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Large Conveyors - The Case for Total System Design

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INTRODUCTION

Traditional methods for conveyor design can lead to situations where the conveyor belting and mechanicals are over or underdesigned. For larger installations, involving high capital cost and strategic considerations, a more careful analysis of the system can lead to substantial benefits.

This paper attempts to address these considerations and proposes a design method which takes account of all the identifiable parameters.

By so doing, the conveyor will be designed as a system, and system optimisation is possible.

1. DESIGN CONSIDERATIONS

1.1 Conveyor Capacity

The correct interpretation of user defined conveyor product transfer rate is possibly the most important aspect of conveyor design. It is not unusual for a user to specify conveyor capacity in tonnes per hour where this figure is arrived at by the ratio of the annual transfer rate and the number of operating days per year, assuming 24 hours per day.

The two commonly used methods to overcome the above shortcomings are for Engineers to call for a defined peak tonnage figure, or by applying an experience factor.

The former method has obvious shortcomings in that the user can rarely define the peak tonnage with any degree of certainty. In order to reduce costs the peak tonnage is likely to be understated by the user. Furthermore in at least one instance in my experience the user interpreted the peak tonnage to mean the tonnage which the conveyor would carry continuously for an 8 hour period.

The value given to an experience factor (ie ratio of peak to average tonnage) in large installations, requires a careful analysis if an optimal design is to be arrived at. The number of feed points, bunker or silo capacity, material consistency and feeder design can all have a bearing on the factor.

As an example, a conveyor taking product away from a number of coal producing faces requires a very high peak/average ratio, whereas a conveyor removing a -25mm product from a large silo through a closed loop feeder requires a relatively small ratio.

It would be advantageous to both user and designer, if a system, similar to that used by electric cable manufacturers, could be compiled for use on conveyor sizing. This would mean assigning factors to readily identifiable conditions which influence the peak/average ratio. A factor of 1 could be assigned to, say, a size factor where the product size variation is small and the product is say -25mm, the factor increasing with product size and size variation. Similarly factors could be assigned for the ratio of surge capacity to average tonnage, type of feeder, method of feeder control etc. By multiplying the various factors, an overall factor for the peak/average ratio could be obtained.

The implications of lump size, surge loading and variations in material bulk density on conveyor design are well known to all who have been involved in conveyor belt design.

In addition to the above, the product bulk density is recognized by all Engineers in designing a conveyor system. However, cognizance must be taken of the range of bulk densities which can occur through a wet, fine product to a coarse, dry product, which are likely to occur from time to time.

It is possible for a conveyor which is sized for a lumpy product to have inadequate power or starting torque when fed with the fine, wet product.

Operationally, the misinterpretation of information, or incorrect information supplied by the ultimate user of the conveyor can lead to conveyors which either cannot start under full load conditions, cannot carry the full load, have excessive spillage or can be destroyed under worst case conditions. Many other more minor problems can arise which are more easily solved.

Corrective action in the case of smaller conveyors, such as plant conveyors, is usually possible at reasonable cost, but where large or strategic installations are involved, design inadequacies could be traumatic. It is therefore natural for Engineers to be very conservative in their approach. It is not the intention to condemn such conservatism but rather to examine some of the design considerations in more detail in an attempt to arrive at an optimal design for any large application.

1.2 Conveyor Tensioning

Due to the stretch on long conveyors through varying loads and during starting, it is common practice to make use of automatic, electrically operated winches to apply the necessary return belt tension to a conveyor belt.

However, the methods of control and potential maximum tension that the system can apply are frequently overlooked by designers.

The simplest method of controlling the winch, hence the cheapest, is to use a tensiometer with a high and low set point. The disadvantages of this system are that it tends to "hunt" between the two set points and either reacts too slowly during starting or too rapidly during running of the conveyor. Unless a more sophisticated control system is used, or the conveyor acceleration rate is low, it is impossible for this system to maintain optimal return belt tension at all times.

A potentially more disastrous side effect of the above is that, due to the frequent stopping and starting of the winch, an intermittently rated motor is required.

A.C. motors are normally rated for continuous duty, and the term CMR (continuous motor rating) is used. An intermittently rated motor is in fact a motor with a higher CMR rating which is suitable for the stop/start duty it is required to perform ie. a 5kW intermittent motor may be a 15kW CMR motor.

Furthermore, standard electrical protection is designed to prevent a motor operating in an overload condition for a period of time, but cannot prevent the motor from producing maximum torque.

Assume now that the winch is designed to produce a maximum tension equivalent to 6kW. Due to the fact that a 15kW CMR motor is required, which could have a maximum torque of 200% of full load torque, it is apparent that the motor is in fact capable of torque equivalent to (2 x 15)/6 or 5 times designed tension, before the electrical protection would operate.

In order to prevent the above occurrence, which incidentally, is possible with more sophisticated systems, we do not believe an ultimate over-tension trip on the winch system is sufficient on a strategic installation, since it is possible for the protection to fail or for the winch motor contactor to burn in.

A gravity take up system with a correctly rated gravity mass positioning winch is preferred by this company for all long conveyors where the take up carriage travel precludes a standard gravity system. The inability to apply extra tension during starting is a common criticism of this method, but it is possible to use a pair of gravity masses, connected by means of a flexible connection, to overcome the problem. During start up, the upper mass and lower mass are used, whereas during normal operation, the pair of masses are lowered until the lower mass rests on a support. However, it is believed that the provision of extra tension during start up is neither necessary nor desirable, as discussed later.

1.3 Spillage

Spillage on long conveyors has a number of ramifications.

The less serious problems are that spillage is unsightly, can cause pollution and results in loss of product. The pollution can be either pollution of the product through loading spillage onto the conveyor which could cause contractual problems, or air or water borne pollution which might result in legal action against the user.

From an operating point of view, however, the potentially most serious result of spillage can be serious tearing of or damage to the belt due to the jamming of idlers etc by the spillage. Apart from the cost of repair, the resultant downtime can have very serious financial implications.

Most reasons for, and methods of overcoming, spillage are well documented, but examination of some points could be worthwhile.

Firstly, with some products, carry over on the return belt can be the major reason for excessive spillage. In a fairly large number of instances, commercially available belt scrapers cannot effectively prevent spillage and they reduce top cover life. For all but the high cost, large installations they are usually perfectly adequate, however belt washing or belt turnover techniques are probably cost justified on large installations. Belt turnovers can result in the edge cords of steel cord belting being subjected to higher than average tensions and it is strongly recommended that this effect be properly analysed and receive due cognizance when rating the belt. The apparent cost savings of a cheaper turnover system might be a fraction of the increased cost due to the necessary increase in belt rating which results, as discussed later.

It has been shown that in the event of a single cord failure in a steel cord belt, the distribution of the tension previously taken by that cord in the healthy cords is not shared equally between the cords, but tends to be concentrated in the adjacent cords (Reference (1)). The failure of overtensioned edge cords in an inadequately designed belt turnover can therefore lead to a catastrophic "runaway" type of failure in a belt.

Obviously the same comment will apply to uneven tension distribution in cords at transitions and vertical and horizontal curves.

