORQUE * eff.(CATALOGUE TORQUE)=	63.06	KNm
CHECK MAXIMUM SYSTEM TORQUE, STEP 1 & 2 (PEAK TORQUE)===	81.98	KNm
FINAL GRIPPER HOLDBACK SELECTION, USE SIZE:-	HB115	NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING, WHICH IS:-	101.700	KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	152.550	KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	210	mm
HOLDBACK SELECTION ON TORQUE FOR PRIMARY & SECONDARY DRIVES:-		
SELECT EACH HOLDBACK ON 60% OF ABSORBED TORQUE========	37.84	KNm
FINAL GRIPPER HOLDBACK SELECTION, USE TWO OFF SIZE:-	1095	NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING, WHICH IS:-	37.968	KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	56.952	KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	145	mm

STEP 5:- MINIMUM PULLEY SHAFT DIAMETER:-

MATERIAL(ENTER 1 FOR EN3,2 # EN8,3 # EN9 & 4 # EN19)________	2	
MAXIMUM WORKING SHEAR STRESS FOR FATIGUE	70	MPa.
MAXIMUM WORKING COMBINED STRESS AT BEARING	100	MPa.
MAXIMUM WORKING SHEAR STRESS STATIC CONDITIONS	120	MPa.
CALCULATED BEARING CENTER TO HUB DISTANCE (a)	310	mm
PULLEY HUB SPACING ..FACE - 100	1600	mm
MAXIMUM ALLOWABLE DEFLECTION eg.1/2500(RINGFEDER 7012)_____	2500	
NORMAL RUNNING TOTAL SYSTEM TORQUE.(PULLEY POWER):-	63.06	KNm
RUNNING TORQUE ON EACH DRIVE PULLEY SHAFT.:-	15.77	KNm
MAX. BENDING MOMENT ON DRIVE PULLEY SHAFT.:-	42.28	KNm
MAX. BENDING MOMENT ON HEAD PULLEY SHAFT.:-	54.74	KNm
NETT TENSION ON DRIVE PULLEY.(P2=T1+T2) :-	273	KN
NETT TENSION ON HEAD PULLEY.(P2=T1+T1) :-	353	KN
CALCULATE DRIVE PULLEY SHAFT(IF DRIVE AT HEAD USE THESE DIA'S.):-		
CALC. DRIVE SHAFT DIA. BASED ON TORSION:-	105	mm
CALC. BEARING SHAFT DIA. COMBINED TORSION.(Te):-	182	mm
CALC. BEARING SHAFT DIA. COMBINED BENDING (Me):-	182	mm
CALC. HUB SHAFT DIA. BASED ON DEFLECTION (5 min.):-	216	mm
CALC .HUB SHAFT DIA. DEFLECTION (ESCOM 1:2500):-	229	mm
SELECTED HUB SHAFT DIA.(STD. RINGFEDER SIZE)==============	240	mm
SELECTED BEARING SHAFT DIA.=============================	190	mm
SELECT MINIMUM DRIVE OR HOLDBACK SHAFT DIA.==============	105	mm
CALCULATE HEAD PULLEY SHAFT:-		
CALC. HOLDBACK SHAFT DIA. BASED ON TORSION :-	166	mm
CALC. BEARING SHAFT DIA. COMBINED TORSION.(Te):-	192	mm
CALC. BEARING SHAFT DIA. COMBINED BENDING (Me):-	192	mm
CALC. HEAD SHAFT DIA. BASED ON DEFLECTION (5 min.):-	231	mm
CALC.HEAD SHAFT DIA. DEFLECTION (ESCOM 1:2500):-	245	mm
SELECTED HUB SHAFT DIA.(STD. RINGFEDER SIZE)==============	260	mm
SELECTED BEARING SHAFT DIA.=============================	200	mm
SELECT MINIMUM HOLDBACK SHAFT DIA.======================	170	mm
MINIMUM HOLDBACK SHAFT DIA. BASED ON PURE STATIC SHEAR STRESS @ Ta.total*%====	152	mm
HOLDBACK SELECTION ON MAX. BORE IS:-		
SELECTED HOLDBACK TO FIT ON HEAD PULLEY, SHAFT DIA.=======	170	mm
FINAL GRIPPER HOLDBACK SELECTION, USE SIZE:-	1105	NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING IS:-	61.020	KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	91.530	KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	200	mm
SELECTED HOLDBACK TO FIT ON DRIVE PULLEY, SHAFT DIA.=======	105	mm
FINAL GRIPPER HOLDBACK SELECTION, USE SIZE:-	1085	NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING IS:-	21.696	KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	32.544	KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	130	mm

