CONSTANT SPEED VERSUS VARIABLE SPEED OPERATION FOR BELT CONVEYOR SYSTEMS

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SUMMARY

Variable speed control for the coal handling conveyors intended for the new Medupi power station have been evaluated in comparison to constant speed operation. This evaluation is based on the assumption that ISO 5048 applies to the design of the conveyor belts operating over the required capacity range for both variable speed control as well as for variable loading of the conveyor (constant speed).

The artificial friction factor 'f' used by ISO 5048 has been calculated according to the procedure explained by the author Ishwar G. Mulani in his publication titled 'Engineering Science and Application Design for Belt Conveyors' for the specific belt speed and load conditions evaluated. This procedure enables the estimation of the individual friction components that make up the artificial friction factor 'f' i.e. bearing rotational, belt indentation, belt bending and the material flexure friction components.

The conveyors are designed for the maximum required capacity but operate normally at a relatively much lower capacity. The reason for this capacity range is as result of ranging coal qualities, boiler unit load factor and demand side implications. From the evaluation of this specific application for the specific capacity range, it is shown that the variable speed control option is favoured in terms of lower energy consumption, less wear at loading areas and the expected improved operating behaviour as a result of better belt alignment due to the optimum loading ratio.

1. INTRODUCTION

Eskom has opted for variable speed control on the belt conveyor systems for the proposed new capacity expansion program on fossil fired Power Generating Stations. There are different opinions in industry regarding the potential advantages and disadvantages when comparing variable speed control to constant speed operation of belt conveyor systems. This paper presents the evaluation process regarding the options of variable speed control in comparison to constant speed operation of belt conveyors from a technical point of view.

The evaluation is for a specific application: transporting coal in the electrical Power Generating Industry for the specific capacity range and component selection from the perspective of the end user. In the case of the Eskom plants the design capacity is based on the full load condition of the boiler units including backlog recovery capability or capacity loss recovery ability as well as the impact on the system availability and the coal quality variation. The normal operating capacity of the system is at as low capacity as can be expected. The focus is thus on high availability and reliability of the system to ensure that production and plant performance targets can be achieved within the operational realities.

The following aspects are considered:

- The belt artificial friction coefficient focusing on the belt indentation resistance at idlers as the main part of the resistance to motion.
- The load versus life implication on rotating components like idlers and pulleys.
- The relative wear implication at load points, skirting zones, tilted idlers and belt cleaners.
- The energy consumption for the operation of the conveyor.

Other technical areas that are discussed include:

- The operation of the gearboxes under reduced speed conditions.
- The design challenge regarding transfer stations in the case of variable speed drives.
- The efficiency and reliability of the electrical variable speed drive system.

2. MOTION RESISTANCE COMPONENTS OF BELT CONVEYORS

The energy consumption of a belt conveyor system is determined by the resistance to motion of the belt system along the carry and return sides of the belt. The total resistance to motion is the algebraic summation of all the resistances. The resistance forces can broadly be classified into 3 categories.

- **Slope or gravity resistance**
  The relative height change from the feed point to the discharge will generate gravity resistance by lifting or lowering of the material that is conveyed. This resistance can therefore be power requiring or regenerative in terms of its resistance component.

- **Load point material acceleration resistance**
  The material velocity component onto the receiving belt is usually different to the belt speed. This results in an inertial resistance at the load point.

- **Frictional resistance**
  This resistance component includes rotational, sliding and internal resistances as a result of the belt/material interaction.

  Many components (idlers and pulleys) rotate along the conveyor length; frictional rotational resistance is generated by these components.

  Sliding friction is introduced at belt cleaners, skirting, tilted idlers etc. as result of belt or material sliding motion.

  The belt sags between idler sets and rises up on approaching idlers, the trough shape also opens out in between idler sets and closes on approaching idlers. This motion also applies to the material load stream and is expressed by the material flexure resistance.

  The belts flexure resistance originates from the motion interface at idler sets expressed as belt bending resistance with the belt sagging motion in between idler sets. Tthe deformation or denting of the belt contact zone at each idler roll results in belt indentation rolling resistance as result of the time related visco-elastic properties of the rubber cover of the belt (Figure 1). In many long conveyors the energy consumption is mainly contributed by the work done to overcome indentation rolling resistance.

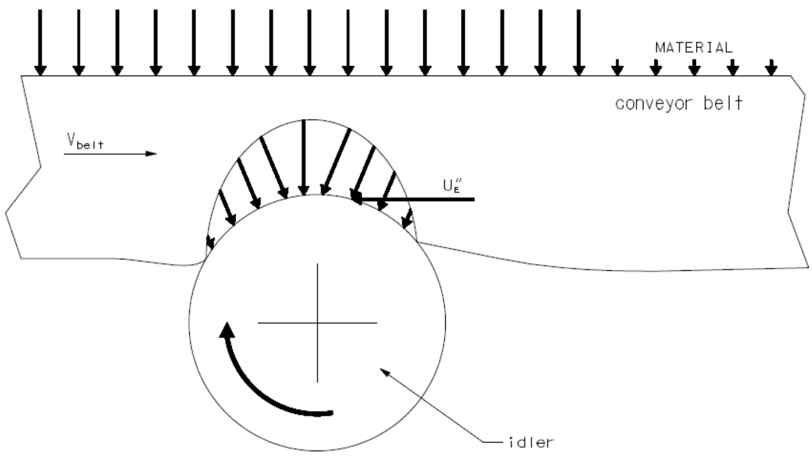


Figure 1: Belt indentation resistance at the idler roll

2.1. MAIN RESISTANCE

This is the most significant resistance on the belt conveyor. The main resistance is dominant for long horizontal conveyors and in the case of inclined conveyors the slope resistance may be the dominant component. The main resistance is encountered as result of the resistance to motion while transporting material on the belt supported on the idlers. This resistance includes the following friction components:

- Idler roll rotational friction
- Material flexure resistance as result of belt sag, and
- Belt flexure resistance comprising of the belt bending resistance due to belt sag and the belt indentation resistance due to belt cover deformation at the idler roll contact zone.

These three friction components are the basis for the artificial coefficient of friction 'f' as used in the ISO 5048 conveyor design standard.

