Goedehoop Colliery – Conveyor B18 revisited

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1. SYNOPSIS.

Those delegates that have been attending Beltcon conferences regularly will recall that at Beltcon 5 in 1989, Mr. Hector Dreyer of Amcoal presented a paper [Ref 1] on conveyor B18 at Goedehoop. Since the original installation in 1984/85, the conveyor has been extended by Amcoal from its original length of around 1 600m to the current 2 400m. This paper describes the design, construction and commissioning of the modifications to the 1 200t/h capacity 2 400m long, horizontally curved overland conveyor. The conveyor incorporates two horizontal curves turning the path through 20,32°, in plan. The extension of the system formed part of the recent upgrade undertaken at Goedehoop Colliery. The entire design, static and dynamic simulations, were carried out within the Central Technical Office of the Anglo American Corporation of S.A. Ltd.

2. PROJECT BACKGROUND.

Goedehoop Colliery is situated around 30km from the town of Witbank. With Landau and Bank Collieries it forms part of South African Coal Estates. During 1995 Amcoal approached the Central Technical Office (CTO) of The Anglo American Corporation and requested that certain conveyors should be designed to be installed by a sub-contractor at their Goedehoop Colliery. These conveyors would form part of an upgrade of the existing facilities at the mine complex (see figure 1). The design of the conveyors was approached in three phases:

a. The design of the underground and shaft conveyors up to the silos.
b. The upgrade and extension of the existing overland conveyors.
c. The design of some new conveyors and the upgrade of others in the treatment plant along with a new stockpile reclaim conveyor.

a. The design of the underground and shaft conveyors up to the silos.

The responsibilities of the CTO started with the design of the two inclined shaft conveyors, from the two-seam shaft and from the four-seam shaft. These conveyors carry the coal delivered by the trunk conveyors to surface. The four-seam conveyor, after emerging from underground, continues to the top of the four-seam silo. The two-seam conveyor terminates shortly after it reaches the surface, where the coal is transferred onto a second conveyor, which transports the coal to the top of the two-seam silo.

b. The upgrade of the existing overland conveyors.

The upgrading of the overland conveyors, including conveyors B15 and B18, was investigated. At the same time an additional overland conveyor, B14, was designed to connect the mine site silos with the extended tail end of conveyor B15, to complete the conveyor chain to the main treatment plant, about 10 km away.

The silos at the mine site feed onto conveyor B14 via belt feeders. B14 is around 2 800m long and in a straight line, although the conveyor undulates a little and the head is 2,9m below the tail. From conveyor B14 the material is transferred onto conveyor B15, which is about 4 870m long with a single horizontal curve. The conveyor has a lift of around 29m. The design of this conveyor provided a few challenges, but nothing like the conveyor it transfers onto, namely conveyor B18, which discharges directly into the primary crusher building bins.

c. New conveyors and upgrade of plant conveyors.

Various conveyors were upgraded and several new conveyors were designed for the treatment plant. These conveyors were all relatively short and no special design was required.

3. HISTORY OF CONVEYOR B18.

When conveyor B18 was first designed in 1984, a capacity of 1 200t/h was required and the conveyor was around 1 600m long. In the original operating condition the belt was fed by belt feeders at the silo at the tail, which is about 32m above the head. Due to the poor dynamic behaviour of the belt, the use of fabric belting and a regeneration of power down the hill, the maximum capacity that was ever achieved on the belt was around 800t/h. Dynamic analysis was in its infancy at that time and was not used when the belt was first designed. The upgraded conveyor is required to carry a similar capacity but over a longer distance with a greater fall from the tail to the head.

4. EQUIPMENT REQUIREMENTS.

a. Capacity.

The material conveyed is run-of-mine coal with a bulk density of between 0,85t/m^3 and 1t/m^3. The design capacity of the upgraded conveyor was initially specified at 800t/h, with the future requirement of 1 200t/h. At the originally specified capacity, the conveyor was operated on only two power packs after installation. As a result of successful operation of the system at this capacity, the higher specified capacity was achieved within three months after commissioning, requiring the third power pack to be installed.

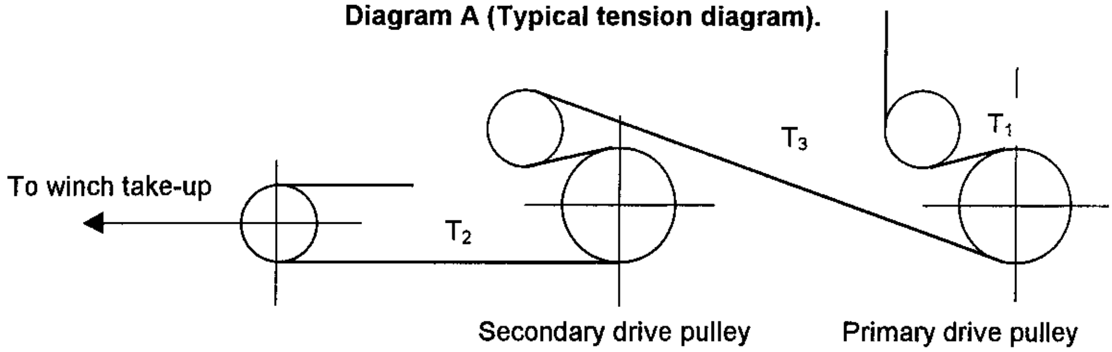
b. Belt width and speed selection.

The belt width of 1 200mm was selected to suit the existing structure and idlers. A belt speed had to be selected in accordance with the required capacity and process requirements. High speed conveying was defined by the mine as conveyors operating at a speed faster than 4m/s. In this instance we advised a belt speed of around 3,75m/s which would be suitable from both a belt loading and operational viewpoint. The belt is loaded at around 62% at a design capacity of 1 200t/h. A 100% belt loading adheres to the minimum freeboard specified in the ISO 5048 standard. The belt selected was a class ST500 steelcord belt to specification SABS1366. Class ST500 is a relatively common specification in that mining region and the selection ensures that spare belting would be readily available.

Additional tension was required since the belt runs downhill for a large section and the normal overall T_2 calculation, based on the overall profile, produced a very low value. T_2 is the tension that prevents the belt from slipping at the drive pulley both under start-up and normal running. An additional 14,5kN tension was added to the system by increasing the take-up tension at the winch. This was the maximum tension that could be added without exceeding the limits of the belt class. Based on the minimum tension determined by the dynamic analysis and the maximum effective tension (T_e), the "tight" side tension T_1 was calculated to be 84,407kN. The well-known ratio of $T_1/T_2 = e^{\mu\theta}$, at the point of drive slip, indicated that the selected design angle of 235° was adequate.

Here: T_1 = Tight side tension (kN)
T_2 = Slack side tension (kN)
T_3 = Tension between the drive pulleys (kN)
e = Naperian log base
μ = Coefficient of friction between the drive pulley and the belt
θ = Angle of belt wrap around the pulley.

Diagram A (Typical tension diagram).



It is interesting to note that the elastic properties of the steelcord belt are very much different to those of the fabric belt used previously. The belt modulus of the steelcord belt is about 28 360 kN/m width whereas the specific modulus of the class 630/3ply fabric belt is about 3 650 kN/m width. This made a marked difference to the dynamic behaviour of the belt. The dynamics of the conveyor in the uprated condition can be compared to the original system by comparing the required minimum torque rise times for the two installations. The minimum torque rise time (t) (the so-called Funke line) can be determined from:

$$t = \frac{5 \cdot L}{\sqrt{\frac{(W \cdot E \cdot S_{ir})}{(M_{ir} + B)}}}$$, where L = Belt line length on the return strand (m)

W	=	Belt width	(m)
B	=	Belt mass	(kg/m)
E	=	Belt modulus	(kg/m width)
S_{ir}	=	Return idler pitch	(m)
M_{ir}	=	Idler mass of rotating parts	(kg/m)

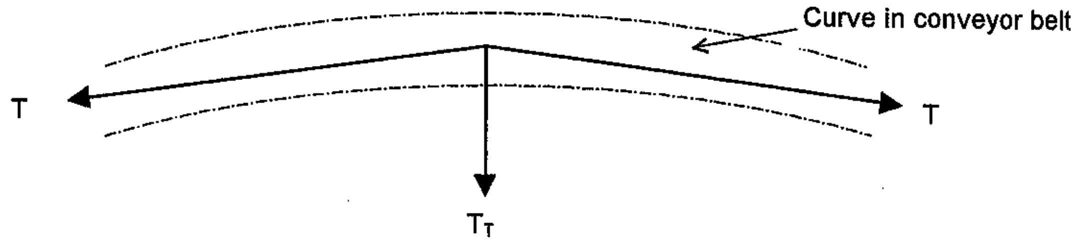
From this a minimum torque rise time for the original installation of 9,6 seconds was obtained, while the value for the lengthened installation was 5,9 seconds. The relatively large torque rise time for the original installation was difficult to achieve with the fluid transmission couplings installed, hence the severe dynamics under accelerating and decelerating conditions.

c. The route.