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HOLDBACK SELECTION STEPS :-

STEP 1:- SELECT HOLDBACK ON TOTAL SYSTEM ABS, POWER * START FACTOR, WHEN THE HOLDBACK IS FITTED TO HEAD PULLEY:-	
CALCULATED..,Ta*eff*%*No.drives.	81.98 KNm
STEP 2:- SELECT HOLDBACK ON TOTAL SYSTEM INSTALLED TORQUE, WHEN THE HOLDBACK IS FITTED TO HEAD PULLEY:-	
CALCULATED..,Ti*eff*No.drives.	93.64 KNm
STEP 3:- SELECTED ON RUNBACK TORQUE :-	
CALCULATED..,mgh - IDLER RESISTANCE FORCE ON INCLINE:-	35.34 KNm
STEP 4:- SELECTION ON MAX. TRANSMITTABLE TORQUE DUE TO PULLEY WRAP :-	
WHEN HOLDBACK IS FITTED TO HEAD PULLEY:-	
TENSION T2..,SELECTED T2 + BELT MASS HEAD TO DRIVE (mgh)	100.88 KN
CALCULATED..,T2(e^a)):-	83.24 KNm
WHEN HOLDBACK IS FITTED TO SINGLE DRIVE PULLEY:-	
CALCULATED..,T2(e^a)):-	98.59 KNm
WHEN HOLDBACK IS FITTED TO PRIMARY & SECONDARY DRIVE PULLEY:-	
CALCULATED ON PRIMARY...,T12(e^u0-1):-	98.59 KNm
CALCULATED ON SECONDARY...,T2(e^u0-1):-	98.59 KNm
CALCULATED USING ESCOM SPEC.(NWS 1556):-	75.53 KNm
CALCULATED USING ANGLO AMERICAN SPEC.(T2*K*Dd/2000):-	374.55 KNm
HOLDBACK SELECTION ON TORQUE USING ONLY ONE, IS:-	
SELECTION @ ABSORBED L.S.TORQUE * eff.(CATALOGUE TORQUE)=	63.06 KNm
CHECK MAXIMUM SYSTEM TORQUE, STEP 1 & 2 (PEAK TORQUE)===	81.98 KNm

FINAL GRIPPER HOLDBACK SELECTION, USE SIZE:-	HB115	NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING, WHICH IS:-	101.700	KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	152.550	KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	210	mm

HOLDBACK SELECTION ON TORQUE FOR PRIMARY & SECONDARY DRIVES:-	
SELECT EACH HOLDBACK ON 60% OF ABSORBED TORQUE=======	37.84 KNm

FINAL GRIPPER HOLDBACK SELECTION, USE TWO OFF SIZE:-	1095 NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING, WHICH IS:-	37.968 KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	56.952 KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	145 mm

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STEP 5:- MINIMUM PULLEY SHAFT DIAMETER:-