2.2. THE ARTIFICIAL FRICTION COEFFICIENT 'f'

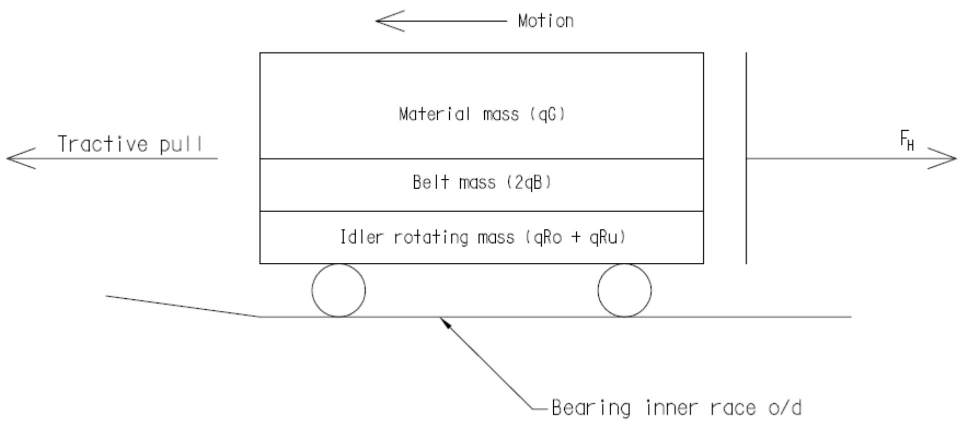
As discussed, the artificial coefficient applies to the combination of the following resistances that opposes belt motion:

- Idler rotational resistance,
- Material flexure resistance, and
- Belt flexure resistance that includes belt indentation at idlers and belt bending as result of sag.

These resistances are directly related to the following motion masses expressed per metre of conveyor length:

- Idler rotating mass,
- Belt mass, and
- Material mass.

The motion masses have a cumulative relation to the frictional resistances. The idler bearing supports the mass of the rotational parts, the belt mass and the material supported on the belt as illustrated in Figure 2. The idler rotating resistance is therefore proportional to the sum total of the three masses mentioned.



$$M = qG + 2qB + qRo + qRu$$

$$F_H = fM$$

Figure 2: Schematic of the mass model for a conveyor belt

The belt flexure is affected by the belt's own weight and the material weight supported on the belt. Belt flexure resistance is therefore proportional to the weight of the belt and the material on the belt.

The material flexure resistance in turn is only affected by the material self-weight, as it is on top of the moving masses.

With this in mind there could be three different friction coefficients applicable to the three different values of mass. In ISO 5048 this is calculated as a single entity to avoid complexity. This common equivalent artificial / fictitious resistance coefficient 'f' applies to the total moving mass force to calculate the main resistance F_H on basis of Coulomb's law of friction.

The main resistance $F_H = f L g[q_{Ro}+q_{RU} +(q_G +2q_B)\cos\delta]$

2.3. THE TOTAL BELT RESISTANCE ACCORDING TO ISO 5048 'F_U'

The total resistance of the conveyor belt includes in addition to the Main or Primary resistance (F_H) also the Secondary resistances (F_N), the Slope resistance (F_{St}) and the Special resistances (F_S).

The Secondary resistances (F_N) are friction and inertia based and occur at specific parts of the belt conveyor.

The Secondary resistances include:

- Material acceleration resistance at the load point,
- Material sliding resistance along the chute at the load point,
- Belt cleaner resistance, and
- Wrap and bearing resistances at the pulleys.

The Secondary resistances are independent of the length of the conveyor and are constant. The significance of the Secondary resistances relative to the primary motion resistance declines in the case of longer belts. A general assumption is therefore permissible; the total sum of the Secondary resistances is therefore being accounted for by means of a length based coefficient C in ISO 5048.

Special resistances (F_S) do not occur on all belt conveyors. These are resistances as result of:

- Idler tilt relative to the belt,
- Sliding friction at chute skirting if present over part or the full length,
- Belt cleaner resistance,
- Belt turn-over resistance,
- Discharge plough resistance, and
- Tripper resistance.

Slope resistance (F_{St}) is the resistance introduced as result of the lifting or lowering of material on sloped conveyors.

The total belt resistance $\mathbf{F_U} = F_H + F_N + F_{St} + F_S$

3. CALCULATION OF THE ARTIFICIAL FRICTION COEFFICIENT 'f' FOR VARYING MATERIAL LOAD CONDITIONS OR VARYING BELT SPEED

3.1. IDLER ROTATIONAL FRICTION FACTOR

The rotating portion of an idler is normally supported on ball bearings. Seals protect the bearings and the space within the seals and bearings are filled with grease for lubrication as well as added sealing effect.

The idler while rotating encounters resistance from:

- Bearing friction resistance,
- Misalignment of idlers relative to the belt line, and
- Resistance as result of the grease that is in contact with stationary and moving components.

The Rotational resistance due to the bearing frictional resistance as illustrated in Figure 3 is as a result of the radial load ‘R’ onto the bearing and the bearing friction coefficient μ. The radial load is proportional to the loading at the idler as result of the belt and material on the belt. The force required to overcome idler bearing resistance ‘F_1’ is therefore F_1= (μ◌R)◌d / D with d= bearing bore diameter and D= idler roll diameter. The typical value for the coefficient of friction μ=0.0015 for ball bearings in the 6200 / 6300 series range.
Thus F_1=0.0015 (d / D) R.

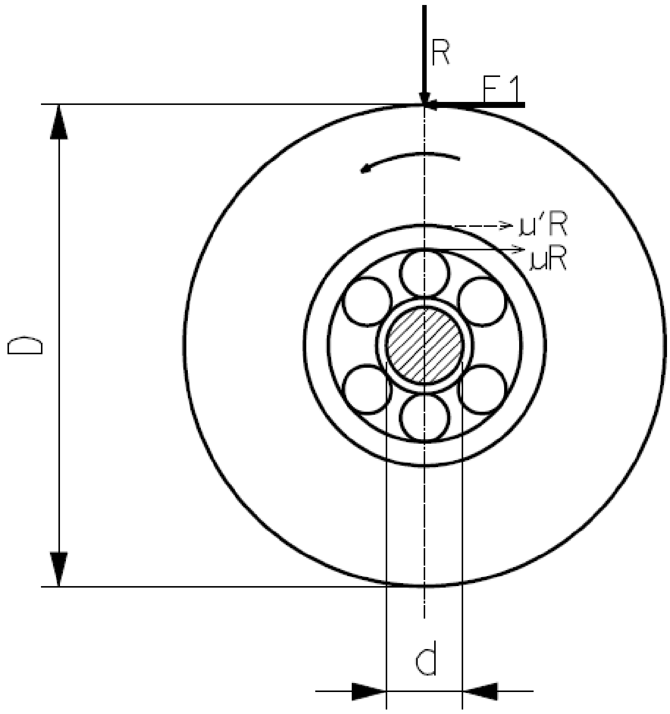


Figure 3: Illustration of the friction force applicable to an idler roll

In the case of misaligned idler rolls, the equivalent sliding friction coefficient=(μ_0)(off-set gradient)=(μ_0)(sinθ). The value for μ_0 is in the range of 0.3 to 0.4 for a rubber belt onto a steel idler roll. The equivalent sliding friction coefficient as result of idler misalignment at an assumed rate of 3 mm for 1000 mm length of idler base yields a value of 0.4x3/1000=0.0012. It is assumed that idler frames will be installed within this offset gradient range.

The typical values for seal resistance range from 1 N to 4 N per roll and can be obtained from the idler manufacturers.

For the purpose of evaluation the load onto the idler rolls are considered on basis of the load condition. The mentioned parameters were assumed to be constant for the evaluations.