Finally, in some instances a conveyor is fed by a high inertia machine such as a longer conveyor or jaw crusher. If the transfer chute is incorrectly designed, excessive spillage and/or idler overloading and power wastage can occur.

The latter point can, once again, influence belt rating. For normal shutdown the machines can be sequenced to allow the conveyor to clear the excess material before stopping, however unplanned shutdowns, such as emergency trips or power failures, will occur and must therefore be allowed for in chute design.

1.4 Conveyor Starting

Anyone who has operated a belt conveyor has witnessed the transient instability caused by high acceleration rates during conveyor starting. These affects on short conveyors are relatively insignificant, but they tend to increase with conveyor length.

Various methods of reducing the acceleration rate of a conveyor have been in use for a considerable length of time, but it is extremely doubtful that designers have, until recently, made any serious attempt to quantify the repercussions as far as conveyor design is concerned.

The acceleration from rest of a conveyor results in two sources of tension in the conveyor cords, namely, the tension in the cords required to accelerate the belt, material and all associated non driven rotating masses, and a tension/compression wave as a result of the spring - mass effect of the belt. The former tension component is readily available but the latter, although obvious, has not been quantifiable. Recently and mathematical models have been produced to enable the transient tension wavee amplitude to be calculated (Ref 2, 3). (In fact the transient effect consists of two waves, a tensile wave on the load carrying side and a compression wave on the return belt side. The wave front velocity and amplitude after attenuation depend on belt mass, tension etc).

The Consulting Engineering firm of Conveyor Dynamics Inc. are able to analyse conveyor designs using software produced by their President Mr L.K. Nordell. The computer simulation is capable of predicting potential transient problem areas in complex designs, such as head and tail drives. The predicted conveyor response has been verified by physical measurements.

In order to limit the overtensioning effect that occurs during conveyor acceleration it is therefore desirable to be able to calculate both of the above tensions.

An important point to note in this regard, is that worst case conditions from a transient tension point of view would usually occur if a conveyor was stopped during the acceleration period. This can occur when high drive torque results in belt slip, with either the operator operating an emergency stop or the belt slip protection device causing the drive motors to trip.

The earlier reference to extreme edge tensions in belt turnovers can be applicable in the above event.

2. CONVEYOR DRIVES

2.1 Conventional Systems

2.1.1 A.C. Motors

At the risk of stating the obvious, an A.C. motor, which is the traditional conveyor drive component, suffers from the problem of being designed to operate at essentially a fixed speed, dependent on the supply frequency, number of poles and the slip commensurate with the load torque.

A.C. squirrel cage motors are the most commonly used drives by virtue of low cost, relatively high efficiency and absence of moving parts, with exception of the bearings.

A.C. wound rotor motors (slipring motors) are equipped with rings and brushes which require periodic maintenance. The motor efficiency in practice is slightly lower and the motor prices are substantially higher. It is obvious that continued purchase of such machines is due to certain benefits of wound rotor motors, which are discussed below.

An A.C. motor switched directly onto mains has therefor to operate at one particular speed, since slip is usually less than 1% at full load for a squirrel cage motor and 2% for a slipring motor. The motor torque varies with speed, at standstill at above 100% full load torque, then increasing to a maximum torque before decreasing to zero at synchronous speed. (Refer to Fig. 1).

Unless a torque limiting device is used, a conveyor belt will be subjected to tension equivalent to the motor torque during start up.

For a conveyor to operate under maximum load conditions, the starting, pull up and full load torques must exceed the torque required by the load. It is not uncommon to find that a designer has failed to take cognizance of this fact. A common manifestation of this is that a conveyor will start under heavy load but on reaching between 15 to 20% of full speed will not accelerate further and will eventually trip out. The reason for this is that the pull up torque is equal to or less than the load torque.

The other point to bear in mind is that the maximum torque the motor will deliver during start up can be between 200% to 350% of full load torque. Provided the drive pulley does not slip, the equivalent tension will be imparted into the belt. To exercise any control over an A.C. motor during the starting of the driven machinery it is therefore necessary to introduce a device between the motor and the machine which can modify the motor torque characteristics, or electrically modify the motor characteristics, or modify the electrical supply to the motors.

2.1.2 Torque Limiting Devices For Use With Squirrel Cage Motors.

2.1.2.1 Fluid Couplings

For long and intermediate conveyor applications, fluid couplings are commonly used in conjunction with A.C. squirrel cage motors which provide varying degrees of torque application rate, as well as torque limitation. However, the selection of the correct coupling is not purely a matter of selecting the cheapest unit.

A factor which is apparently overlooked in many instances refers to earlier discussion on the tonnage that can be fed onto a conveyor. It is understandable that an unsophisticated operator could make use of generous free board on a conveyor while feeding a fine wet product and thereby substantially exceed the designed maximum torque requirement, leading to continuous coupling slip and eventual fusable plug failure.

Furthermore, in the case of dual drive conveyors, both the motor slip and coupling slip characteristics of the pair of drives require to be identical in order for the drives to share load. (Refer to Fig 15). In the likely event that one drive is more highly loaded than the other, coupling slip can occur on the loaded drive at less than conveyor full load. Any attempts by operating staff to improve on load sharing invariably leads to both couplings filled to maximum capacity, which might be well in access of the conveyor designer's maximum torque figure used to design the belt mechanicals.

At the risk of stating the obvious to delegates fully conversant with fluid couplings, I would like at this stage to correct a misconception that is fairly commonplace.

A fluid coupling does not reduce an A.C. motor's inherent maximum torque, neither does it reduce the torque delivered by the motor.

At start up the motor accelerates to approximately synchronous speed. At this stage, the torque delivered by the coupling is virtually zero, and the motor torque is equal to the coupling torque plus a small torque equivalent to the mechanical losses in the bearings etc of the coupling. The output shaft speed is zero, or stated in a different way, the coupling slip is 100%.

As more fluid is introduced into the coupling, the coupling torque transmitting capability increases, or in other words, there is more drag between the input and output shaft of the coupling. As a result of the increased drag, the motor slows down slightly until the motor torque has increased to a torque equal to the coupling output torque plus mechanical, and now, additional small hydraulic losses.

In other words, the motor torque follows the increased drag, and hence output torque of the coupling, but apart from a very small ($\pm$ 1%) loss in speed, the motor continues to operate at full speed. The difference in speed between the input and output shaft of the coupling results in the coupling absorbing some of the motor power, proportional to the product of the torque and the difference in speed between the input and output shaft, which is converted into heat.

The coupling therefore raises the output speed of the drive by varying the slip in the coupling but does not reduce the motor torque. However, if the coupling can only transmit a torque which is less than maximum motor torque, the coupling slip increases, and by so doing, does not allow the motor to slow down to the speed equivalent to maximum torque.

The above explanation assumes that the rate of torque application by the coupling has allowed the motor to reach operating speed before full load torque is applied.

Should this not be the case, the motor will draw excess current and the coupling will be required to absorb far more power. This will invariably occur if traction couplings, or short delay couplings, are used on high inertia machines. Unfortunately, this is a common occurrence, undoubtedly due in most cases to machine suppliers attempts to reduce costs.

Obviously too, further requirements of dual drives are that motors and couplings should be as closely matched as possible and that the pulley diameters are closely matched.