Coal has to be conveyed from the head of conveyor B15 to B18 along a route that was predetermined by existing servitudes and land tenure (see figure 1 and 2). The choice was between the installation of an additional straight-line conveyor from the head of conveyor B15 to an extended tail of conveyor B18, or the extension of the existing conveyor B18 though a horizontal curve to reach the head of B15. The curve radius, selected in accordance with the design, could easily be accommodated within the space available. If the 'two conveyors' option was selected, the tail of B18 would have had to be extended anyway, since there would have been little space for a transfer if left in its original position. The original tail end also incorporated a manual winch take-up, which was used for belt permanent elongation, when the original system was equipped with a fabric carcass belt. The tail take-up was used only to reduce the overall length of the automatic (elastic) take-up at the head end. The tail take-up was therefore a further obstacle to a transfer at that point. Either way, the conveyor path had to negotiate a truck access-way to a nearby mini-pit. The mine easily solved this problem by using culverts to form a bridge over the conveyor structure for the trucks. The existing conveyor already crossed underneath a series of railway lines near the head-end curve in a similar fashion. Access to drive houses and transfer houses was in no way affected by the extension of the tail of the conveyor. It made economic sense to extend the existing conveyor rather than install a new conveyor, thereby eliminating an extra transfer point and an additional drive station, along with its associated maintenance and potential spillage. Figure 2 shows a plan of the conveyor route.

The horizontal tail curve and head curve radii are 3 040m and 1 000m respectively, and the overall included angle was 20.32°. The subtended angle at the tail-curve is 10,944° and 9,374° at the head-curve. The drive house is located on the ground about fifty metres behind the head with the automatic take-up structure at the drive-house. Revision of the existing secondary curve near the head of the conveyor was most important in the upgrade since the distance to the tail had increased by 800m. The probability of belt pull-out in the second horizontal curve increased in proportion since the tension (T_x) in the belt at the first tangent point of the second horizontal curve had increased, thereby increasing the resultant force T_T (see diagram B). A review of the idler packing and the curve-banking angle at this point was therefore required.

Diagram B (Typical curve vector diagram).



5. DESIGN PARAMETERS.

Based on the above conveyor requirements, the initial designs were prepared by the design office. The following results were produced:-

Belt width:-	1 200mm	
Belt specification:-	Class 500/steelcord, to SABS 1366	
Belt speed:-	3.75m/s	
Tension distribution:-	T_1 = 84,407kN (Tight side tension)	
	T_2 = 14,500kN (Slack side tension)	
	T_e = 69,907kN (Effective tension)	
	T_t = 40,134kN (Tail tension)	
Take-up	Horizontal winch type	
Power requirements:-	Empty belt -	142kW (Absorbed power)
	Loaded belt -	255kW (Absorbed power)
Motor selection:-	3 x 110kW motors	
Reducer ratio:-	20,9:1	
Start-up procedure:-	Soft start using differential flow, drain type couplings.	

Pulley diameters:-	
Drive pulleys:-	1 000mm
High tension:-	800mm
Low tension:-	630mm

Length:-	2 445m
Lift:-	-21,65m (total from tail to head)
Max. lift:	-42m (from bottom of dip to head)
Design capacity:-	1210t/h
Material carried:-	Coal
Lump size:-	-150mm
Bulk density:-	0,85 - 1,0t/m^3

A conservative design philosophy was adopted during the design of the upgraded conveyor since an attempt was made to accommodate some of the existing equipment. Tried and tested technology was used rather than new, unproven methods. With this in mind, only equipment familiar to the mine was selected.

6. DYNAMIC ANALYSIS.

The use of steady state design techniques alone is usually only adequate in the case of relatively short plant conveyors where belt flexibility does not significantly affect the behaviour of the belt during starting and stopping. In the case of long overland conveyors of variable gradient, tension waves generated at the drive and braking pulleys take a considerable amount of time to propagate along the length of the flexible belt. More detailed dynamic analysis is therefore required to ensure acceptable system behaviour under start-up and shut-down conditions. Elements of system behaviour that cannot be adequately predicted using static analysis alone include peak belt tensions, displacement of take-ups, belt slippage over drive pulleys and forces generated in holdback devices. During the original design of conveyor B18 some years ago, dynamic analysis was something very new and still unavailable to the average engineer. It was therefore much more difficult to make accurate predictions prior to originally installing and commissioning conveyor B18.

Dynamic analysis of the up-rated overland conveyor was carried out within the AAC CTO using ACSL (Advanced Continuous Simulation Language). This is a general-purpose language based on FORTRAN. [Ref. 2]

6.1 Start-up and stopping sequences.

For long centre distance conveyors, start-up and stopping times are critical for the efficient operation of these machines. Therefore these sequences have to be engineered very carefully in order to ensure that dynamic stress waves are minimised. The main source of stress waves in the belt is the sudden change in tension giving rise to a tension stress wave. It follows then that the introduction of torque in a controlled manner will minimise the dynamic misbehaviour of the conveyor during acceleration (see paragraph 4). In the case of conveyor B18, the start-up control was achieved by utilising differential flow fluid couplings, similar to units that have been used successfully on other mines and projects within Amcoal, particularly the curved overland conveyor at Landau Colliery [Ref 3].

6.2 Controlled start-up

The start-up of the conveyor is controlled by starting the primary and secondary drive motors sequentially at an arbitrarily selected time delay. This delay is not important since the fluid couplings are initially empty, and no torque is transmitted to the reducers in this condition. As soon as the motors are running at operating speed, oil is pumped into the couplings with the oil flow to each coupling initiated simultaneously. Torque is then transmitted through the reducers to the drive pulleys. At the re-commissioning stage, the belt reached full speed in a very short time, around 45 seconds. This produced high peak tensions and was due to a rapid rate of oil input into the coupling. The effect of this was poor belt training in the second horizontal curve under start-up and high load readings on the take-up winch load-cell. A contributing factor was that the automatic winch take-up was very slow in responding and the initial tension settings were too high. The dynamic analysis indicated that the belt should run up to speed in around 130s. The starting time was extended in stages from 45 seconds to 90 seconds in order to improve start-up characteristics. At 90 seconds the start-up behaviour was good enough not to warrant any further extension thereof.

6.2.1 The fluid coupling.

In the drive arrangement of this conveyor, the fluid couplings are the only method of acceleration control and therefore had to be selected and modelled with some care. Fluid couplings are used extensively in mechanical drive systems to accelerate loads with high inertia. The fluid coupling transmits torque generated by the induction motor to the drive pulley through the reducer. The amount of torque transmitted depends directly on the volume of oil in the coupling and the speed of the output shaft relative to that of the input shaft. In this instance drain type (differential flow) couplings were chosen due to their simplicity, cost effectiveness and their ability to control the belt start-up to any desired format. The successful application history of this type of coupling and the experience gained at other collieries was also taken into consideration.

Each coupling consists of a normal traction coupling surrounded by an oil-tight enclosure. The rotor is fitted with orifices in the periphery. The coupling is initially empty of oil. The drive motor is started under no load with a start-up torque limit imposed to control the power applied by the motors. This maintains start-up torque below the limit of the electric motors and achieves the required extended start-up time. Oil is pumped into the coupling from a separate reservoir to the rotor and controlled by a solenoid valve in an on-off fashion. Measuring motor power through a PLC effects control of the solenoid valve and the rate of oil flow, thus obtaining the pre-determined start-up time. As oil is pumped to the coupling and it fills, torque is transmitted to the output stator which steadily accelerates. Oil continuously drains from the coupling, but the coupling fill is determined by the higher flow entering the coupling. Once the system is up to speed the rate of oil discharge is equal to the intake, and the coupling is effectively full. In the case of the belt stopping, the system is set in such a way that there is a five-minute time delay before the belt can be re-started. This is to allow the oil in the coupling to drain completely prior to a re-start. The conveyor is therefore protected against aborted start-stop and restart situations which could damage the belt. Initially there was an imbalance in the motor power sharing which the supplier corrected, by installing balance tubes in the couplings. The existing drive arrangement power-shares to within 1%, as indicated by the ammeters in the motor control centre.

The original installation had a different starting cycle. The primary drive started first. This drive was equipped with a scoop controlled fluid coupling. The coupling scoop-tube was inserted at a predetermined rate to control the oil supply to the coupling, which, in turn regulated the primary drive torque build-up. The secondary drive, which was equipped with a fixed fill delay-fill fluid coupling, was driven by the belt through its gear reducer and coupling. At a pre-selected time delay of about seven seconds after the primary drive start-up, the secondary drive started and ran up to full speed. The delay fill coupling had an internal regulator that regulated the oil flow from storage to operating chambers inside the coupling. Oil flow in this coupling was sustained by centrifugal force. The difference between primary and secondary coupling characteristics and therefore the difference in torque build up gave rise to poor dynamic behaviour on start-up. On activation of the secondary drive the velocity of the belt changed very rapidly and a stress wave was initiated. experiments with time delays gave marginal improvements [Ref 1], but the overall performance of the original system remained unsatisfactory.

6.2.2 The induction motor.

Since fluid is only introduced into the couplings once the motors are running at speed, motor speed variation has minimal effect on the dynamics of the conveyor. A constant speed representation of the motors would be adequate for most start-up and shut-down simulations.