3.2. CONSTITUENTS OF THE ARTIFICIAL FRICTION COEFFICIENT ‘*f*’

According to Ishwar G Mulani in the publication ‘Engineering Science and Application Design for Belt Conveyors’ the main resistance F_H is calculated as follows:

F_H = Roller Resistance + $(f_{dc} + f_{bc} + f_m).(q_G + q_B).g + (f_{dr} + f_{br}).q_B.g$

With the following friction coefficients:
- f_{dc} = belt denting flexure carry side
- f_{bc} = belt bending flexure carry side
- f_m = material flexure carry side
- f_{dr} = belt denting flexure return side
- f_{br} = belt bending flexure return side

Development of a relationship between the carry and return strands of the conveyor yields the following:

$f_{dc} = f_{dr} .a_o^{(1/3)}/a_u^{(1/3)}.((q_G +q_B)/q_B)^{(1/3)}$

With: a_o = idler pitch on the carry side
a_u = idler pitch on the return side

$f_{br} = f_{bc}$.(average belt sag return side / average belt sag carry side)

$f_{dr} = f_{dc}.1/(a_o^{(1/3)}/a_u^{(1/3)}.((q_G +q_B)/q_B)^{(1/3)})$

The ISO artificial friction coefficient f can then be expressed as a non-dimensional unit on the basis of the combined effect of the individual components by multiplying each of the above friction coefficient components (f_{dc}, f_{bc}, f_m, f_{dr}, f_{br}) with the mass component applicable to the specific coefficient and expressing it in terms of the total mass in motion per linear metre on the conveyor.

The typical values for the individual coefficient components at belt sag limited to 1% are:
f_{dc} = 0.012,
f_{bc} = 0.0033 for steel cord belting and 0.0066 for fabric belting, and
f_m = 0.008

According to the calculation procedure of the artificial friction coefficient f adjustments are required in the specific coefficient component affected in terms of:

- The belt width for width less than 800 mm,
- Belt cover softness,
- Operational conditions,
- The material flexure characteristic,
- The idler trough angle greater than 35˚,
- The idler roll diameter less than 108 mm,
- Belt speeds slower than 3.75 m/s,
- Belt alignment,
- Belt sag greater than 1% or less than 0.66%, and
- Temperature adjustment for temperature less than 0˚C

This approach is used as a basis for the calculation of the required driving force at the drive pulley of the conveyors evaluated on basis of the ISO 5048 procedure. The belt tensions and loads induced onto the system components are then calculated from this for design and comparison purposes.

Performance measurements obtained from tests conducted on the existing coal overland conveyor at Matimba Power Station were used for comparison with the calculated values derived on basis of the ISO 5048 procedure using the calculated artificial friction coefficient f on the basis of the method explained above (Figure 4).

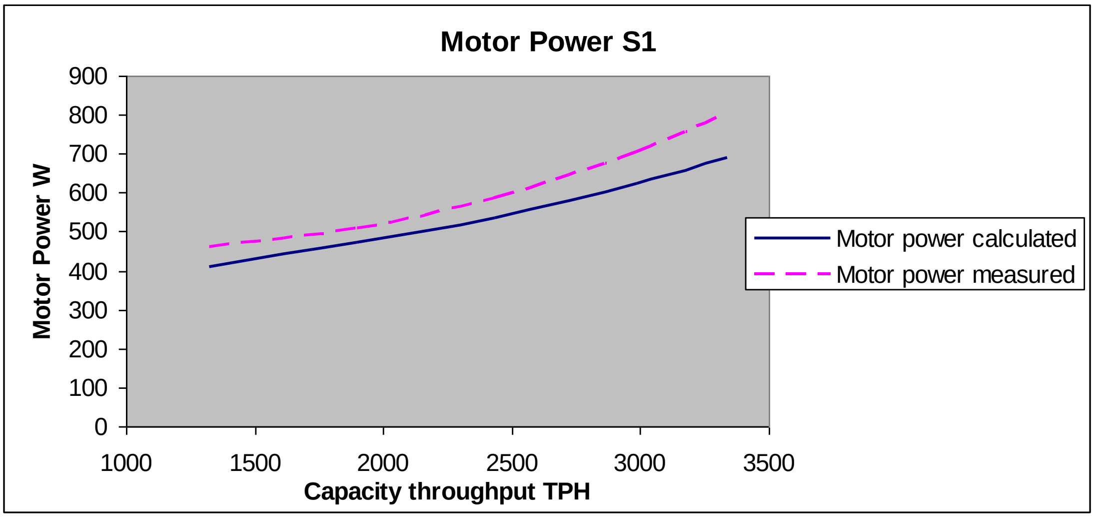


Figure 4: Comparison of measured and calculated power requirements for Matimba S1 conveyor

The correlation between the results from the performance measurement and the calculated values is acceptable for the Matimba S1 conveyor. The calculated values are lower than the measured values, the most likely reason for this is the inaccuracy in the mass measurement while performing the performance measurements.

On the basis of this finding it was decided to assume that the calculation of the ISO 5048 artificial friction coefficient is applicable to the range of conveyors evaluated for the Medupi application. Performance measurements will be undertaken on all of the Medupi conveyors upon installation and commissioning to verify the correctness of this assumption.

The calculations of the artificial friction coefficient for the Medupi conveyors are presented in graphical format for the constant and variable speed options.