MATERIAL(ENTER 1 FOR EN3,2 # EN8,3 # EN9 & 4 # EN19)_____	2
MAXIMUM WORKING SHEAR STRESS FOR FATIGUE	70 MPa.
MAXIMUM WORKING COMBINED STRESS AT BEARING	100 MPa.
MAXIMUM WORKING SHEAR STRESS STATIC CONDITIONS	120 MPa.
CALCULATED BEARING CENTER TO HUB DISTANCE (a)	310 mm
PULLEY HUB SPACING ..FACE - 100	1600 mm
MAXIMUM ALLOWABLE DEFLECTION eg.1/2500(RINGFEDER 7012)____	2500
NORMAL RUNNING TOTAL SYSTEM TORQUE.(PULLEY POWER):-	63.06 KNm
RUNNING TORQUE ON EACH DRIVE PULLEY SHAFT.:-	15.77 KNm
MAX. BENDING MOMENT ON DRIVE PULLEY SHAFT.:-	65.61 KNm
MAX. BENDING MOMENT ON HEAD PULLEY SHAFT.:-	78.45 KNm
NETT TENSION ON DRIVE PULLEY.(P2=T1+T2) :-	423 KN
NETT TENSION ON HEAD PULLEY.(P2=T1+T1) :-	506 KN
CALCULATE DRIVE PULLEY SHAFT(IF DRIVE AT HEAD USE THESE DIA'S.):-	
CALC. DRIVE SHAFT DIA. BASED ON TORSION:-	105 mm
CALC. BEARING SHAFT DIA. COMBINED TORSION.(Te):-	200 mm
CALC. BEARING SHAFT DIA. COMBINED BENDING (Me):-	200 mm
CALC. HUB SHAFT DIA. BASED ON DEFLECTION (5 min.):-	241 mm
CALC .HUB SHAFT DIA. DEFLECTION (ESCOM 1:2500):-	256 mm
SELECTED HUB SHAFT DIA.(STD. RINGFEDER SIZE)==============	260 mm
SELECTED BEARING SHAFT DIA.==============================	200 mm
SELECT MINIMUM DRIVE OR HOLDBACK SHAFT DIA.==============	105 mm
CALCULATE HEAD PULLEY SHAFT:-	
CALC. HOLDBACK SHAFT DIA. BASED ON TORSION :-	166 mm
CALC. BEARING SHAFT DIA. COMBINED TORSION.(Te):-	209 mm
CALC. BEARING SHAFT DIA. COMBINED BENDING (Me):-	209 mm
CALC. HEAD SHAFT DIA. BASED ON DEFLECTION (5 min.):-	252 mm
CALC.HEAD SHAFT DIA. DEFLECTION (ESCOM 1:2500):-	268 mm
SELECTED HUB SHAFT DIA.(STD. RINGFEDER SIZE)==============	280 mm
SELECTED BEARING SHAFT DIA.==============================	220 mm
SELECT MINIMUM HOLDBACK SHAFT DIA.=======================	170 mm
MINIMUM HOLDBACK SHAFT DIA. BASED ON PURE STATIC SHEAR STRESS @ Ta.total*%====	152 mm
HOLDBACK SELECTION ON MAX. BORE IS:-	
SELECTED HOLDBACK TO FIT ON HEAD PULLEY, SHAFT DIA.=======	170 mm

FINAL GRIPPER HOLDBACK SELECTION, USE SIZE:-	1105 NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING IS:-	61.020 KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	91.530 KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	200 mm

SELECTED HOLDBACK TO FIT ON DRIVE PULLEY, SHAFT DIA.======	105 mm

FINAL GRIPPER HOLDBACK SELECTION, USE SIZE:-	1085 NRT
CHECK HOLDBACK "CATALOGUE TORQUE" RATING IS:-	21.696 KNm
CHECK HOLDBACK "PEAK TORQUE" RATING, WHICH IS:-	32.544 KNm
CHECK HOLDBACK MAXIMUM BORE CAPACITY:-	130 mm