6.3 Controlled stopping of the belt.

When motor torque is lost due to trip out or power cut, the tension on the carry strand at the head will drop rapidly. A low tension wave then propagates towards the tail. In this case the drop in tension will quickly lead to slack belt in the valley. A method of avoiding the problem is to attach flywheels to the drives. These will then maintain driving torque for a time after a trip out and prevent the rapid drop in head-end belt tension. In this case flywheels alone were not enough to prevent slack belt in the dip and a brake had to be fitted near to the tail-end of the conveyor. When the motors stop, the brake is activated and raises the belt tension at the tail end of the carrying strand. This compensates for the loss in tension at the head-end, thereby raising the average tension and avoiding slack belt. Since the purpose of the brake is to compensate for the rapid loss of driving torque, its rate of application should ideally be similar to the rate of torque loss at the drives. For this reason a simple rapid acting brake was recommended in the CTO design proposal. Variable brake tension and time would in this instance not have been effective. By doing a dynamic simulation, a fine-tuned configuration was suggested which would optimise start-up and shut-down of the conveyor. The following were some of the findings:-

- Use 110kW motors with differential flow drain type couplings.
- 700Nm nominal torque set-point on the drives during start-up with a tolerance of around 5% if an on/off valve is used for coupling control.
- 50 kg.m^2 flywheels on the high speed side of each reducer together with a brake in the return strand capable of rapidly applying 20kN braking force at the belt line in the event of a motor trip out.

Figures 3 to 6 are results obtained through simulation [Ref. 2] and formed the basis of the selection of the flywheels and the brake. Results are an overview of the behaviour of the belt and tensions were not considered at a specific point. Special attention was however given to the behaviour of the belt in the dip.

Figure 3 illustrates the behaviour of the belt using 40kg.m^2 flywheels and no brake. Coasting time is around 33 seconds and severe slackening of the belt occurs in the dip.

Figure 4 illustrates the behaviour of the belt using 40kg.m^2 flywheels and a 20kN brake. Coasting time is around 25 seconds and slight slackening of the belt occurs in the dip. The tension ratio across the braking pulley remains within the friction limits except for two 'spikes' which is not considered to be a problem.

Figure 5 illustrates the behaviour of the belt using 50kg.m^2 flywheels and a 20kN brake. This was the recommended configuration and was installed. Coasting time is around 27 seconds and there is no slackening of the belt in the dip. Again, the tension ratio across the braking pulley is within the friction limits.

Figure 6 illustrates the behaviour of the belt using 100kg.m^2 flywheels and no brake. This simulation serves to illustrate the extent to which the flywheels would need to be enlarged in order to avoid the need for a brake.

7. TAKE-UP TENSION, WINCH OR GRAVITY?

The original operating philosophy employed by the mine was based on the philosophy developed for the shorter version of the conveyor, which was equipped with fabric belting and a winch take-up. The belt in this condition was subject to reasonably severe dynamics under starting and stopping, which limited the capacity of the system and influenced the control philosophy of the take-up winch.

Since we accept that dead mass is cheap, reliable, and low maintenance, a gravity type take-up was preferred even though a previous installation on the same conveyor had failed.

In order to remain objective, start-up and shut-down simulations were repeated, using a winch type take-up rather than a gravity take-up. The belt is initially pre-tensioned to 4 500kg by the winch. The winch is then locked and remains locked during the start-up phase. The controls were set to recognise a running tension of around 1 480kg and the tension in the system at the take-up falls naturally to its running value. This philosophy was applied and the results were very good. The starting of the conveyor with the locked winch take-up was simulated and is shown in figures 7 and 8. Again the behaviour of the belt was entirely satisfactory since the belt tensions during stopping were similar for a gravity or winch type take-up. Definite disadvantages of the winch were the fact that it needs to be PLC controlled, needs maintenance and depends on a signal from a load-cell for setting tensions.

After original installation in 1984/5, it was suggested that the winch should pay out in the event of a power trip. The purpose would be to release tension in the belt. On investigating this philosophy for the upgraded conveyor, simulations showed that the release of tension has the very undesirable effect of causing the belt to become slack in the dip. Slipping at the brake pulley also occurs due to the reduction in tension. Although peak belt tensions during the stopping phase are reduced, this is not considered significant since even with the winch locked, these tensions are less than the tension at the head pulley during normal running of the conveyor. The stopping behaviour was therefore predicted to be satisfactory without releasing tension in the belt.

The existing winch type take-up was retained and is operating satisfactorily.

8. IDLERS AND IDLER BANKING.

During commissioning of the completed conveyor, start-up proved to be difficult. On CTO recommendation, alternate carry idler sets were removed. This process had to be repeated with alternating idlers until they were all run in. This is not an acceptable practice since the problem is really the high rolling resistance of the idlers. Manufacturers should therefore be aware of the potential dangers of idlers with a high breakaway torque. The simulations allow the user to specify idlers, with breakaway force in accordance with the SABS 1313 specification.

Tensions in the horizontal curve near the tail were such that idler banking was not required, and the predicted drift of the belt was acceptable. In the curve near the head, tensions were, against expectations, not so much different that additional idler packing was required. In this particular curve though, 4-roll idlers were used in the days of the old fabric belt. If the belt then drifted out of its normal 3-roll trough, it went onto the fourth roll. These idlers were retained after the upgrade but proved to be a bonus safety item rather than a necessity.

9. THE BRAKE.

The original brake pulley was installed near the tail of the conveyor. The drum brakes attached to the brake-pulley shaft were electric-solenoid-released and spring-applied failing to safety. Variation of brake torque could be obtained by adjusting the brake-shoe travel. This brake was however never really successful. After a period of time the mine stopped using the brake and little is known regarding the operation of the belt during the period when no brake was used.

On redesigning the system, the client indicated that a brake was an undesirable item, because of previous bad experience. However, through dynamic simulation we proved beyond any doubt that a brake would enhance the stopping characteristics of the conveyor and the client agreed to re-commission the brake accordingly. Applying the brake gently was considered but as is shown in the dynamic simulation, this option was not desirable.

The brake pulley has a diameter of 630mm. In order to achieve 20kN brake force, the torque was calculated to be 6,3kNm. A disk brake of ϕ750mm was selected and using only one caliper, the brake force was calculated at 20,323kN. The final installation had a ϕ1000mm disc.

A fail-safe, hydraulic disk brake system, a new product in the country in 1995, was installed and proved to be successful. The brake is spring applied and hydraulically released. Therefore, in the event of a power trip, the hydraulic pressure will drop and the brake will apply itself instantly. An electric pump motor supplies oil and pressure to the system. As soon as a pre-set pressure of 75 bar is reached, the pressure switch disconnects the supply to the motor and hydraulic pressure is retained. If the pressure should fall to a minimum setting of around 62 bar, the pressure switch will re-connect the supply to the motor and boost the pressure back to the pre-set value.

When the belt is started and the brake motor is energised, the brake releases. If not, the power pack start sequence will not proceed. When the conveyor is running and the "brakes off" fails to indicate that the brake is in fact off, the power packs will trip in order to stop the belt and safeguard against possible damage.

The same rules apply as in 1984/5, when the belt was first commissioned. Poor maintenance of the brake may result in binding during normal operation, or excessive braking torque under stopping. This in turn will result in the initiation of destructive dynamic stresses at the tail end of the belt during the stopping cycle. [Ref 1]

10. 1989 – OBSERVATIONS AND RECOMMENDATIONS.

Mr. Dreyer made certain observations and recommendations in 1989, to improve the operation of the original conveyor.

a. Observation (1989):- Dynamic stresses in the conveyor were found to be initiated both during starting and stopping from the rate of applying and removing the conveyor driving power.

Findings in 1997:- By increasing the start-up time and adding the flywheels at the drives, the dynamic stresses induced by the rate of applying and removing power at the drives were eliminated in the new conveyor.

b. Observation (1989):- The brake has to be maintained and care should be taken to ensure that brakes are correctly adjusted at all times to operate effectively without inducing excessive belt stress at the tail end of the conveyor.

Findings in 1997:- The behaviour of the belt under stopping depends largely on the effective operation of the brake. Not only was this the case for the old installation but also applies to the new installation.

c. Observation (1989):- Releasing slack into the belt take-up during stopping proved to be partly successful in absorbing the shock wave initiated by the sudden removal of driving power.

Findings in 1997:- This applied to the old installation but is no longer necessary since the addition of the flywheels mainly removed the shock wave previously initiated by the sudden removal of driving power. Releasing slack also has the undesirable effect of inducing slack belt in the valley.

d. Observation (1989):- The take-up winch on the conveyor must be maintained to provide pre-tension before the belt start-up sequence is initiated. Once the belt is up to full speed, the take-up tension can be released to a running tension.

Findings in 1997:- This philosophy was successfully applied on the new installation, albeit after certain modifications. The winch is pre-tensioned and locked in position prior to start-up. After start-up, the tension will fall automatically to the running tension.