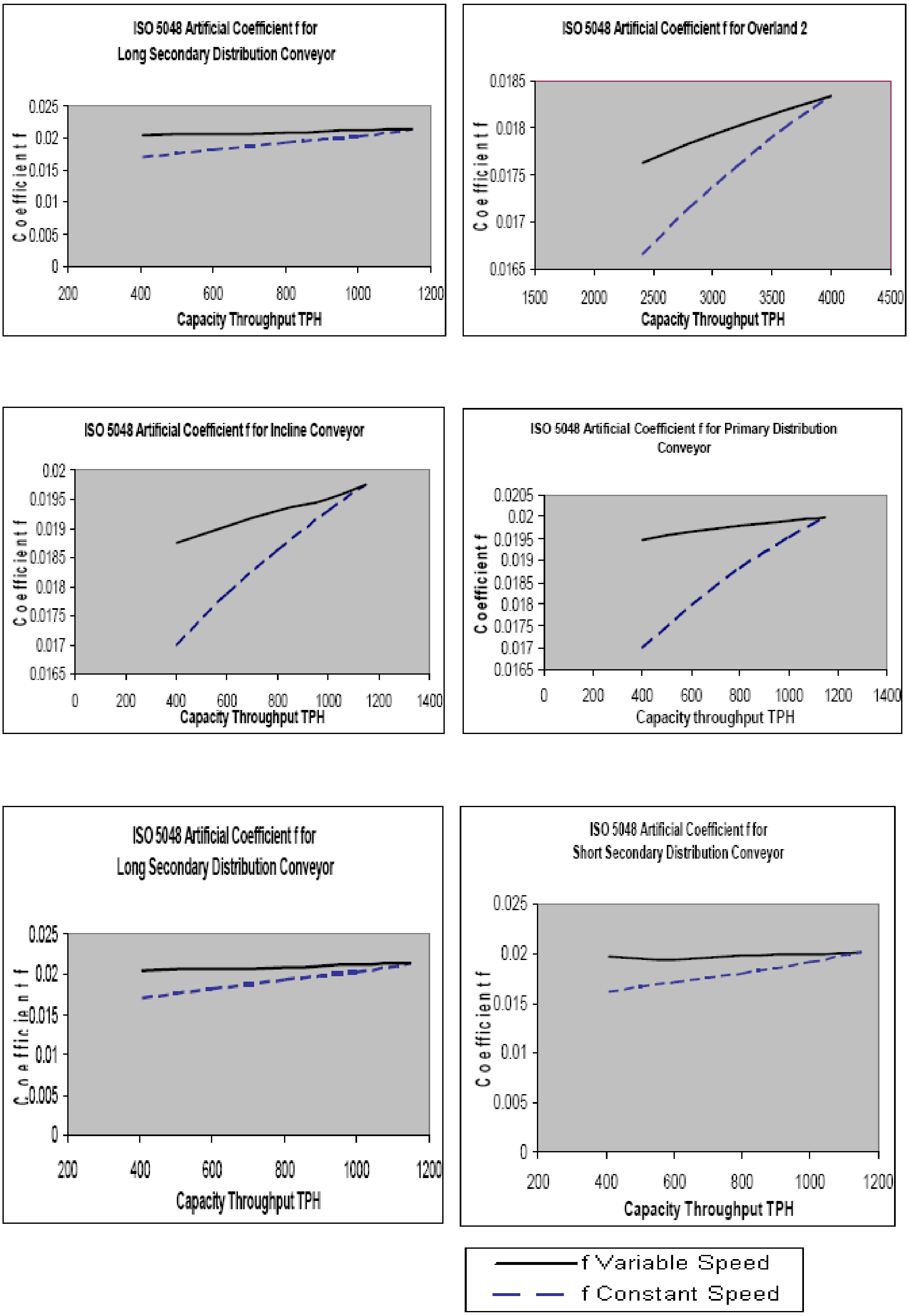


Figure 5: Comparison of the artificial friction factor for variable and constant speed operation

CONCLUSION ON THE RESULTS OF THE ARTIFICIAL FRICTION FACTOR CALCULATION

From the results on the calculation of the artificial friction coefficient as represented in Figure 5, it is evident that the constant speed operation results in a lower friction value at the reduced capacity range of operation for the conveyors evaluated. The friction coefficient is therefore greater in the case of the variable speed drive as result of the larger material load stream.

The potential benefit in this case for constant speed operation has to be considered on the basis of the combined implication of friction and speed that constitutes the energy consumption. The contributor that affects the greatest impact to this relationship will be the governing entity in terms of friction factor versus speed in determining the energy requirement.

4. COMPARISON OF THE ENERGY CONSUMPTION BETWEEN CONSTANT SPEED AND VARIABLE SPEED OPERATION

The required driving force on the driving pulleys of the conveyors and the operating power requirements for the capacity range is calculated on the basis of ISO 5048 by utilising the artificial friction coefficient values calculated above. The maximum operational belt sag tension is limited to a maximum value of 1% for the static operating condition in terms of pre-tensioning.

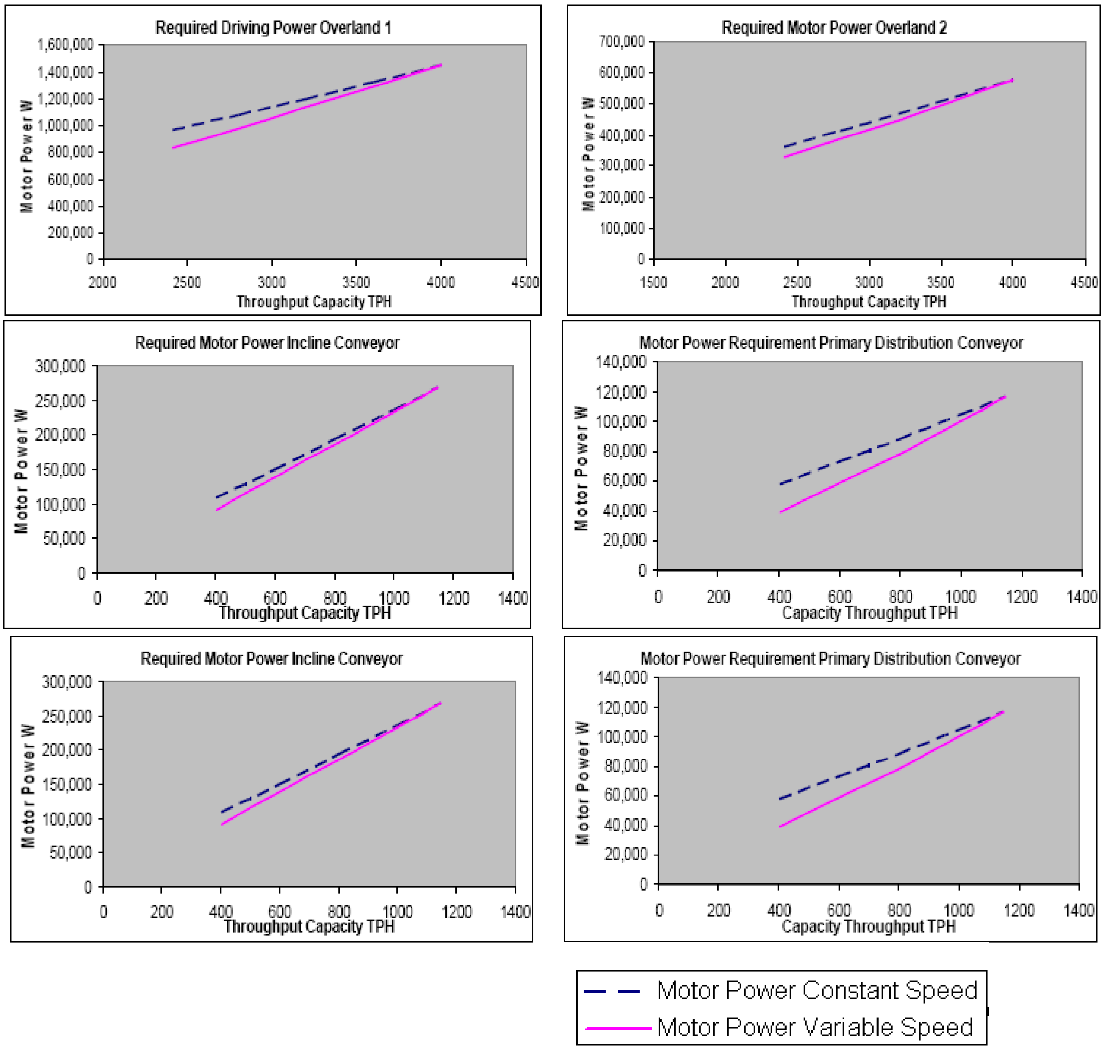


Figure 6: Comparison of the energy requirement for variable and constant speed operation

With reference to the graphs as illustrated in Figures 5 and 6 above, the results of the power requirement for the conveyors shows an opposite trend with reference to the artificial friction coefficient. In this case the variable speed option requires **less driving power** than the constant speed for the lower capacity range. The effect of the speed reduction in the case of the variable speed option compared to reduced friction in the case of constant speed operation has an overriding impact on the power requirement and the **variable speed** option presents a distinct **advantage** in this regard.