The selection of pulley diameter can be difficult owing to the increased stretch resulting from the higher tensile load in the belt on the primary drive.

With scoop controlled couplings, the rate of torque application and maximum transmitted torque are readily controllable and it is possible to get relatively good load sharing. Cognizance must however be taken of the waste heat generated in the coupling with the greater slip in order to stay within the rating of the coupling.

If the torque/speed mismatch on dual drives is relatively substantial, and an electronic load sharing system is in use, the additional coupling slip on the high torque drive can be far higher than a coupling application Engineer would allow for.

2.1.3 Electronic Soft Starts

2.1.3.1 Variable Voltage Systems

The advances made in industrial electronics led to the appearance in the 70's of the so called "soft starters" in order to overcome the problems related to the uncontrolled torque of the squirrel cage motor.

The soft starters essentially use thyristors or GTO's (gate turn off devices) as high speed switches. The high speed switching allows current to flow for a portion of each cycle. Since the time of switch-on can be controlled, the end effect is similar to being able to vary the voltage applied to the motor.

The output torque of an A.C. motor is proportional to the square of the voltage, and it is thus possible with these devices to vary the motor output torque.

Fig 6 shows the motor output torque against speed at reduced voltages.

From the figure it is apparent that the torque still varies with speed. Starting a motor at reduced voltage does not therefore guarantee that a predetermined torque will be applied by the drive.

The output torque of a motor is also not directly related to the current drawn by the motor throughout the speed range of the motor. Had it been the case, it would have been a simple matter to control the switching rate to give a constant motor current, and thereby a constant motor torque.

However, the versatility of modern electronic equipment makes it possible to provide acceptable electronic models of an A.C. motor, which can in effect provide varying degrees of sophistication in the complete drive system.

In the simplest form, a soft start unit will start at some predetermined low voltage and increase the voltage until the full voltage is applied to the machine.

Fig 7 indicates the resultant speed/time curves at various ramp times for the operation of such a soft start system when used on a conveyor or similar machine with an essentially constant torque/ speed characteristic.

Fig 8 indicates the motor output torque versus time characteristic for ramp times varying from 5 to 20 seconds.

It is apparent from the torque/time curves that the only affect of the simple soft start system is to reduce initial starting torque. Unless the motor has reached full load speed during the ramp up time, the normal full voltage torque/speed curve comes into effect.

The peaky nature of the torque/time curve as compared to the torque/speed curves considered earlier, is due to the high applied torque in the pull out torque region, resulting in rapid acceleration and, therefore, shortening of the time scale.

The high pull out torque at speeds approaching operating speed is responsible for the often witnessed overrun phenomenon seen in "direct-on- line" started conveyors or transmission chain driven devices.

The computer programme which generated the speed/time curves was not written to indicate overrun. It is obvious that the high acceleration near operating speed will in fact result in a greater or lesser amount of overrun, depending on the inertia and, in the case of a conveyor, the elasticity of the belt.

What is however evident from the speed curves is that a simple soft start system has the effect of delaying the start and reducing the snatch at start up. It does not have any beneficial effect above approximately 10% of operating speed.

There is however a disadvantage associated with such systems, namely, the fact that the motor is stalled for a period dependent on the time ramp and conveyor load. Although this occurs at reduced current, and therefore reduced heating, the motor ventilation is not operative. Motor burn out rates are likely to increase with the use of simple soft start systems.

It is unfortunate that a number of ignorant or unscrupulous suppliers have in fact sold many such soft start systems for conveyor applications. Needless to say, they have not been successful and might in fact have damaged the reputation of soft start electronic systems as a whole.

In our own organisation some of our Engineers have been convinced by suppliers that the Head Office Consulting Engineers' rejection of these systems is uncalled for, and successful applications for the starting of pumps and fans are sited as evidence of the performance of the drive. The load curves of pumps and fans are suited to the characteristics of conventional soft start systems, and the satisfactory application in these cases is obviously no indication of their performance with conveyors.

As with fluid couplings, a customer gets what he pays for. The more expensive soft start systems incorporating current over-ride ramp hold and tachometer feed back facilities are more expensive. A larger ramp time also requires a higher component rating in the devices which are bypassed once "fully on", once again affecting price.

Some of the electronic soft start systems are provided with a facility to ramp up and ramp down, and could therefore apparently prove useful with long inclined conveyors, where rapid deceleration might be problematical.

Before leaving soft variable voltage starts, the motor heating effect of the device must be considered. An A.C. squirrel cage motor starting from rest has a very low power factor of the order of 0,3 to 0,4. Expressed in simple terms this means that approximately 30 to 40% of the current drawn by the motor at this stage is being converted into mechanical energy. However, as far as the motor is concerned, the extra current required to provide the torque is heating the motor. In addition, during the starting period, the motor ventilation is operating at below full efficiency. The use of soft start techniques reduces the motor current, ideally to a level providing adequate starting torque, but increases the starting time.

Since the motor heating effect is proportional to the product of time and the square of the current, (I^2t), the selection of the motor does require attention.

As an example, a typical 300kW squirrel cage motor has a direct on line torque of 1,6 x full load torque. To reduce the torque output to 1,1 x FLT, the voltage must be reduced to $(1,1/1,6)^{\frac{1}{2}}$ or 83% of full voltage. This corresponds to a current of 5,8 x full load current as compared to 7 x full load current at full voltage. Simplistically, but to a reasonable degree of accuracy, this means that the start up time with the soft start system should not exceed 1,45 times the direct on line starting time if motor heating is not to exceed that of direct-on-line starting.

A solution to this problem is to use a motor with a higher rating. The problem as far as the conveyor designer is concerned, is that the pull out torque rises proportionately. In the above case, if a start up time of 3 times D.O.L. starting time was required, the motor pull out torque will increase to 6,62 x the required full load torque.

Designers should take cognizance of this torque since a possibility exists that the soft start can fail or even be bypassed if faulty.

2.1.3.2 Variable Frequency Systems

A soft start system which is available at a cost premium of 300 to 400% of a conventional soft start, is very similar to the system used for variable speed A.C. squirrel cage motor drives. This system converts the 50Hz 3 phase supply into a variable frequency 3 phase supply to the motor.

This system allows the motor to operate in the stable portion of the torque speed curve. Consequently the motor is operating at a high power factor and the motor current is approximately equal to the required accelerating torque. Long smooth start ups are therefore possible without overloading the motor.

A further advantage of this system is that the motor speed is virtually independent of torque on this part of the curve, since 1% slip is roughly proportional to 100% change in torque. The acceleration time is therefore dictated purely by the ramp time and not by the load.

Since the motor operates in a slip range where the torque is proportional to the current, over torque protection is available, which being electronic and acting on the thyristors, can operate at high speed and thus prevent the motor pull out torque being applied to the conveyor. A back up circuit breaker, or overload, rated close to the motor full load current, can give added security.

2.1.4 Slipring (Wound Rotor) Motors

2.1.4.1 Lossy Method of Control

A further method of controlling the rate of application of torque is by means of a slipring (wound rotor) motor. By varying the resistance connected to the rotor, the slip and hence the torque can be widely varied. A typical series of speed/torque curves for varying rotor resistance is shown in Fig 2.