11. AUDIT/TESTING.

After the actual re-commissioning of B18 the conveyor was tested under normal operating conditions. The method of testing was fairly simple. The belt was loaded to its design capacity, ran, and stopped while fully loaded. The belt was then restarted. This was repeated several times, observing the belt from different points and (in the control room) measuring current drawn during start-up (see figure 9). Results obtained from the control room were assessed and compared with the theoretical design. This confirmed existing theories and methods. During the entire testing exercise start-up was exceptionally smooth and no dynamics were apparent. The conveyor behaved in a manner that satisfied both the client and the design team.

12. CONCLUSION.

As part of the development envisaged by the client, the overland conveyor, with its curves, was designed and commissioned. From the dynamic analysis, certain changes were made in order to improve the start-up and shutdown characteristics. Additional inertia was added in the form of flywheels at the drives and a brake was installed to assist in managing the belt in the dip under stopping. Since commissioning in 1996, the conveyor has been operating successfully and has conveyed around 2,5 million tons of coal.

Although designing curved overland conveyors is no longer new to the CTO, conveyor B18 offered certain challenges. It is now obvious that in the case of a conveyor with a profile such as this, steady state design is not enough. Special attention should be given to the design in the form of dynamic analysis. We have learnt that the use of a winch type take-up can have some benefits, that torque control during start-up is essential and in this case that controlled stopping protects the entire system. If in future we were presented with the opportunity to design more of this type of conveyors, we would approach the design in a similar manner.

13. ACKNOWLEDGEMENTS.

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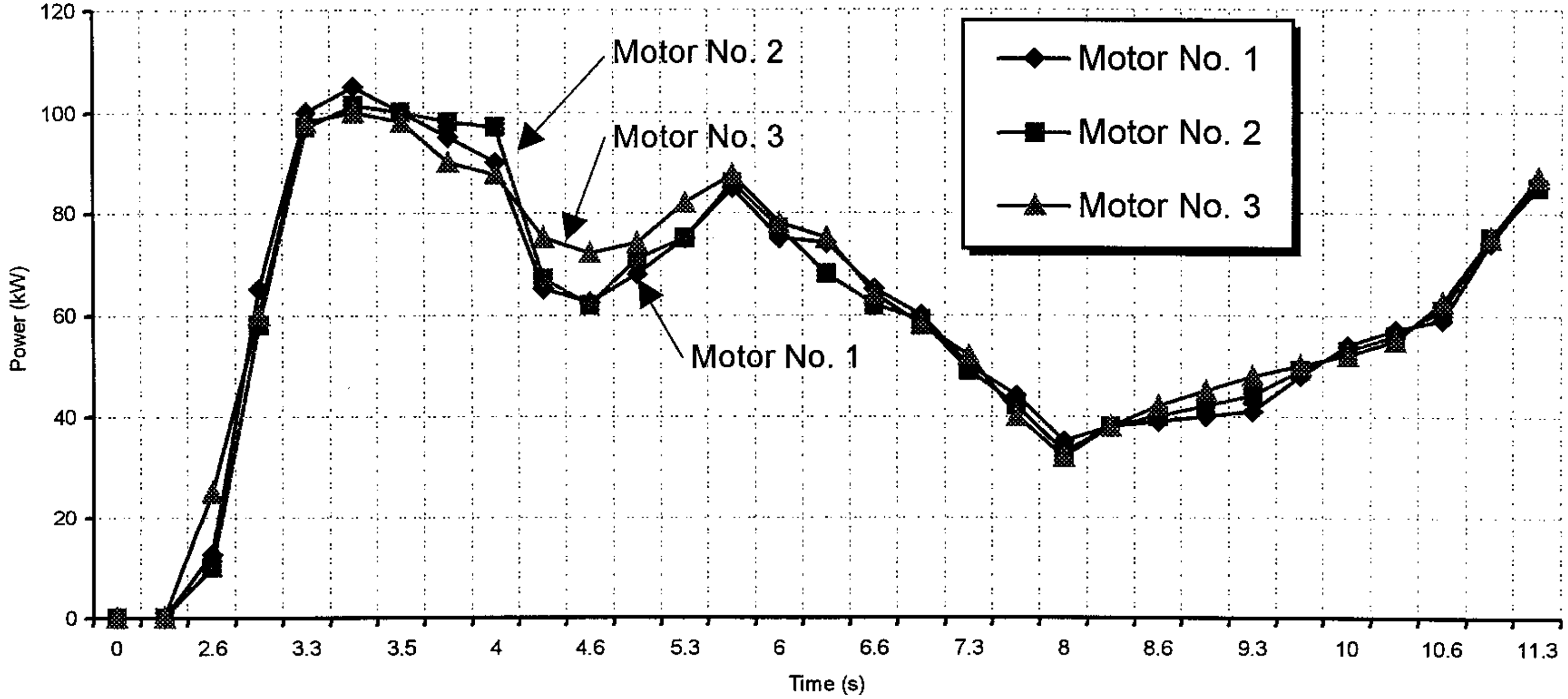
14. REFERENCES.