5. CONVEYOR COMPONENT EVALUATION

As discussed, the proposed conveyors operate normally under reduced capacity, however, the design is based on the maximum capacity required to maintain boiler load under the most unfavourable conditions.

In the evaluation of variable speed versus fixed speed technology, the maximum design parameters of the conveyor remain a common requirement. This implies that the component selection is for the same peak load condition and belt strength, pulley shaft sizing, idler maximum static load basis etc. is identical. It was therefore decided to compare the drive options in terms of the relative benefit expressed as a benefit factor relative to the peak load criteria in terms of life for the rotating components like pulleys and idlers. This benefit is calculated on basis of the reduced dynamic load condition that applies either in terms of reduced loading for fixed speed operation or the benefit as result of the reduced speed in the case of variable speed.

To enable this comparison it was decided to express the overall impact of the operating capacity range on the basis of the equivalent load for the combined impact of operational time spent at maximum design capacity versus normal operational capacity. The procedure as defined in publications like the NSK Bearing Manual express the equivalent condition in terms of rotational speed as follows:

$n_m = (n_1t_1 + n_2t_2 + \ldots\ldots\ldots + n_nt_n) / (t_1 + t_2 + \ldots\ldots\ldots + t_n)$

As far as abrasion or sliding wear impact is concerned at loading points or any other sliding wear interfaces like belt cleaners, it was decided to base the design evaluation between variable speed and fixed speed operation on the principle of the relative wear number as defined by the author Prof A.W Roberts in the publication titled 'Relative Wear'. The following equation for determining the relative wear number applies:

$N_{wr} = \delta_w / (\rho.g.B).(v_s / v_o).\text{ Tan } \Phi$

Where: N_{wr} = relative wear number
δ_w = Normal pressure at the boundary
ρ = Material density
B = Chute width
v_s = Entry velocity
v_o = Material velocity relative to the boundary
Φ = Friction angle between material and boundary

6. DESIGN EVALUATION OF THE CONVEYORS

The conveyors evaluated for the Medupi coal handling plant include the combination of overland conveyors that link the supply mine to the coal stockyard located in close proximity to the Power Station as well as the linkage conveyors from the stockyard to the station terrace storage and finally the terrace conveyors that includes the incline conveyors into the boiler house as well as the over mill bin distribution conveyors.

As far as the capacity sizing of these conveyors is concerned the following table applies:

Conveying Link	Mine to Stockyard	Stockyard to Terrace silo's	Terrace Silo's to Mill bins
Line configuration	*Single line*	*Dual line*	*Line per unit (6 lines)*
Maximum design capacity TPH	*4000*	*3200*	*1150*
Average capacity (most likely coal quality) TPH	*2409*	*1205*	*402*

Table 1: Capasity Sizing of Conveyors

The intended operation of the system is based on utilising all conveyors under normal operation. In cases of redundant conveyors, for availability reasons both conveyors will operate under shared load conditions rather than running only one of the dual conveyors with the second on stand-by. In the event of failure of a conveyor, the capacity shortfall will be compensated for by running the other conveyor in the pair at greater capacity. Refer to the schematic layout of the coal conveyors for Medupi Power Station as shown in Figure 7 on the proceeding page.

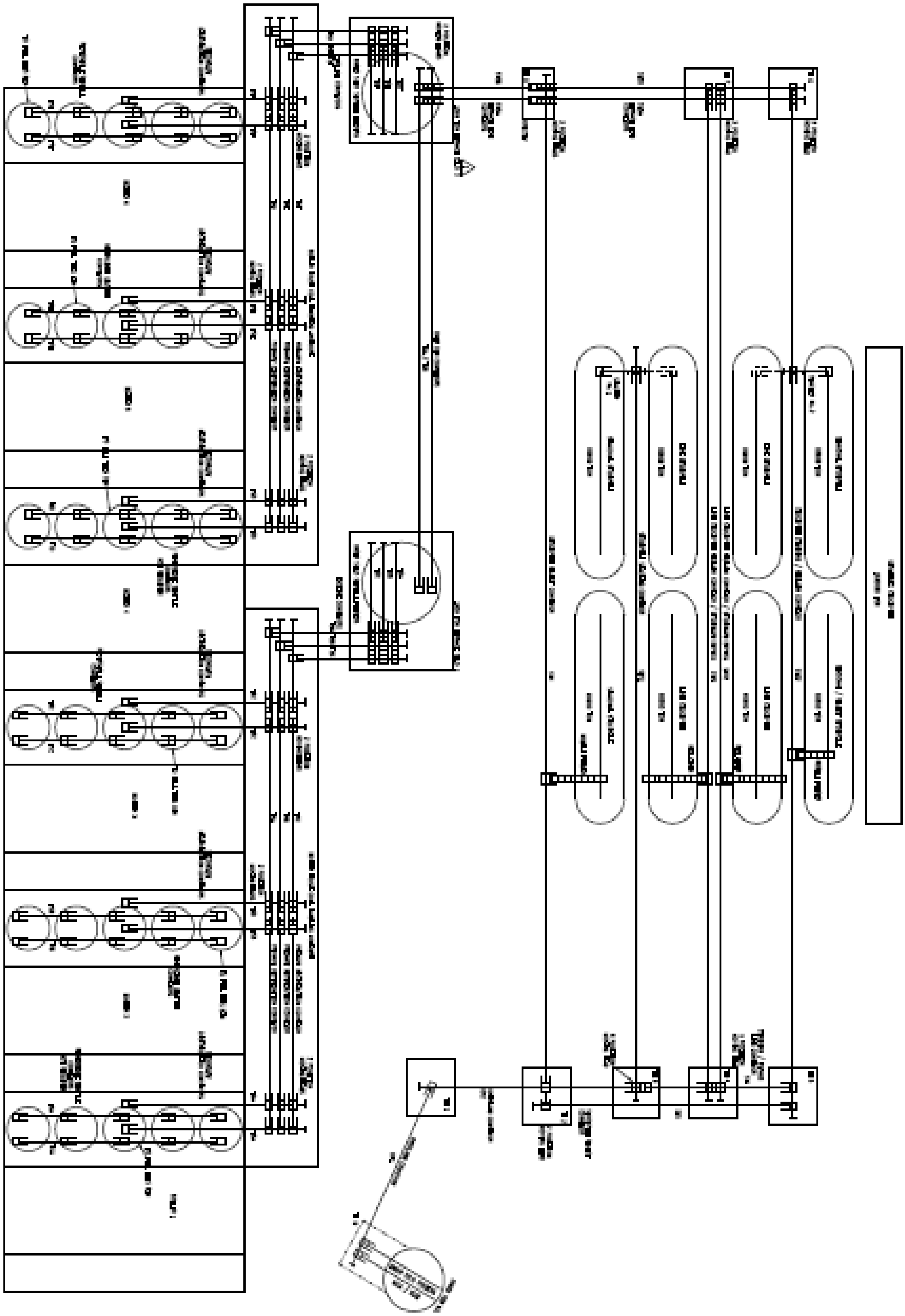


Figure 7: Schematic layout of the Medupi coal plant

The technical data pertaining to these conveyors are as follows:

Material data:	Unit	Overland 1	Overland 2	Incline	Primary Distribution	Secondary Distribution 1	Secondary Distribution 2
Capacity	TPH	4000	4000	1150	1150	1150	1150
Density	kg/m^3	850	850	850	850	850	850
Belt data:							
Belt width	m	2.1	2.1	1.2	1.2	1.2	1.2
Belt speed	m/s	2.86	2.86	2.65	2.68	2.65	2.65
Belt mass	kg/m	60.85	42.51	29.48	25.93	25.93	25.93
Belt sag	%	1	1	1	1	1	1
Belt factor of safety		6.7	6.7	6.7	6.7	10	10
Belt class	kN/m	2000	1000	1000	500	200	200
Conveyor data:							
Total length	m	4220	1165	338	243	60	21
Total Height	m	0	14	61	8	0	0
Idler data:							
Carry idler rotating mass	kg	55.9	55.9	25.1	25.1	25.1	25.1
Carry idler Pitch	m	1.5	1.5	1.5	1.5	1.5	1.5
Return idler rotating mass	kg	55.1	55.1	17.2	17.2	17.2	17.2
Return idler pitch	m	4.5	4.5	3	3	3	3
Trough angle	deg	45	45	45	45	45	45
Drive data:							
Drive pulley wrap	Deg	220	220	210	220	200	180
Pulley dia	m	1.05	0.7	0.7	0.6	0.6	0.6
Motor speed	RPM	1483	1483	1483	1483	1483	1483
Gear ratio		27.80	18.53	20.00	16.95	17.14	17.14
No of drives		2	1	1	1	1	1
Motor size	kW	800	800	300	150	50	30
Total no pulleys		13	12	8	16	6	2

Table 2: Conveyor Data - Medupi Coal Plant

The carry side idlers are of the 3 roll 45˚ trough arrangement and the return side uses 2 roll 10˚ 'Vee' arrangements.

6.1 COMPARISON OF THE CONSTANT SPEED VERSUS VARIABLE SPEED OPERATION IMPACT ON THE EXPECTED IDLER ROLL PERFORMANCE AND LIFE

The idlers are evaluated with the objective of comparing them in terms of the potential benefit with reference to the reduction in the dynamic load from the peak load criteria for the equivalent load, on the basis of the combined effect of the full load and normal operation load utilisation as illustrated in Figure 8.

The equivalent idler load benefit ratio is calculated for the combined effect of the carry and the return idlers. It is evident that the benefit in terms of the dynamic life is similar for both the constant and variable speed options. The impact of the reduced idler loading in the case of the constant speed technology and the reduced speed in the case of the variable speed technology on the dynamic load is almost equal as shown in Figure 8.

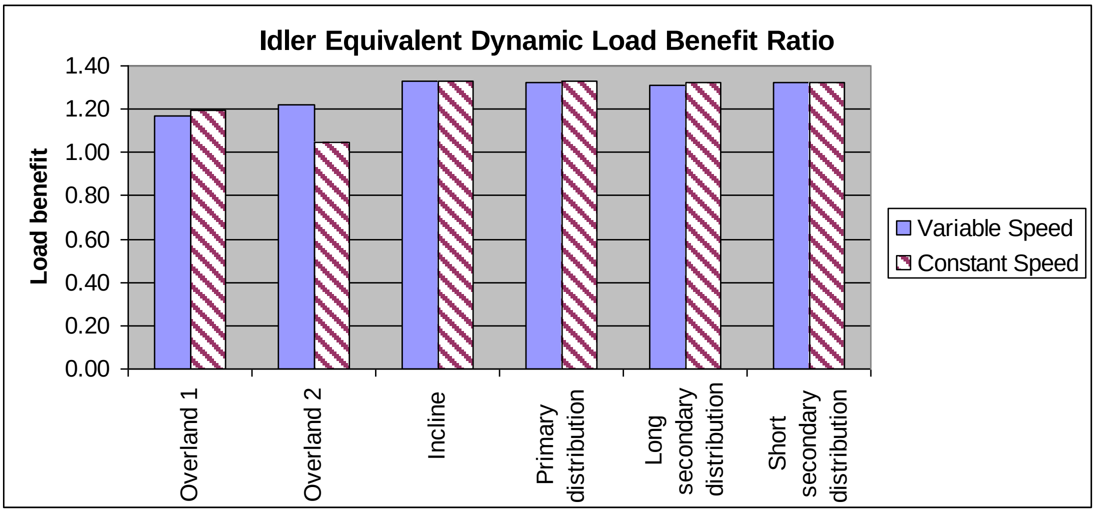


Figure 8: Results of the comparison of idler performance for the constant and variable speed operation

In conclusion, the potential benefit on idler life and performance for the variable and constant speed operation is very similar.

6.2 COMPARISON OF THE CONSTANT SPEED VERSUS VARIABLE SPEED OPERATION IMPACT ON THE EXPECTED CONVEYOR PULLEY PERFORMANCE AND LIFE

For the evaluation of the conveyor pulleys the method is similar to that applied to the idler rolls.

The complete evaluation of the pulley performance for all the conveyors is presented in Figure 9.

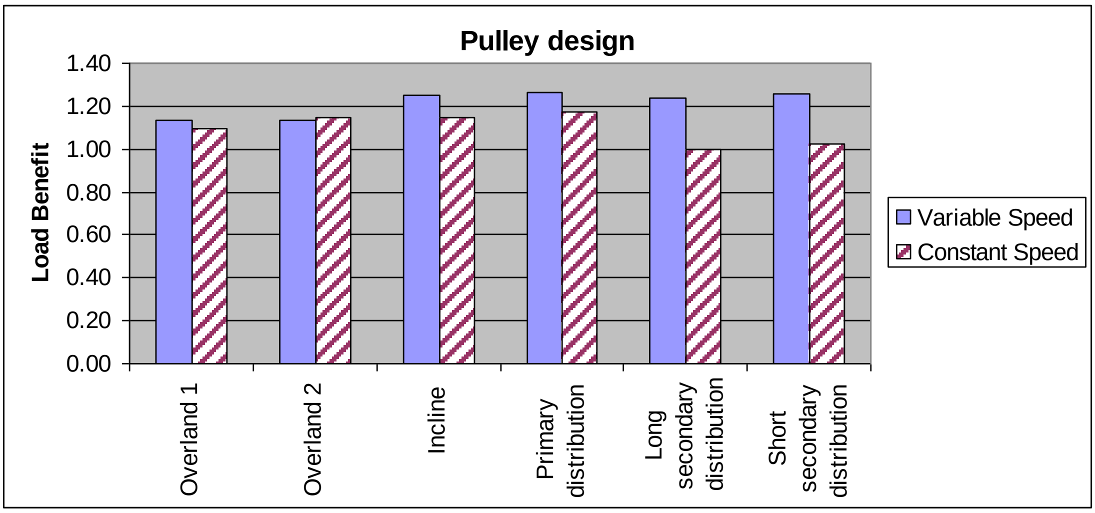


Figure 9: Summary of the conveyor pulley performance evaluation for variable and constant speed operation

The finding is that the variable speed operation presents a benefit in the expected pulley performance on basis of dynamic life expectancy.