It will be noticed that the effect of increasing rotor resistance is similar to moving the vertical axis to the right and expanding the horizontal scale back to the original scale. Furthermore the torque speed curve with no resistance in the rotor is very similar to that of the squirrel cage motor.

The varying torque with speed for any particular resistance is a problem which has to be overcome in order to maintain a relatively consistent accelerating torque. Electrolyte systems are available which can give continuous variation of resistance at relatively high cost or a number of steps of variation at lower cost.

Alternatively, a much higher number of steps of fixed resistance can be switched.

The latter two methods have a disadvantage in that a torque step is applied with each switching operation (see Fig. 3). A sudden application of torque gives rise to a tension transient in the belt, with two or more transients being additive if the belt natural frequency is the same as the switching frequency.

A beneficial side effect of introducing resistance into the rotor circuit is that the current drawn by the motor at any particular speed is reduced.

Liquid resistors can be used and consist of basically two types, namely, those using movable current carrying vanes or electrodes and those using fixed electrodes in an electrolyte bath.

The latter are usually cheaper and less maintenance intensive but are generally available only for lower motor power. An additional disadvantage is that for conveyor starting applications with long conveyors, the required resistance range is usually greater than a single device can give. Consequently, more than one device is used and they are switched in progressively. The resultant torque steps are similar, but not as severe, as those shown in Figure 3.

The movable electrode rotor starter enables the user to get continuous variation of resistance over a relatively wide range. A further advantage of this type of starter is that torque control is very simple by use of a current sensitive relay switching the power to the electrode motor. By selection of electrode speed and electrolyte solution strength it is theoretically possible to obtain very smooth starts with this system, and my company is about to experiment with such a system.

As with fluid couplings the variation of the torque with speed is accomplished by absorbing power. In this case electrical power is removed from the rotor by the resistors. The thermal capacity of the system must therefore be selected to suit the duty.

2.1.4.2 Loss-Less Rotor Control

The frequency of the rotor voltage is the difference between the electrical field rotation speed and the speed of the rotor. It therefore varies from 50Hz at start up, to effectively 0Hz at operating speed.

Using electronic switching (thyristors) it is possible to convert the variable rotor voltage frequency to a fixed frequency, and the rotor power previously dissipated in resistors can be fed back into the power circuit. Since the power is now being recovered, the drive can be used as a variable speed drive with an efficiency and power factor approaching that of a standard motor.

As with the lossy system, a rotor circuit fault can result in maximum torque of the order of 3 times full load torque.

In both squirrel cage/fluid coupling and slipring motor applications, starting times are often dictated by the capacity of the coupling or rotor resistor, and the use or otherwise of forced cooling.

Superficially it makes economic sense to reduce the starting time as much as possible. However the much higher starting torques required could result in increased belt ratings and possibly more costly drive mechanicals.

2.1.4.3 Rotor Reactor Control

Systems using low loss reactors are sometimes proposed as a more efficient method of wound rotor motor starting.

While this method reduces the motor current without the disadvantage of heat generation that occurs with resistors, the motor torque is drastically reduced. In addition the shape of the torque speed curve is not greatly different from that of a squirrel cage motor. This system therefore has no application for conveyor starting.

Before leaving A.C. motor drives it is important not to overlook the fact that conventional motor protection systems do not control motor torque. Reference to figure 4 will indicate that there is no direct relationship between motor current and torque at speeds much below the motor operating speed.

2.1.5 Load Sharing With A.C. Motors

As previously mentioned, an A.C. motor delivers no torque at synchronous speed, and full torque at a speed approximately 1% below synchronous for squirrel cage motors, and 2% below synchronous for slipring motors. The actual amount of slip at full torque is a function of rotor resistance and as can be seen from Fig. 2, increases with increasing resistance.

Fig. 15 shows the load speed/curve redrawn to show the torque as a function of slip for a 4 pole "1500 RPM" motor. The actual slip at full torque is usually of little consequence to motor manufacturers, and a 1485 RPM rated motor may in fact develop full rated torque at 1483 RPM or 1488 RPM. Normally this small difference in speed at rated torque or power is also of little consequence to a machine designer.

To the conveyor designer, however, it is a matter that requires consideration, since with multimotor drives the motors are effectively coupled together and must all operate at the same speed for equal pulley sizes.

Two torque/slip curves are shown in Fig. 15 for each of three different motor applications, namely, an A.C. squirrel cage motor, an A.C. slipring motor with ring mounted shorting gear and an A.C. slipring motor with the rings connected by cable to a shorting contactor. Each set of curves show the torque/slip curve for a motor which develops full load torque at 1 RPM above and below nominal.

Firstly, 1 RPM can be considered to be a closer tolerance than most motor manufacturers would work to. Secondly as can be seen from Fig. 15 the motor torques for the squirrel cage motors in this case would be 88,5% and 117,5% at nominal speed, so the two motors would share full load in the ratio 114/86. In the slipring motor cases the ratios would be 108/92 and 104/96 respectively.

It is evident that the cable connected shorting contactor case has the lowest discrepency, due to the highest rotor resistance, and the squirrel cage motors, the highest.

If the motors are conservatively over-rated, the overload condition on the one motor should not give rise to motor problems and would be partly compensated for by increased coupling slip, if couplings are used. However, the coupling associated with the overloaded motor will get appreciably hotter than the other, which in turn is self compensating, provided the speed mismatch is not too severe.

Where fluid couplings other than scoop-controlled couplings are used, the usual outcome of motor power mismatch is that maintenance personnel attempt to correct the situation by increasing the fluid in one coupling, invariably over compensating. They then increase the level in the other until once again the original situation is reached. Finally both couplings are totally filled and the power mismatch is worse than ever. However, by this stage, the maximum torque the coupling can transmit may have increased well above the designed torque. This problem is exacerbated if the designer allowed generous thermal capacity in the coupling by oversizing the coupling in the first instance.

Mismatching with slipring motors can be corrected by introducing fixed resistance in the rotor of the higher loaded motor, which reduces the motor efficiency by a small and relatively insignificant amount.

It is fairly rare for motor mismatching to cause major problems other than those discussed above with new, equally rated motors. However, if the motor ratings are not equal or the original motors were rewound, major problems can result. In these cases the only feasible solution is usually to force the overloaded motor to operate at a higher speed by decreasing the pulley diameter.

In one such instance with unequal motors of 80kW and 120kW rating, the motor power ratio at full load was 140/60, and was corrected by increasing the pulley lagging by 3mm in thickness. This problem was only identified and solved after roughly 2 years of continuous motor or coupling overloading.

From the above it is obvious that relagging one pulley in a dual drive configuration could result in a torque mismatch if the lagging on the other pulley is a few millimetres thinner.

Torque and power matching with scoop controlled couplings can be readily achieved by adjustment of the scoop travel. However, thermal overloading of the coupling must be watched for.

Automatic compensation by means of electronic controllers can also be used with scoop controlled couplings, and this can be satisfactorily used to ensure the correct torque application during acceleration.

Since the protection system is designed to prevent thermal overload of a motor, the protection will not trip the motor instantaneously as the current increases above full load. Two situations therefore arise where the maximum motor torque can be applied to a conveyor, namely, during start up, and when additional load on the system causes the motor to operate at below designed full load speed.