1. Dreyer H., Solutions for dynamic stresses in a catenary profile overland conveyor, Beltcon 5 1989.

2. Hamilton R., Goedehoop #4, B18 overland conveyor (Model:B18.CSL),1995-09-07

3. Page J.L. et al, Design of a long overland conveyor with horizontal curves, Beltcon 6 1991.

4. Funke H, Longitudinal Vibrations during transient operating conditions of belt conveyors. Beltcon 4 1987.



Conveyor B18 - Start-up

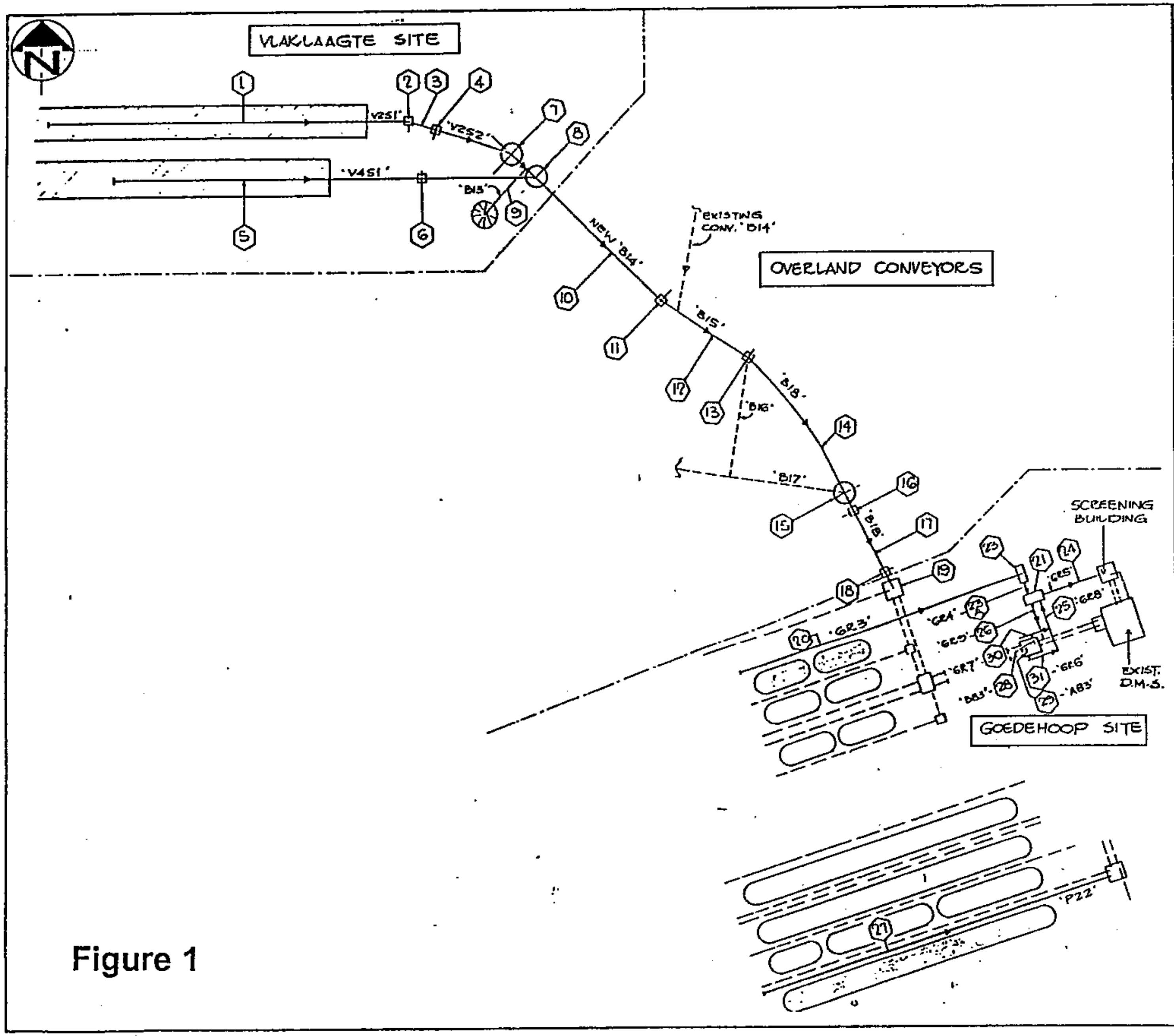
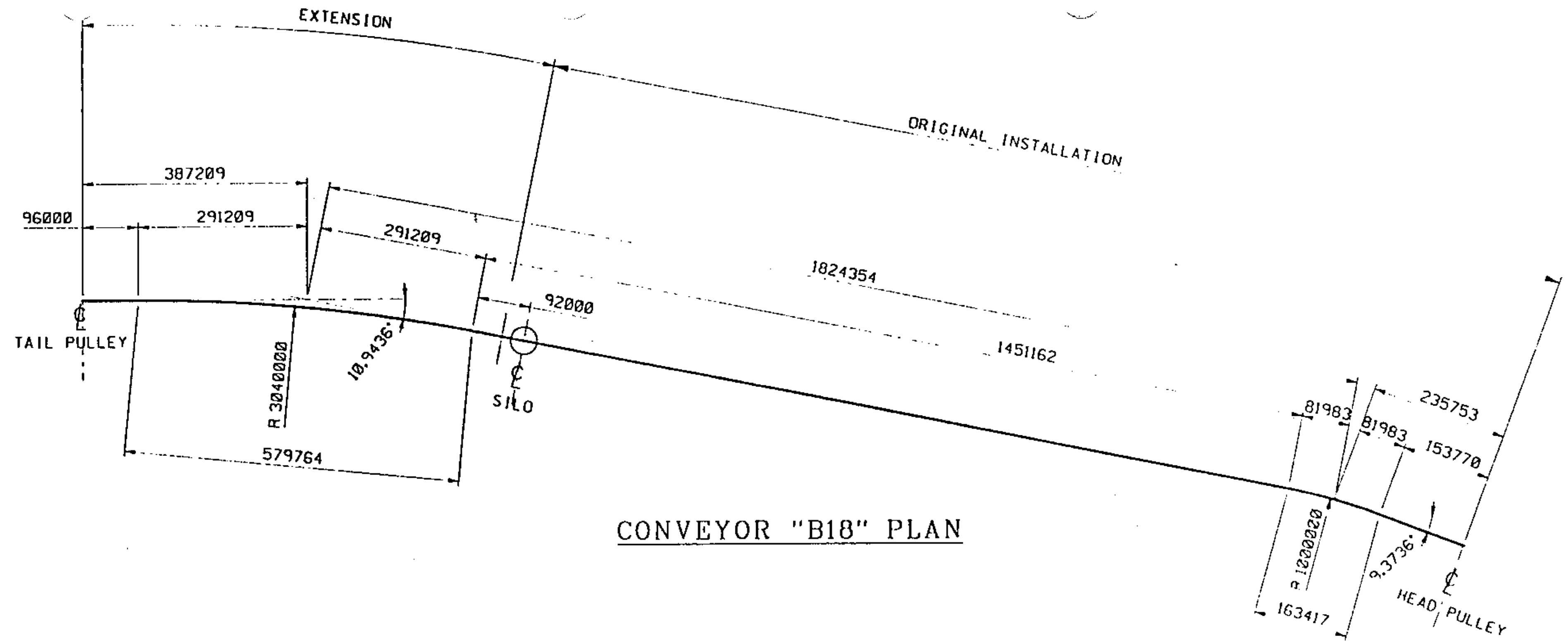
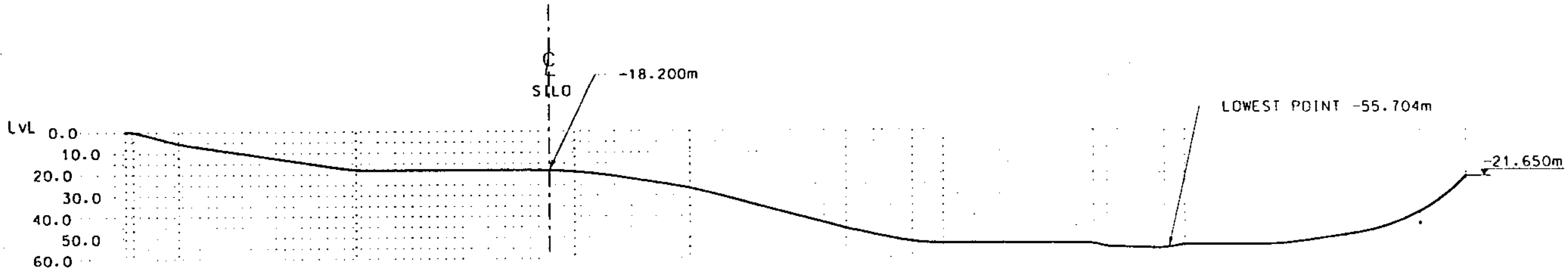


Figure 1

ITEM Nº:	EQUIP. Nº:	CONV. WIDTH	DESCRIPTION	SCOPE OF WORK
1	V2S1	1200	Nº2 SEAM SHAFT CONV.	NEW
2	V2S1	1200	CONVEYOR DRIVE AND TRANSFER HOUSE	NEW
3	V2S2	1200	SILO FEED CONVEYOR	NEW
4	V2S2	1200	CONV. DRIVE HOUSE	NEW
5	V4S1	1200	Nº4 SEAM SHAFT CONV.	NEW
6	V4S1	1200	CONV. DRIVE HOUSE	NEW
7	B06	—	6400T SILO (2 SEAM)	NEW
7A	B07	2100	BELT FEEDER	NEW
8	B08	—	6400T SILO (4 SEAM)	NEW
8A	B09	2100	BELT FEEDER	NEW
9	B13	1050	CONV. FOR TRUCK LOADOUT FASCILITY	NEW
9A	—	—	PROPORTIONING CHUTE	NEW
10	'B14'	1050	OVERLAND CONVEYOR	NEW
11	'B14'	1050	CONV. DRIVE & TRANSFER HSE.	NEW
12	B15	1050	EXISTING OVERLAND CONV.	NEW EXTENSION TO TAIL END
13	B15	1050	CONV. DRIVE & TRANSFER HSE.	MODS TO DRIVE AND NEW HEAD CHUTE
14	B18	1200	EXISTING CONVEYOR	NEW OVERLAND TAIL SECTION
15	B04	—	1000T SILO	EXISTING
16	—	—	AUDIT BIN	EXISTING.
17	B18	1200	CONVEYOR	EXISTING SECTION
18	B18	1200	CONVEYOR DRIVE HOUSE	EXISTING - MODS. TO DRIVE
19	—	—	CRUSHER HOUSE	EXISTING
20	GR3	1050	RECLAIM CONVEYOR	NEW
21	—	—	TRANSFER HOUSE	NEW
~~22~~	~~GR4~~	~~1050~~	~~CONVEYOR~~ NOT REQD	~~NEW~~
23	C82	—	DMS FEED BIN - 100T	NEW
23A	GR4	1500	CONVEYOR	NEW
24	GR5	1050	CONVEYOR	NEW
25	GR8	1050	CONVEYOR	NEW
26	GR9	1050	CONVEYOR	NEW
27	P22	1500	RECLAIM CONVEYOR	NEW
28	BB3	1350	BELT FEEDER	EXISTING - TO BE REVERSIBLE
29	AB3	1350	BELT FEEDER	EXISTING - TO BE REVERSIBLE
30	GR7	1200	CONVEYOR	NEW
31	GR6	1200	CONVEYOR	NEW



CONVEYOR "B18" PLAN



CONVEYOR "B18" PROFILE

Figure 2

Conveyor	:	Goedehoop #4, B18 overland conveyor (Model: B18A.CSL)
Features	:	487TPE couplings, 6553 kg gravity take-up, 40 kg.m^2 drive inertias, HP holdback
Simulation	:	Power cut with full belt at 10 sec, no brake