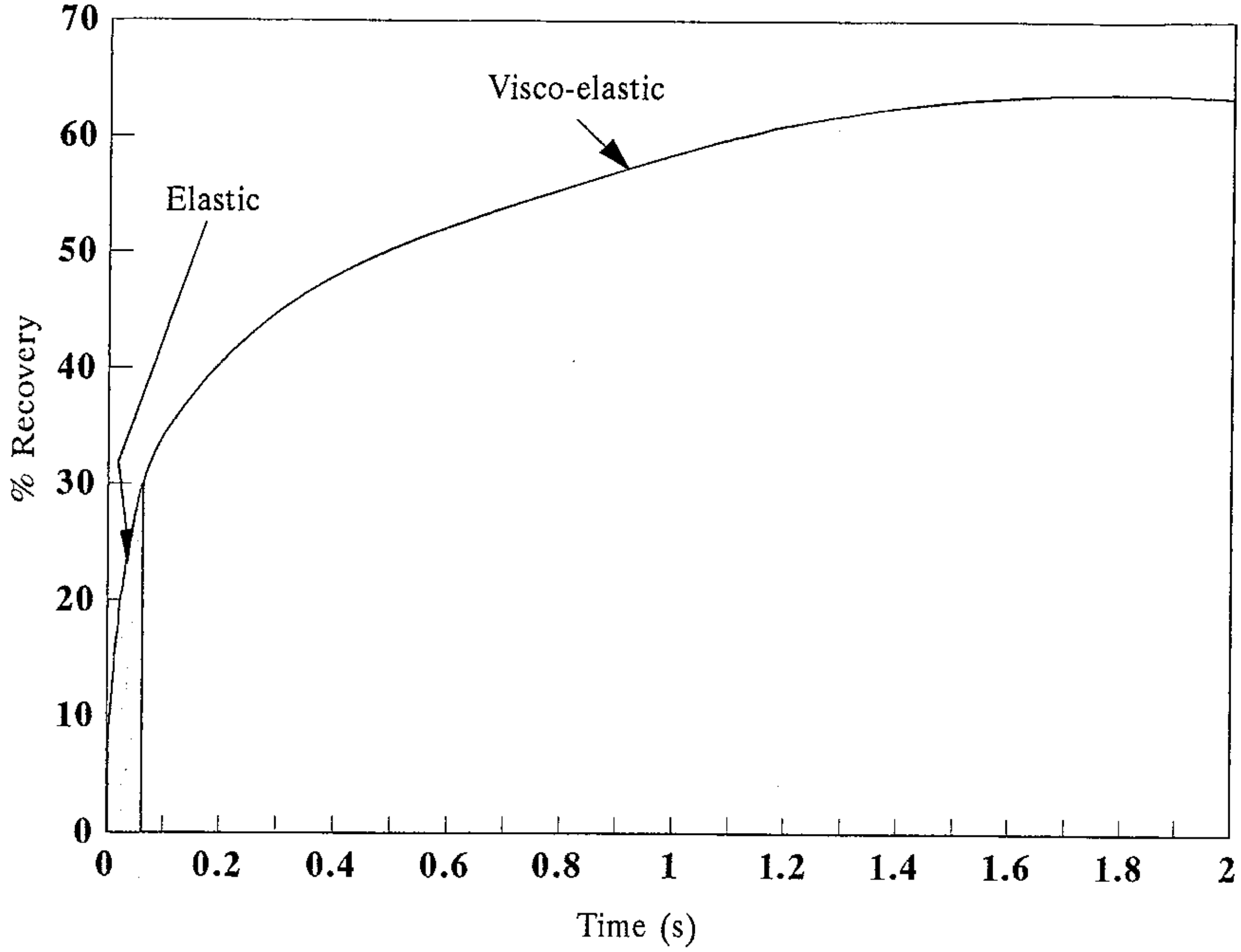

FIGURE 3: SHEAR RECOVERY

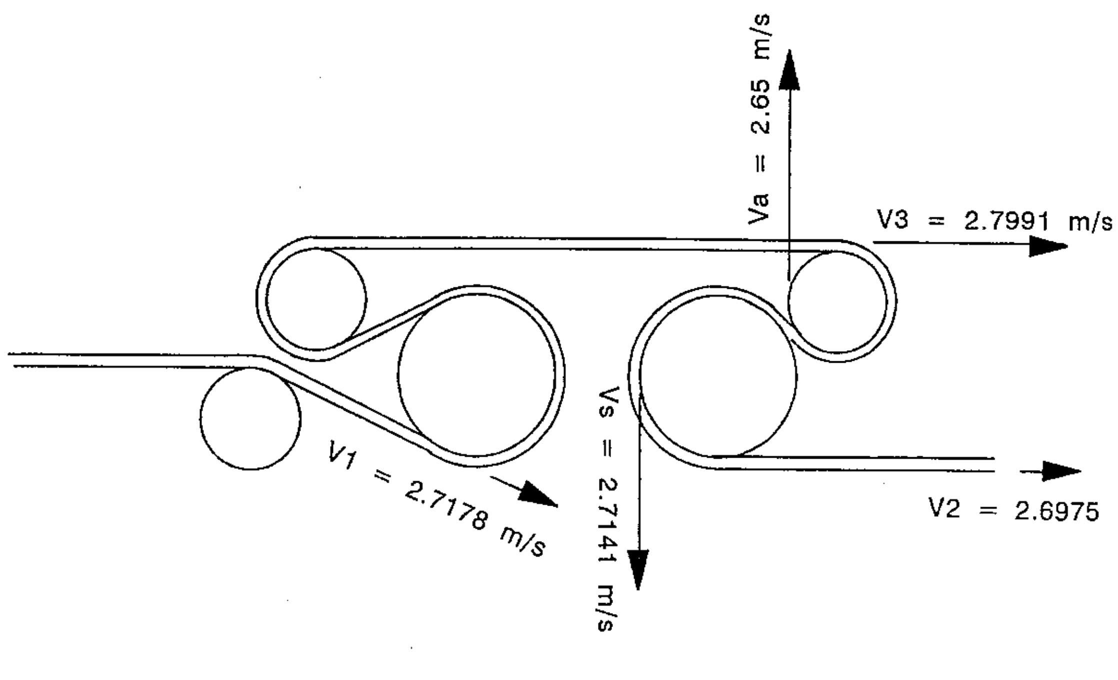

FIGURE 4: BELT SPEEDS THROUGH WELBECK DRIVE

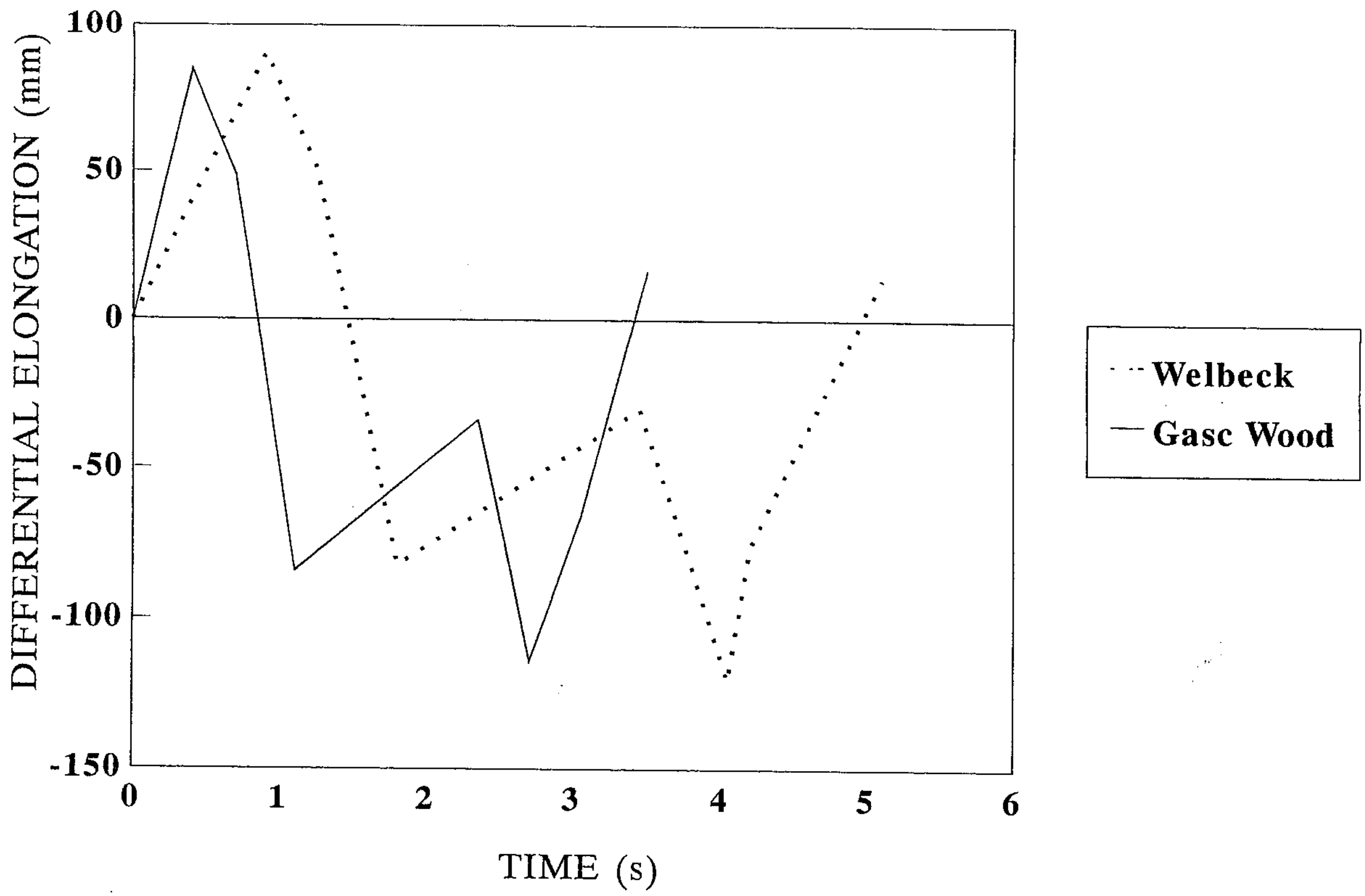