In the absence of a properly designed torque limiting device, such as a fluid coupling, this high torque could prove disastrous in a conveyor system.

2.2 Torque Controlled Drives

A number of electric drives are available with which the maximum torque does not exceed the rated torque by an unacceptable margin. These drives can be loosely classified as;

(a) those in which the motor is incapable of producing high maximum torques and;

(b) those in which the torque can be computed through simple electrical measurements.

Synchronous motors fed from variable frequency supplies are examples of the former, whereas A.C. motors fed from variable frequency supplies and D.C. motors fed from variable voltage supplies are examples of the latter.

A benefit of the above torque controlled drives is that they are capable of operating indefinitely at high efficiency at virtually any speed, provided that sufficient ventilation is available.

It is therefore possible to reduce the torque to a value which is adequate to accelerate a conveyor at a slower rate than would normally be acceptable for the earlier described systems.

In addition, the ability to rapidly measure an increase in load torque allows for corrective action to be taken before the conveyor reaches an overloaded situation. This alleviates the need to design a drive with sufficient thermal capacity to start and run an overloaded conveyor, with the consequent increase in motor torque and belt, gearbox and pulley rating.

A further benefit of these types of drives relates to the transient tension waves mentioned earlier. Since the transient tensions are directly proportional to the change in rate of acceleration, the drives can accelerate the load using a smooth "S" speed time curve. Similarly during a normal stoppage, the conveyor can be powered down on a mirror image speed curve. It has been our experience, however, that provided the rate of acceleration is reduced to below 0,05 to 0,1 m/sec^2, depending on application, the transient tensions set up in the conveyor are negligible in comparison with the normal operating tension.

The speed ramp can also be simulated by generation of a simpler $\sin^2 wt$ curve which provides satisfactory results. (See Figure 5).

2.2.1 Continuous Operation At Reduced Speed

As mentioned above, all of the torque controlled electric drives can operate continuously over a speed range of 20 to 100% of rated speed or better. The potential benefits of operating conveyors at varying speeds will therefore be examined.

It is a well known fact that between 80 to 100% of the power used for material transfer on long overland conveyors is required to overcome friction, drag, scuffing, belt cover deformation and related effects. The drive torque required to overcome these losses is largely independent of speed, which means that a reduction in speed of the conveyor belt will result in a similar reduction in power consumption.

In conveyor applications where the transfer rate of the conveyor will vary considerably, substantial power savings can result from running the conveyor at a load related speed. It has been the author's experience that very few conveyors are operated at close to 100% loading except for during relatively brief periods of time.

Furthermore, the facility is available to use a higher speed of operation to cope with any occasional high transfer rate, with negligible affect on component life. In general, higher power drives will be required in this case rather than wider belts, which in larger conveyors could be the cheaper alternative.

A further potential benefit will be the proportional reduction in operating cycles of all components when the conveyor is operated at a load related speed.

It is necessary to be aware of the possibility of the conveyor operating at a speed which gives rise to resonance in the return belt. The problem can be alleviated by changing return idler spacing in affected regions or by preventing operation at critical speeds.

2.2.2 Choice of Variable Speed Drives

Essentially, three classifications of variable speed drives will be considered, namely, variable frequency A.C. drives using thyristor or GTO inverters, variable voltage, thyristor controlled DC drives and Ward Leonard systems.

2.2.2.1 Thryristor Supplied DC Drives

The speed of a D.C. drive is virtually proportional to the applied D.C. voltage.

The voltage of a Thyristor controlled DC drive is varied by delaying the gating or "switching on" of thyristors fed from an AC source. As a result of the delayed gating, the power factor of the drive varies in direct proportion to the speed, approaching unity at full speed and a power factor of approximately 0,5 lagging at 50% speed. Unless a system is used to correct the power factor of the drive, any potential cost savings as a result of reduced power consumption, are negated by virtue of the low power factor. This is particularly the case in those regions where Escom levy a maximum demand charge based on kVA demand.

In order to benefit sufficiently from the power factor control equipment, it should be capable of reacting rapidly and continuously to changes in power factor. This system is more expensive and sophisticated than the more conventional, but less flexible, switched capacitor bank type of control.

The DC variable speed drive system requires a relatively simple thyristor rectifier. On the other hand the motors are more expensive than a similarly rated AC motor and require more maintenance.

In simple terms, the torque of a DC motor is proportional to the product of armature current and field current. It is therefore possible, by relatively simple means, to measure and control the motor torque. Unlike the variable speed AC motor, the motor torque is not subject to pulsations at lower speeds and the motor and load can be smoothly accelerated from rest to full speed.

2.2.2.2 Variable Speed A.C. Drives

Variable speed AC drives consist essentially of a thyristor controlled or diode rectifier, similar to that used for a DC motor, followed by a thyristor or GTO switched invertor that converts the DC back into a variable frequency AC supply.

The conversion of DC to AC can be done by switching the DC at a relatively low frequency, related to the supply frequency of the motor, or at a high frequency to simulate a lower frequency supply. The latter system is at present confined to smaller drives. The effect of the two different systems on power factor is that the low frequency system suffers from similar power factor problems as are experienced with the DC drive, whereas the high frequency system operates at a relatively high power factor.

For most conveyor drive applications the low frequency system will be used by virtue of the size of drive and will therefore require power factor correction.

A potential disadvantage of some of the AC drives is that at low speeds the motor acts like a stepping motor. The resulting torque pulses have got to be taken into account in the design of the gearbox and high speed coupling. If residual torque pulses are transferred to the conveyor, and the frequency of the pulses coincides with the natural frequency of the conveyor belt, it is possible for unacceptably high transient tensions to be generated in the belt.

2.2.2.3 Ward Leonard

This "old fashioned" method of DC motor control can still have it's uses despite some operational and cost disadvantages.

In Ward Leonard control, a DC generator is used as a power amplifier in which low power generator field circuits are controlled, giving rise to high power armature output which is fed directly to the DC motors.

The DC generator is usually driven by means of a synchronous motor which although more expensive than an AC induction motor has some beneficial characteristics.

The control of the DC field of the generator is accomplished by means of a thyristor controlled rectifier, although other methods have been used.

The drive control equipment used for the above will usually be identical to that used for static rectifier fed DC motors.

The major difference between the controlled rectifiers used for Ward-Leonard and rectifier fed DC motors, is in the size or rating of the components used. However, in operational terms the latter generates harmonics which can seriously affect the power supply, whereas the former has a negligible effect on the supply.

The presence of "power" harmonics can result in filters being required to reduce the effect of the harmonics on the power supply.

In addition, the power factor of a rectifier fed DC motor is virtually the same as the ratio of motor operating speed to maximum speed, as mentioned above.

By contrast a Ward-Leonard device can operate at a leading power factor when the DC motor is operating at reduced load. By operating at a leading power factor, the drive can compensate for other equipment operating at lagging power factors.

The present Escom tariffs place a high cost on maximum demand, to encourage consumers to reduce the maximum demand as far as possible. The cost per kVA of maximum demand at the time of writing was over R15 per kVA in those regions in which demand is measured in kVA.