6.3 COMPARISON OF THE CONSTANT SPEED VERSUS VARIABLE SPEED OPERATION IMPACT ON THE EXPECTED LIFE OF ALL AREAS OF PLANT EXPOSED TO SLIDING WEAR AND ABRASION

EVALUATION OF VARIABLE SPEED VERSUS CONSTANT SPEED OPERATION ON THE EXPECTED WEAR RATE FOR THE INCLINE CONVEYOR

The results for the wear comparison between variable speed and constant speed operation for all the conveyors are presented in Figure 10.

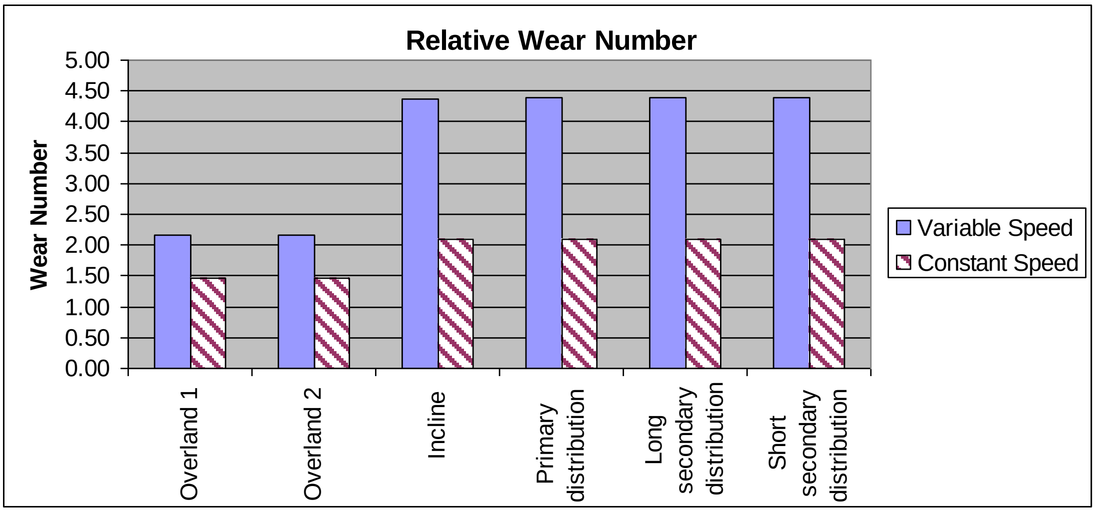


Figure 10: Summary of the wear impact between variable speed and constant speed operation for the conveyors

A reduction in speed results in a considerable reduction in the wear rate of the system, variable speed presents a distinct benefit in this regard. A higher wear number presents longer wear life expectancy.

7. GEARBOX OPERATION UNDER REDUCED SPEED

The design of the gearbox is based on the maximum load criteria. Normal operation in the case of variable speed drives implies high torque and low speed. The consideration in terms of the gearbox life of plant in this respect included the thermal implication as well as the lubrication effectiveness under reduced speed. Consultation with gearbox suppliers confirmed that both these aspects would be manageable on the basis of the gearbox selection. Lubrication should be of the type that ensures partial submersion of the rotating components rather than splash type lubrication.

It is also possible to control the operation of the system outside of the natural frequency range. In the case of the electric variable speed drive, this frequency range can be bypassed by control.

8. TRANSFER STATION DESIGN

The real challenge with variable speed operation of conveyors is the design of the transfer stations. The objective is to design profile plates in the upper and lower sections of the transfer arrangement that ensures that the position changes of the entry trajectory are intersected within a pre-determined range of a curved plate. The geometry of the chute is based on the material flow characteristics as well as the movement momentum of the material and the boundary friction implication. Optimisation is then applied by means of discrete element modelling (DEM) of the material as it moves through the chute. The wear liner material of choice is ceramic tiles for the high wear areas.

THE UPPER CHUTE (BONNET)
The trajectory is carefully guided by means of a smooth convex curve (bonnet) that narrows the flow channel along its contact length for the high speed range of operation. The intersection angle between the trajectory and the bonnet is maintained at a minimal value. This design enables the control of the material flow onto the bottom profile plate (ladle) as illustrated in Figure 11. The narrowing design of the bonnet along its sliding length ensures a more manageable material cross section profile when angular directional change is required in plan between the incoming and receiving conveyors.

In the case of the low speed range, the trajectory does not intersect with the bonnet. In this case it free falls by gravity and intersects the ladle at a minimal relative angle.

THE LOWER CHUTE (LADLE)
The ladle is profiled to ensure that the in-feed of material from the upper chute zone within the speed range is directed onto the ladle that ensures a minimal intersection angle to facilitate that directional changes are applied gradually.

This collection area of the ladle is followed by a concave curved profile to further enable speed control of the material as well as ensuring a velocity component in the direction of the receiving conveyor.

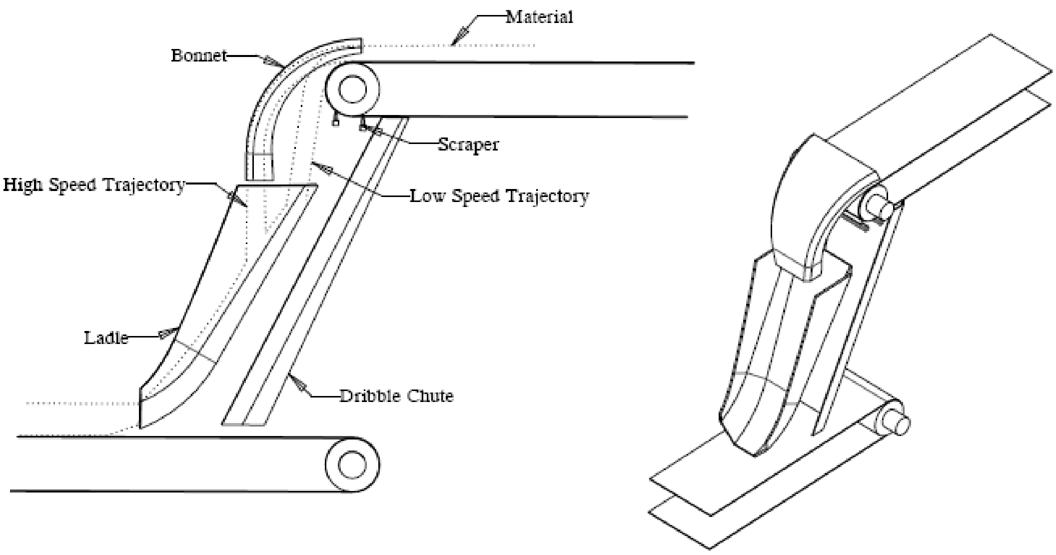


Figure 11: The illustration of a guided flow transfer point with bonnet and ladle arrangement