FIGURE 5: DIFFERENTIAL ELONGATION WRT TIME

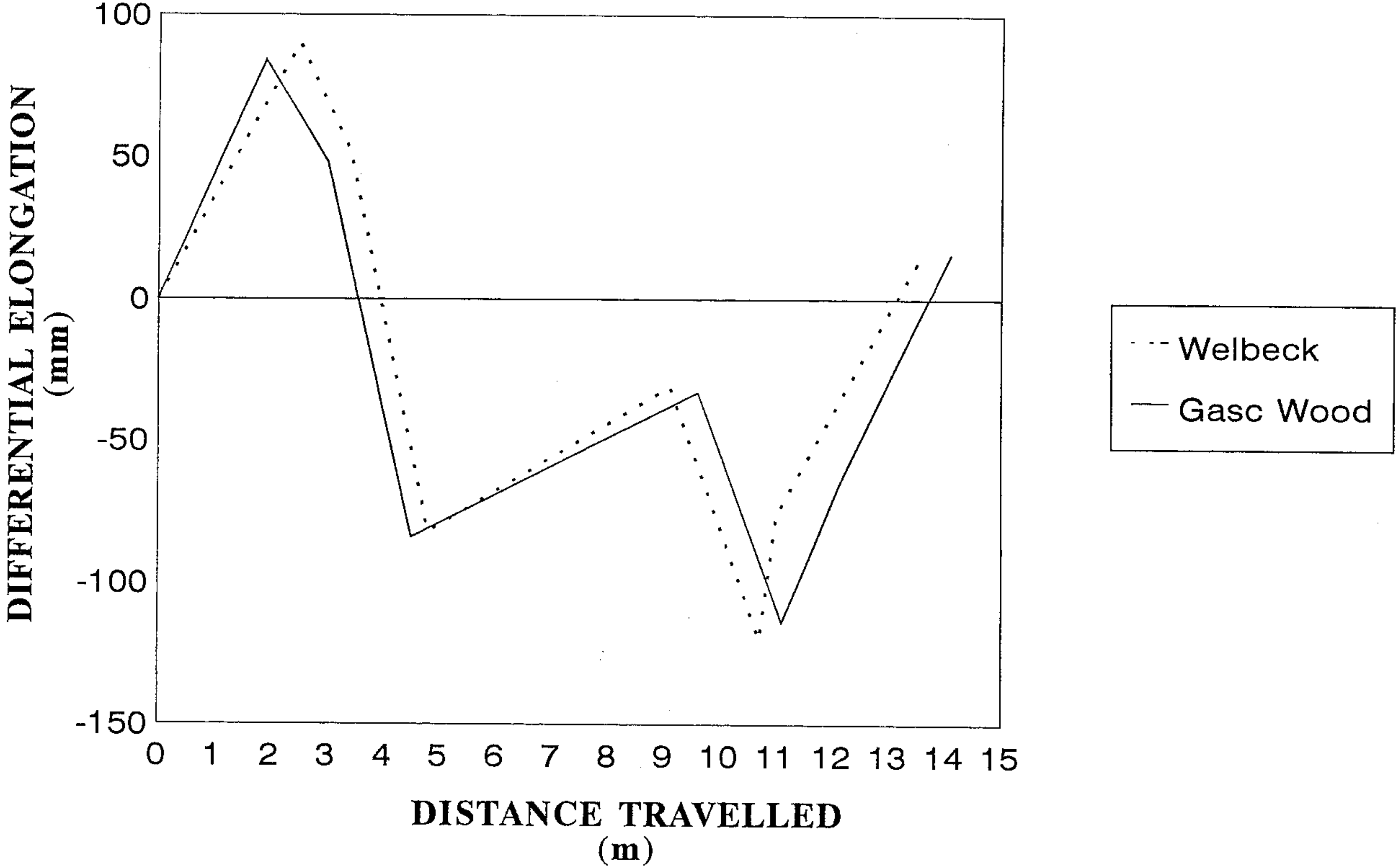

FIGURE 6: DIFFERENTIAL ELONGATION WRT DISTANCE

11. REFERENCES

- DEUTCHSCHE NORM - Belt conveyors for Bulk Materials (DIN 22101)

- CSIR REPORT -

  "Investigation into the catastrophic failure of an underground conveyor belt system.
  by a James & R. Fricke

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