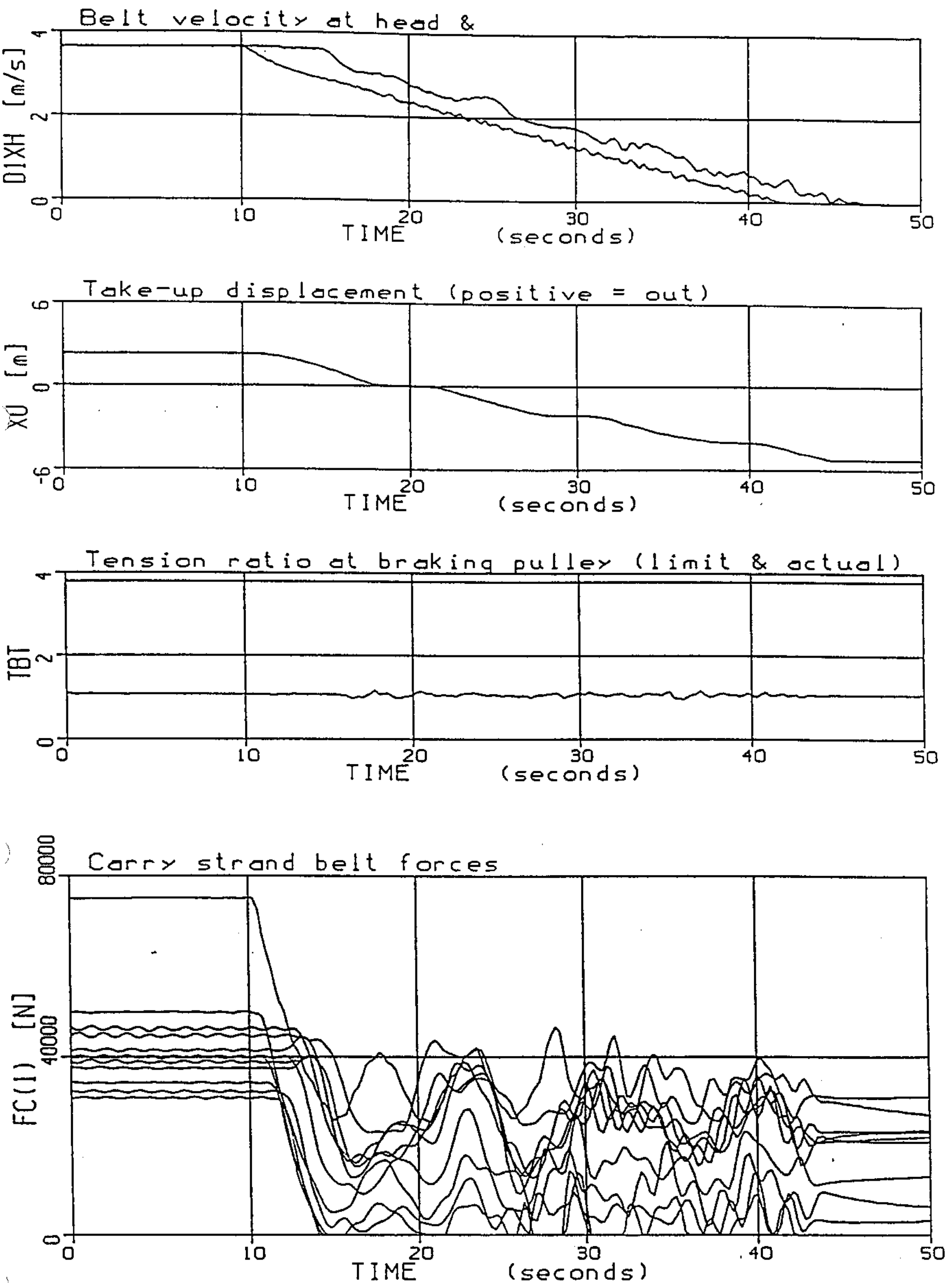


Figure 3

Conveyor	:	Goedehoop #4, B18 overland conveyor (Model: B18A.CSL)
Features	:	487TPE couplings, 6553 kg gravity take-up, 40 kg.m^2 drive inertias, HP holdback
Simulation	:	Power cut with full belt at 10 sec, return strand brake giving 20 kN on belt-line

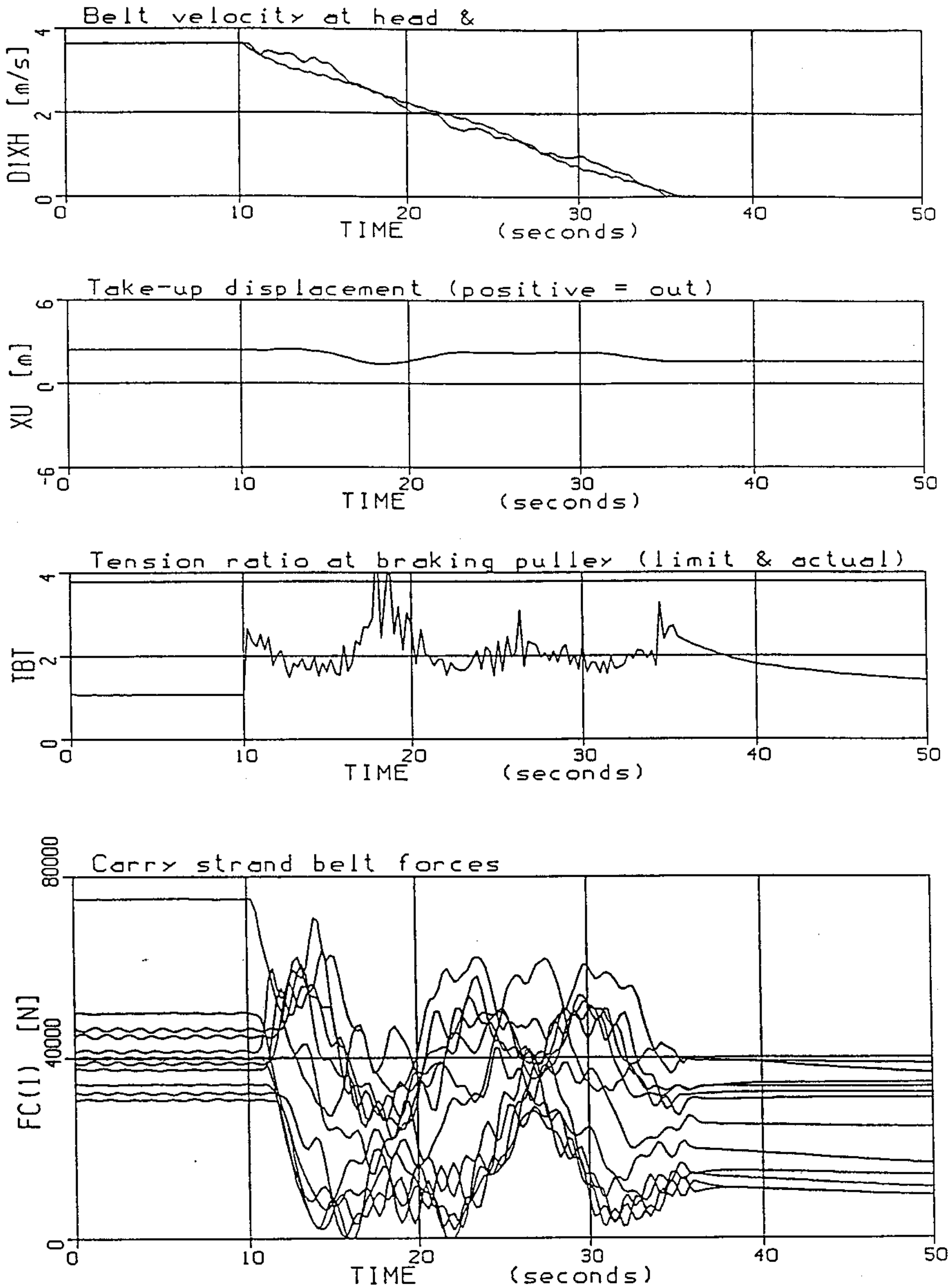


Figure 4

Conveyor	:	Goedehoop #4, B18 overland conveyor (Model: B18A.CSL)
Features	:	487TPE couplings, 6553 kg gravity take-up, 50 kg.m^2 drive inertias, HP holdback
Simulation	:	Power cut with full belt at 10 sec, return strand brake giving 20 kN on belt-line