DRIBBLE CHUTE
Dribble chutes are mostly arranged separate of the main transfer chute to ensure adequately steep angles for the sustained flow of the belt scraper discharge that usually contains low speed, low flow momentum and great cohesion and adhesion characteristics. Dribble chute walls are lined with a poly plastic type of liner based on its low adhesion quality in the case of moisture. Material is then discharged onto the transition section of the receiving belt.

9. DISCUSSION ON VARIABLE SPEED ELECTRIC DRIVES VERSUS THE HIGH SPEED FLUID COUPLINGS FOR CONSTANT SPEED.

The need for controlling the dynamic behaviour of a conveyor belt system via the drive is to affect the following:

- Smooth starting,
- Energy saving,
- Increased plant life, and
- Process control requirements.

Direct on-line starting of electric motors can cause the following problems:

- Slipping of belts at the drive pulley,
- High wear and tear on couplings, gearboxes, bearings and other mechanical components,
- High inrush (starting) current,
- Equipment damage when starting against rotating machinery,
- Torque spikes in Star-Delta and Soft-Start systems, and
- Safety issues

The basic drive technology of the modern frequency converter electric variable speed drive has a few basic components (Refer to Figure 12):

- Rectifier,
- Fixed DC link voltage, and
- An inverter that controls output voltage and frequency
  -The inverter section is a number of electronic 'switches'
  -This is used to convert the DC link energy, by a series of pulses

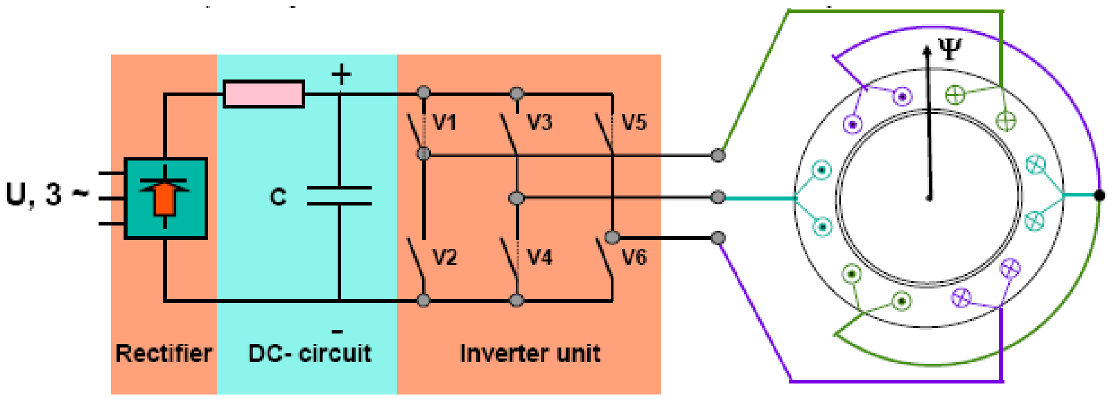


Figure 12: The components of a frequency converter variable speed drive

The conventional technology to control the starting characteristic in the case of a constant speed conveyor is a Fluid Coupling (FC). In the case of variable speed drive control an Electrical Variable Speed Drive (VSD) is used.

In the case of the Fluid Coupling the following features can be mentioned:

- Soft and shockless starting of machines and conveyor drives,
- Acceleration of very large masses without the necessity to use oversized motors,
- Load relieved and faster motor start since coupling torque grows proportional to the second power of motor speed. Negligible heating-up of motor, as the high starting current is only drawn for a short time,
- Starting of heavily loaded machines by induction motors also with flat motor characteristic (voltage drop, high voltage motors) by utilizing the motor pull-out torque,
- Limitation of torque when starting conveyor belts,
- Load compensation in multi-motor drives as result of the ability to slip and varying the oil filling level; successive starting of motors by reducing starting torque and avoiding simultaneous starting current peaks,
- Little slip of couplings at nominal static load condition,
- Easy adjustment of transmittable torque by varying the fluid level,
- In case of overload, protection of the fluid filling is possible by means of electronic or mechanical thermal control devices, and
- Water as operating medium is possible in a special coupling design.

An Electric Variable Speed Drive has the following key features:

- Automatically adjustable torque limitations,
- Load variations automatically compensated for (no need to adjust oil levels),
- VSD can supply a 150% overload condition,
- VSD can give operators sufficient warning before tripping,
- No oil spills – environmental impact,
- Load sharing in multi drive motor system applications
- Energy savings
  - Power factor (0.96 for VSD compared to 0.85 for constant speed),
  - Speed control is based on demand, and
  - Load dependant control.
- Load dependent belt speed adjustment,
  - Soft starting,
  - No in-rush current or high starting currents (Transformers and switchgear do not need to be oversized),
  - No limit on the number of starts per hour (electronic starting),
  - Adjustable starting and stopping ramp times,
  - Load sharing control is independent of capacity loading on the belt, and
  - Regenerative ability for a controlled ramp down.

The main reasons for considering a variable speed drive option are:

- No high motor starting current,
- Synchronising (load sharing) is much easier with VSD'S, the control adjustment is immediate,
- Controlled start up times on conveyors is possible on basis of the actual load condition,
- On regenerative conveyors runaway can be controlled by means of the VSD,
- No belt slip problems at drives
- Reduced maintenance cost,
- Energy savings,
- Reduced risk to the performance in terms of human influence by applying incorrect maintenance interventions, and
- it is possible to manage the natural frequency range by means of 'skipping' thereof in the case of the VSD.

10. CONCLUSION

This paper presents the engineering process followed in deciding the type of conveyor drive technology for application to the coal plant conveyors for the new Eskom Medupi Power Station.

The conclusion is that the variable speed drive (VSD) presents a distinct benefit in terms of energy savings and reduced risk with reference to incorrect maintenance interventions that result in plant unavailability. The calculated energy cost saving for the conveyors (excluding all coal stockyard machines) evaluated in the case of Medupi for a projected life of 50 years equates to R21million in the case of the variable speed option in NPV terms base dated April 2009.

The finding of this study is applicable to this specific application. These conveyors evaluated are designed for the maximum load condition as far as component sizing is concerned, although the normal operation of the plant occurs at reduced capacity throughput. The VSD has fewer moving parts or wearing parts and the reliability is therefore enhanced.

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