The value of every kVA saved by operating the synchronous motor at a leading power factor is fairly substantial. Since many conveyors operate at reduced load for a fairly substantial period of time, the potential value of the leading power factor can more than offset the reduced efficiency of the Ward-Leonard (refer to App A) drive.

This type of drive will only be feasible for large installations.

3. CONVEYOR SYSTEM DESIGN

As mentioned previously the designed operating full load tension in individual conveyor cords can be somewhat lower than those experienced in practice owing to:

(a) Belt transitions, vertical and horizontal curves, and belt turnovers.
(b) Drives designed for overload conditions.
(c) Excess drive torque due to motor or drive characteristics.
(d) Transients.
(e) Take up winches.

3.1 Effect of Drive Systems on Factors of Safety

To illustrate the effect of particularly the drive parameters, a conveyor design meeting the following requirements is shown:-

Peak tonnage 2500 tph, belt length 800 metres, lift 200 metres, lump size - 300mm, minimum bulk density of product $1000kg/m^3$, maximum bulk density $1200kg/m^3$, belt speed 4m/sec. A vertical curve of length 70 metres and radius 250 metres is required immediately behind the drive position. A 2 step tensiometer controlled take up winch is to be installed.

The results of possible designs based on these parameters are indicated in Figures 9 to 11. The assumptions and ancillary calculations used are shown below.

In the calculated results the following must be pointed out:

(a) The increase in edge tension as a result of the vertical curve has been calculated using Dr Oehmens's theory for calculation of edge tension effects. (Ref (3)).

(b) The motors are sized on the basis of a 90% gearbox efficiency but in calculating the torque a figure of 95% has been used. A thermal allowance of +10% has been allowed over and above the motor rating based on 90% gearbox efficiency. The motors are thus 16% over capacity.

(c) The directly coupled squirrel cage motors will <u>always</u> deliver maximum torque during a stage of the start up and each motor will deliver the maximum torque simultaneously with others.

(d) The slipring motors are being operated in a controlled torque system but one of the motors is out of control. This can happen if the motor remains shorted out at standstill, the resistor switching or solution strength is faulty or the control system fails. It is assumed that only one motor fails. In the case of a single motor drive the results will be virtually identical to that of a single squirrel cage motor except that the condition will only arise if the control system is faulty.

(e) The fluid coupling drives were assumed to be able to transmit 160% of full load torque, based on motor full load torque, in the case of non-controlled drives, and based on system full load torque in the case of scoop-controlled couplings.

(f) For an AC variable speed drive, the torque control system is assumed to limit the torque to 140% based on actual required torque. It could be possible to improve on the degree of control.

(g) For a DC variable speed drive, the torque limit is taken to be 120% of actual required torque. More precise control is possible.

For both (f) and (g) the assumption is made that two independent torque measurement systems with independent reaction systems are used. In the case of (e) independent measurement systems are possible, but a single reaction system, namely scoop control is assumed.

(h) Where torque measurement is unavailable the motors are sized to allow for starting of a flood loaded conveyor, and the conveyor is sized to allow the correct transfer rate at the lower bulk density, but the drives are sized for the higher bulk density. This results in drives which are rated at 160% of the full belt load rating. In this instance the conveyor belt is rated for the operating tension at the higher drive power. If the belt had been rated for nominal tonnage the factors of safety shown in red would have resulted.

(i) The belt rating was calculated using conventional accepted design methods with a factor of safety of 7 related to the maximum operating tension. Certain design procedures assign factors to the motor torque depending on the type of drive chosen. These factors were not used in the calculations.

It is of interest to note that the maximum tension to which the conveyor belt will be exposed in the worst case is 3,8 times the value for the best case.

The contribution to the total tension of the various constituent tensions is coded as shown in Fig. 9. It should be pointed out that the return belt tension was assumed to be constant for all configurations. Ideally the tension should increase slightly for single motor drives, but since the increase in tension would be small, it was disregarded for the sake of the calculations.

The results indicate firstly that the practice of using the operating tension as the basis of rating of the conveyor belt, while disregarding the drive characteristics, has a large effect on the de facto factor of safety under starting conditions.

The values chosen for maximum torque can be debated at length and the intention is not to open such a debate. The intent is to indicate that the maximum torque applied to a conveyor is not defined by motor rating or belt capacity but is dependent on the drive system.

It is interesting to note that the de facto factor of safety relative to maximum predicted tension in the belt varied from 1,12 to 4,31 and from 0,7 to 2,67 where the full load tensions are calculated on the basis of flood load tonnage and design tonnage at maximum bulk density respectively.

It is patently obvious that disregarding the drive characteristics can lead to over-or underdesigned systems.

This company has, following discussions with Dr Harrison, decided to base the belt rating on a de factor of safety of 3 relative to the maximum calculated tension in the conveyor, as shown in Fig 10. The de facto factors of safety as shown in Fig. 10 vary from 63% below to 44% above the proposed factor of safety, depending on the drive characteristics.

Fig 11 indicates the belt rating which would be required to achieve a de facto factor of safety of 3 for the various drive alternatives.

An assessment of the cost implications of providing more sophisticated, costlier drives with improved operational control must be weighed against the other alternatives of cheap drives with commensurate increases in belt rating.

In addition the efficiency of the drive systems and the impact such systems make on the electric power bill should also be assessed in the case of major installations, since the power cost associated with a major conveyor can be as high as R0,5 million to R1 million per annum.

3.2 Factors to be Considered to Optimise Conveyor Design

Although most of these factors were considered earlier, their implication on conveyor design will be considered.

3.2.1 Belt Rating

Traditionally, belt ratings are selected to ensure that the nominal tensile strength of the belt is 6,7 to 7 times that of the operating tension. The splice stength is assumed to be at least that of the belt.

DIN 22101 specifies that the safety factors should be greater than 6,7 with reference to the splice strength and 7,44 with reference to the nominal belt strength based on operating tension. Alternatively safety factors of 4,8 and 5,33 should be used if the starting tension can be calculated.

According to the Clouth Industrie Gummi handbook "Conveyor Belting" the recommended tensions as a percentage of nominal belt strength are 12% for the operating tension, 6% for accelerating tension and 22% for "additional tensions", corresponding to safety factors of 8,5, 5,56 and 2,5 respectively.

Although the Clouth method uses a factor of safety of 2,5 based on maximum tension, we have preferred to use 3 which makes allowance for unequal tension distribution in individual cords and provides an adequate margin above the ultimate strength of the belt.

The unfortunate consequence of the coventional methods of belt rating is that they provide no incentive to improve the design features which could contribute to poor conveyor performance. A highly competent conveyor design is likely to be more costly than one which barely satisfies the minimum standards.

To the user, the competent design might be satisfying, but he is likely to find it difficult to justify the increased costs, which to the accountants might appear to be nothing more than over engineering. A further difficulty the user's engineers experience is the fact that the shortcomings of a poor design are not immediately apparent. It is therefore difficult to use experience based assessments in the design evaluation.

Refering again to figures 10 and 11, in the present circumstances it is possible for a tenderer to offer a large conveyor equipped with squirrel cage motors without fluid couplings and to use the same belt as would be used if say, well engineered D.C. motor drives were used. In the one case the belt is badly underdesigned and in the other case is overdesigned. The ratio of price is however likely to be as high as 1,5, 2 to 1.