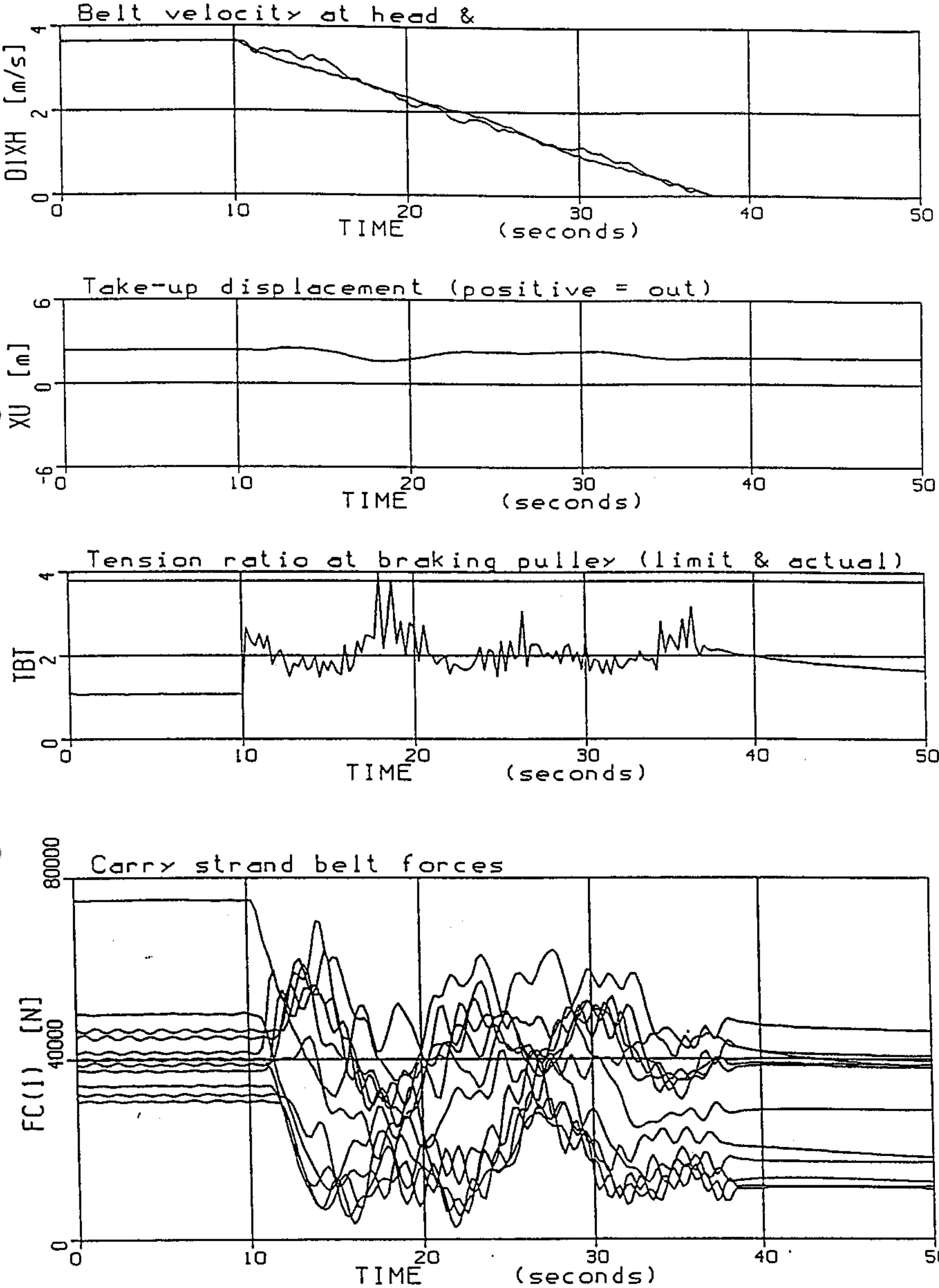


Figure 5

Conveyor	:	Goedehoop #4, B18 overland conveyor (Model: B18A.CSL)
Features	:	487TPE couplings, 6553 kg gravity take-up, 100 kg.m^2 drive inertias, HP holdback
Simulation	:	Power cut with full belt at 10 sec, no brake

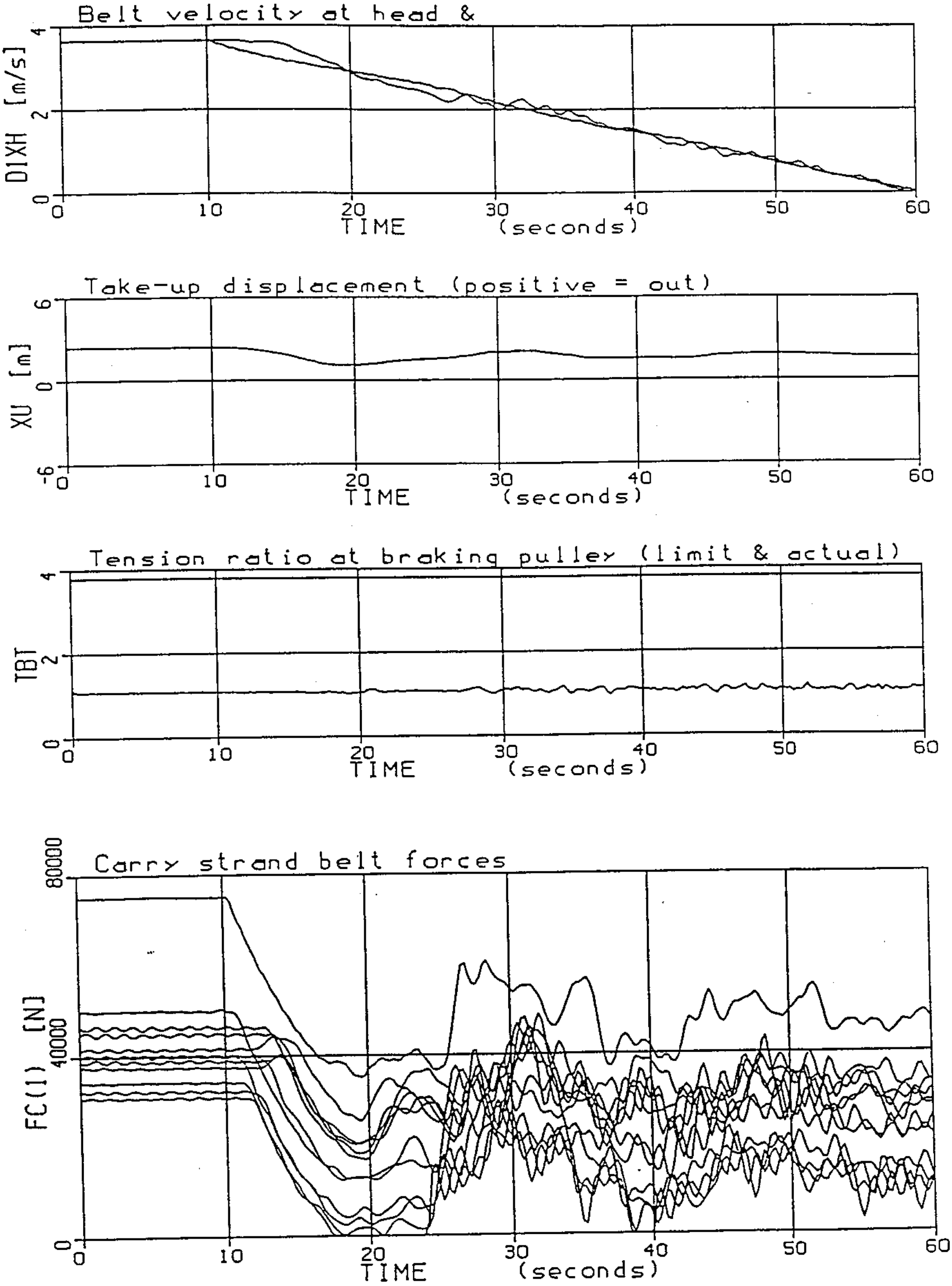


Figure 6

Conveyor	:	Goedehoop #4, B18 overland conveyor (Model: B18A.CSL)
Features	:	487TPE couplings, winch take-up, 50 kg.m² drive inertias, HP holdback
Simulation	:	Full belt start-up with winch locked after pre-tension of 45 kN

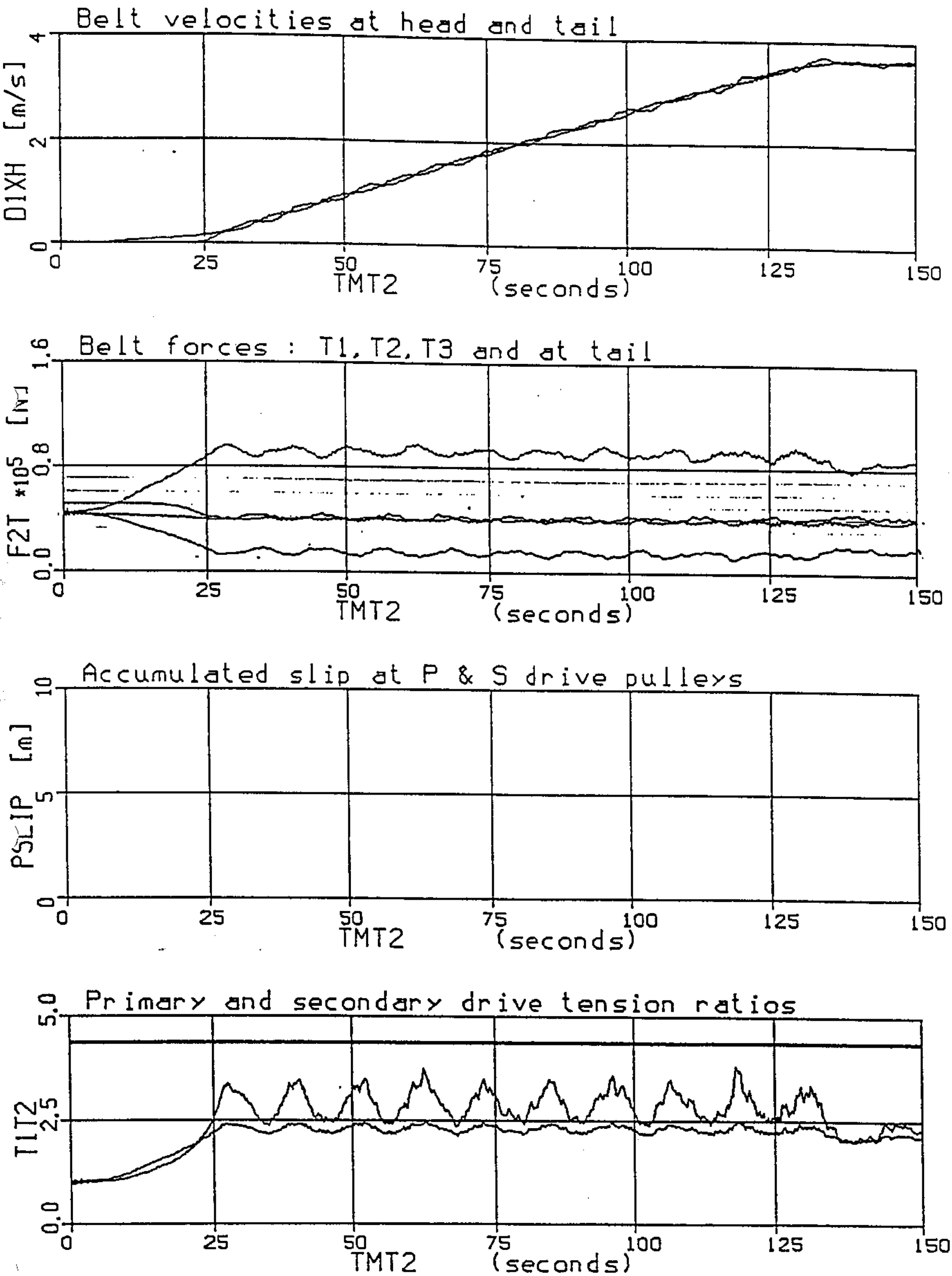


Figure 7

Conveyor	:	Goedehoop #4, B18 overland conveyor (Model: B18A.CSL)
Features	:	487TPE couplings, winch take-up, 50 kg.m² drive inertias, HP holdback
Simulation	:	Full belt start-up with winch locked after pre-tension of 45 kN