If however the de facto safety factor of 3 was used, the former conveyor was equipped with a ST12000 belt and the latter with a ST3150 belt the overall cost is likely to be similar.

If additionally the structural design was to further allow for the head pulley total tension being 7,6 times higher in the former case, and the gearboxes and other mechanicals were rated accordingly, the D.C. motor application could be appreciably cheaper.

In addition I would recommend that consideration be given to a figure of splice efficiency which is reasonably achievable. The splice efficiency of steel cord conveyor belts is theoretically higher than 100%, but whether this figure is repeatedly achievable in practice is debatable. The complexity of the splice, conditions under which the splice is carried out and the splice table design features, such as the method and efficiency of pressure and temperature control, should be considered.

Furthermore, the effects of splice failure should be considered. Long inclined conveyors situated in a mine shaft, as an example, present possibilities of loss of life, and major downtime in clearing of product and belting in the event of belt failure. Damage to adjacent conveyors or other structures is a further possibility. As an example the conveyor which is to be installed at our Majuba Colliery, will contain 250m tons of coal and has a total belt length of over 3000 metres. The consequences of most of the coal and belting ending up near the shaft bottom are obvious, and reclamation work could take weeks. The splice efficiency we selected for this application was 75%. On overland conveyor installations where the above considerations do not apply and good splicing conditions are obtainable, we use an efficiency of 100%.

With the modern methods available to non-destructively test belts, the splice performance must be a major consideration. The non-destructive testing of splices has not developed to the same extent as that for belts. It is prudent to confirm that the belt operating tension is below the tension commensurate with the dynamic splice efficiency for the selected belt class.

3.2.2 Product Feeders

Any belt conveyor user has encountered the never ending problem of conveyor overloading.

The belt width is selected to provide freeboard for prevention of spillage and/or material lump size. To those not acquainted with this rationale, however, the freeboard appears to indicate under-utilisation. This problem usually raises it's head during periods of reduced supervisions such as night shift.

If the drives are not rated to cope with flood-loading, the repeated trip-outs invariably lead the frustrated electricians to increase motor overload settings, resulting in frequent motor burnouts. This problem in turn might be "solved" by installing larger motors, often with the mistaken belief that the motor overloads will prevent excess torque and belt tensions.

Alternatively, the user's Engineers might require the conveyor to be designed for a flood loaded condition. This can of course lead to the conveyor rating increasing by up to 75%, with the required increase in rating of conveyor mechanical components.

Considering the cost of either of the above options on large conveyor systems, it is not difficult to justify the cost of a good feeder control system.

Ideally the feeders should be designed to deliver no more than the designed tonnage. This is of course easier said than done, particularly if silo or bunker design necessitates the use of an excess number of feeders to ensure adequate material recovery from the storage system.

It is however possible to use a "closed loop" control system which measures the material flow rate and adjusts the feed rate accordingly.

To prevent continuous "hunting" or surging of the control system, the measuring device should be situated as near to the feeders as possible. Problems in this regard can be experienced with long storage systems where the furthest feeders may be a fair distance from the belt weigher. Careful design of the control system can partially compensate for this problem.

However, whether or not the control system can cope with the feeder layout, an override control system activated by the drive motor load sensing device is advocated. The utilisation of modern micro-processor based controllers allow a variety of actions to be taken on sensing an overload. In addition, it is possible to predict an eminent overload by monitoring the rate of increase of the load.

The reaction to a potential or real overload can be any combination of;

automatically reducing the controller set point,

sequentially stopping the feeders,

stopping the conveyor prior to an unacceptable overload occurring.

The overload sensing devices should be matched to the load control tolerance required, but relatively accurate load sensing is possible with all types of drive.

It is essential to ensure, however, that the feed rate controller cannot be tampered with or overridden by the operating staff.

The micro-processor controller mentioned above can provide a number of functions other than that of load control. The controller can monitor the drive system, provide fault diagnostics and can generate the speed/time characteristic for acceleration and deceleration.

3.2.3 Ability of System to Transmit Torque

It might be argued that the calculated maximum torque/tension figure of over 400% in the case of squirrel cage motors, or an out of control single slipring motor drive, does not take account of drive pulley slippage. However, it is normal to design the angle of wrap based on a fairly conservative coefficient of friction between the drive pulleys and the belt. Secondly, the T2 tension depends on the belt tensioning system used. A T2 tension of 110% of designed tension is likely in practice, particularly when winch take ups are used.

Fig 12 indicates that an effective tension of approximately 250% of the design value is possible for a single drive and 500% for a dual drive since the coefficient of friction on a clean dry belt may be as high as 0,55. No account of an increase in T2 tension is included in the above figures, thus a simple winch control system, particularly if the speed of operation of the winch is high, will increase the possible effective tension. If drive systems with high applied torques are used, a high speed take up winch will probably be required to compensate for rapid belt stretch. The recipe for disaster in such cases is therefore considerably compounded.

3.2.4 Measurement of, and Reaction to, Belt Slip

An additional protection measure against excess torque can be provided by a well designed belt slip monitoring system.

The cheapest, and most commonly used method of belt slip detection is a mechanical speed detector similar in design to a governor. The accuracy of these devices depends on the accuracy of the return spring characteristic, governor masses, drive wheel diameter and hysteresis effects caused by friction in the device. Furthermore it can only be set to operate at a particular speed and therefore is out of operation during the acceleration period. The required setting must also take account of changes in operating speed resulting from changes in conveyor load.

It is therefore unlikely that such a device can be set to closer than 90% of the operating speed. Partial slippage can occur, and due to the lack of sensitivity and response of the mechanical system, the device is unable to detect these partial slippages. With high power drives partial slippages can cause heating of the pulley and damage to the pulley lagging, and can increase fire risk.

Finally, belt slip is most likely to occur at start-up due to the higher torques involved, and monitoring during this period is highly desirable.

Devices are available which measure and compare the peripheral speed of the drive pulley or pulleys and the belt speed. Since these devices can operate from virtually standstill, it is possible to detect slip during the acceleration period. A small degree of belt slip druing run up and full speed operation can therefore be detected and acted upon.

If drive systems incorporate an external control means, for example scoop controlled couplings, slipring motors or variable speed drives, the applied torque can be reduced on detection of slip and the conveyor can be stopped under controlled deceleration. Additionally, the use of a fault diagnostic system can indicate that the conveyor was tripped due to belt slip, enabling the maintenance staff to investigate and correct the cause of the slippage. It can thus provide indication of a fault in the drive system torque control mechanism.

Unfortunately, since the actual friction coefficient has a fairly marked effect on the effective tension, this method can only protect against a substantial over-torque condition arising. It would however support the argument that the take up winch, if used, should be designed to suit the designed conveyor acceleration. Any large increase in acceleration rate would result in a rapid decrease in T2 tension and promote the possibility of belt slip.

It follows that any designer deciding to use this method of over-torque protection would need to have a lot of confidence in the belt slip device, which should in that event be tamper proof.

3.2.5 Splice Life

Research carried out by the "Institut für Fördertechnik und Bergwerksmachinen" of the University of Hannover has demonstrated that splice life is a function of operating tension.