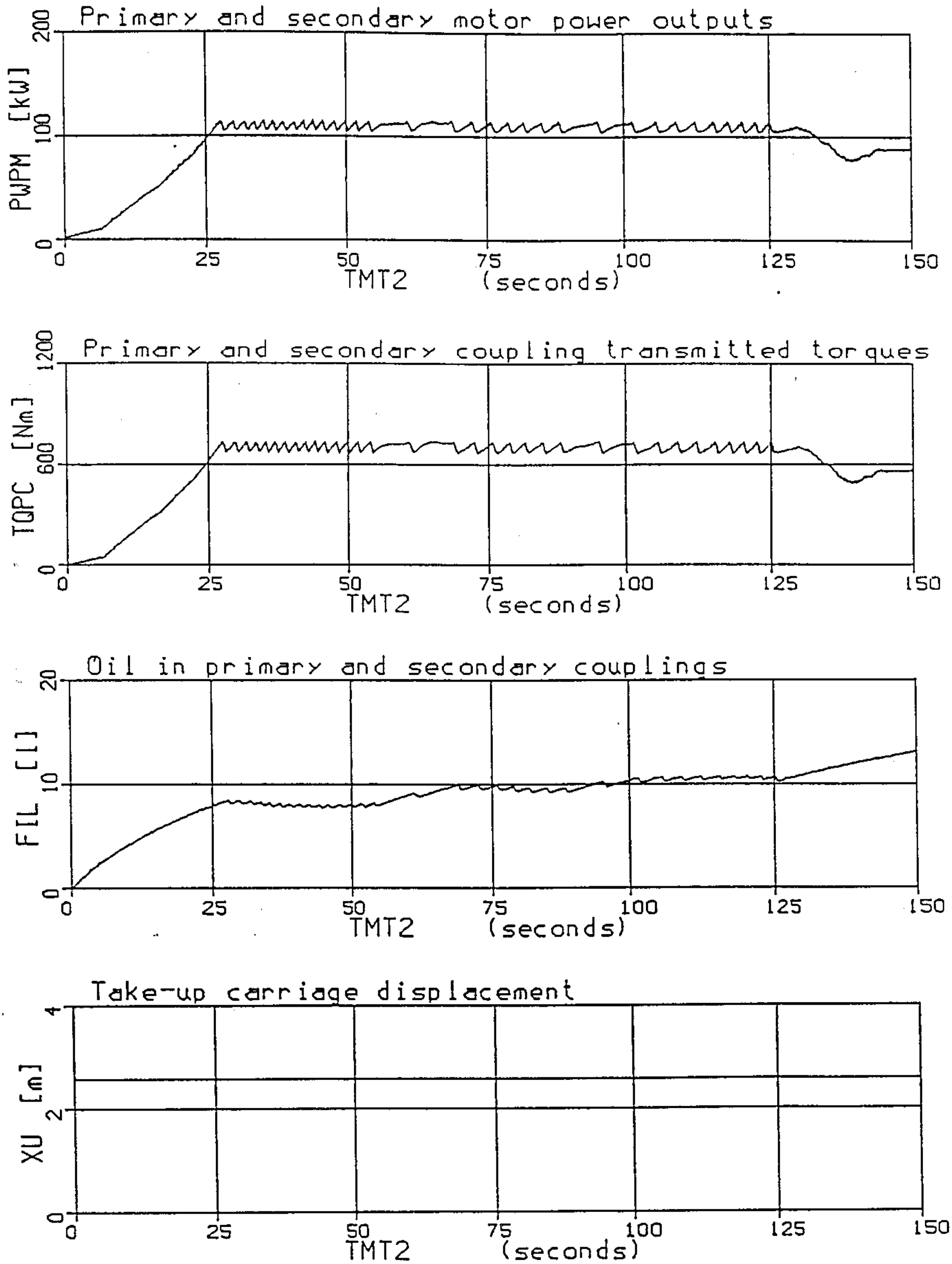
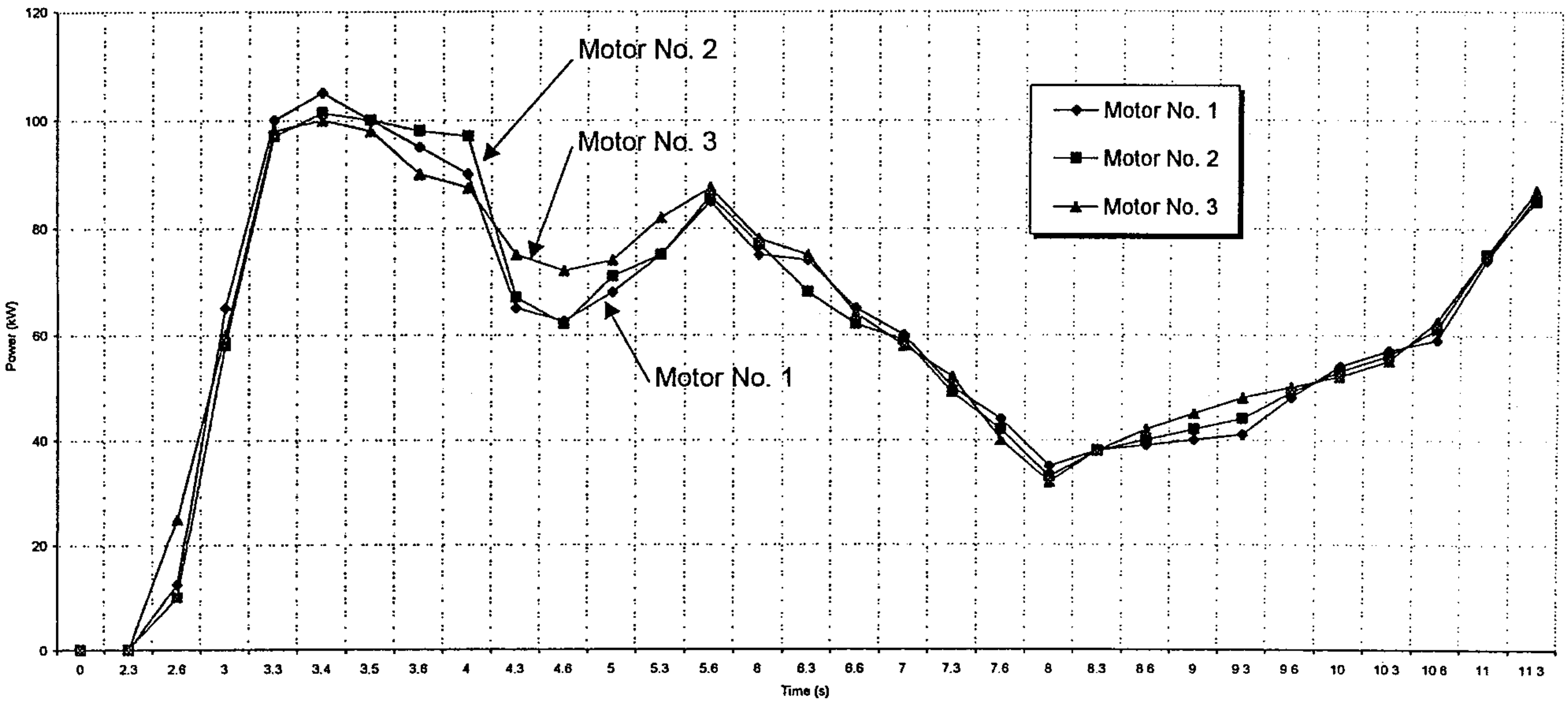


Figure 8



Conveyor B18 - Start-up

Figure 9

Paul Nel

The author comes from a structural background and has been involved with materials handling projects for the last ten years. He holds an Advanced Diploma in Materials Handling from the South African Institute of Materials Handling and is also a professional member of the same organization.

He spent the last 3½ years with the Anglo American Corporation and currently holds a position of Chief Designer – Materials Handling, within the Mechanical Engineering Department of the AAC Central Technical Office.

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Member of the South African Institute of Draughtsmen