A short spliced conveyor belt was run on a test stand consisting of a driven pulley and a non driven pulley and the tension was applied by hydraulic cylinders. The tension was cycled between predetermined limits and splice life was determined.

A typical plot of life against maximum tension, plotted as a percentage of nominal belt strength, is shown in Fig 13.

The horizontal asymptote is defined as the dynamic splice efficiency. This figure varies not only with type of splice but also tends to be lower the higher the belt rating.

Typically an ST5000 belt has a dynamic splice efficiency of approximately 40% of nominal belt rating.

3.2.5.1 Relationship between Splice Life, Wrap Angle and Tension

It is felt, however, that if the belt is rated at 3 times maximum tension, the dynamic splice efficiency criterion should automatically be met. Should this not be the case, the calculated tensions should be checked to confirm the calculated maximum tension is correct. Certain special designs could result, however, in the dynamic splice efficiency being the deciding factor for belt rating.

Some manufacturers firmly believe that the splice life depends on the number and severity of high tensile bends the belt undergoes and not purely on the load cycles. This view is supported by some members of staff of the University of Hannover.

A reduction in the degree of high tensile bending requires an increase in operating tension, and other manufacturers believe that the increase in tension overrides the benefit of reduced total wrap angles.

Empirical methods exist to calculate a drive tension - wrap angle factor, which takes account of the mean tension - wrap angle product at each pulley (see fig. 14), but the factor has no physical meaning. To my knowledge no research has been aimed at determining the relationship between tension, overall wrap angle and splice life, and such research could provide the means to optimise drive design and remove this existing "grey area".

3.2.6 Belt Construction

During the Beltcon 3 conference a video recording of x-ray inspections of various belts was screened. This video clearly indicated a tendency for the pitch of the steel cords to vary considerably. It was also obvious that a periodic displacement of the cords occurred which was most marked at the edges of the conveyor.

This phenomenon is usually caused by the side thrust imparted on the cords by the pressure on the rubber during the belt manufacturing process. At each end of the table the cords are prevented from moving laterally whereas they are prevented from moving within the table purely by the applied pretension.

It is important, however, to ensure that the lateral movement is restricted as far as possible and to ensure that the cords are equally and adequately pretensioned.

An inadequately manufactured belt can result not only by bad-tracking, but also unequal distribution of tension between the cords with possible subsequent catastrophic failure.

The belt construction should not make major inroads into the factor of safety allowed.

A method of determining the tensile distribution is desirable and is under consideration by my company.

3.2.7 Power Costs

Power efficiency considerations in plant evaluation are totally foreign to the average South African Engineer. By comparison the Europeans, particularly the Scandinavians, consider power audits to be an integral part of plant design evaluation.

It has been argued that electric power costs in this country are comparitively low, which was the case 15 or so years ago. Since then the average annual compound increase in power costs is 22%, which is substantially higher than the inflation rate on most other operational input costs.

Invariably if a plant design is proposed with the aim of reducing electrical energy costs, this is brushed off as being "a drop in the ocean compared to the capital costs associated with the project development". A request is usually made some time after the plant is operational to investigate the electrical energy costs, because they now appear to be a significant contribution to operating costs. In one case electrical energy costs accounted for 22% of the operating costs on a particular mine, whereas on a similar mine, the electrical energy costs were only 8% of the operating cost. This difference was found to be almost entirely due to differences in plant design, which could not be corrected without major reconstruction.

It is therefore worth examining the electrical energy cost implications of a conveyor drive design. A distinction must be drawn between power savings and power cost savings, which do not necessarily mean the same thing. Plant operating at poor power factors, or alternatively, drawing large amounts of power for short periods of the day will result in much higher power costs per unit of energy than plant operating at a high power factor and consistent loading, although the actual amount of power consumed might be identical.

To illustrate the point, power costs have been calculated for a conveyor 10km long, conveying 5500 tons per hour of product. The feed and discharge points are taken to be at the same elevation and the details are shown in Appendix A.

The difference in power costs in this case stems largely from the fact that the conveyor is underutilised in the early stages. Similar conditions could arise where additional capacity is allowed for strategic reasons.

As far as the power factor correction benefits of the Ward-Leonard synchronous motor are concerned, the under utilisation of the motor mechanical power in the early stages allows the electrical capacity to be put to use.

It can be argued that the power factor correction benefits of the Ward-Leonard system should not be accounted for, since power factor equipment could be installed to provide the correction. However the cost of such a system is likely to be between R2 and R3 million, and therefore in the system evaluation this amount should then be deducted from the cost of the Ward-Leonard drive, which is likely to make it the cheapest of the 3 alternatives.

Alternately, it can be argued that the product tonnage build up would indicate that the overall power demand will show a similar build up and that the motors are supplying excess correction which cannot be utilised. Although the case for the demand is valid, it is a fact that plant running at well below full capacity exhibits a poor power factor, and therefore the amount of power factor correction which can be utilised is higher than the prorata power consumption would indicate.

In the final analysis, the maintenance and operating costs as well as other possible limitations, such as availability of staff suitably qualified to maintain the equipment, would require evaluation.

4. CONCLUSION

It is to be hoped that the foregoing considerations indicate the need for a system design approach to be adopted in the case of major conveyors.

There is no single solution for every application, and for that matter, changes in raw material, component and power costs could mean that optimal designs for identical applications might change from time to time.

Computers are widely used for conveyor design, but in many cases their use is confined to rapid calculation using the same basic formulae that were used in the days of scratch-pad calculations. A number of "rules of thumb" are still frequently used, which can today be improved on by utilising the latest mathematical techniques.

Design optimisation is difficult to perform if based on hand calculations. However, modern computers are capable of performing more calculations in an hour than a man with a hand calculator can do in a year. Conveyor system optimisation is therefore perfectly feasible.

However, a conveyor is in turn a part of a material handling system and should not be considered in isolation from the rest of the system. Installation of a product crusher may be cheaper than designing, installing and operating a materials handling system to cope with large lumps.

Total system optimisation is only possible if the party responsible for such optimisation possesses all the necessary facts, which may rarely be the case as far as conveyor manufacturers are concerned. In the event that the client is unwilling or incapable of supplying all the necessary information, the conveyor design may still be optimised as a system by the involvement of drive suppliers.

However, unless the characteristics of the drives are appreciated by the designers, and reasonable safety factors are used which take due cognisance of all factors, optimal system design will not be possible. This will benefit neither supplier nor user.

The analysis of all contributing factors toward the maximum tension imposed on the belt, is required. The cost of controlling these factors, can then be compared with the resultant cost savings.

Application of these methods will effectively eliminate the cheap but poor design and thus put the competent designers in a competitive position. The client will benefit by not having to attempt to justify the apparent costliness of a competent design. Finally, the installation is likely to be reliable and cost effective.

APPENDIX A

POWER COST ESTIMATES

1. EFFICIENCY

At full load:

(a) AC squirrel cage motors and scoop controlled couplings.

Motor 94%, coupling 98%

Drive 92%

(b) DC motors with thyristor drive.

Motor 95%, thyristor pack 98%

Drive 93%

(c) Ward-Leonard controlled DC drive.

Drive 86%

2. ESCOM TARIFFS

Based on kVA maximum demand.

Energy cost - 2,787 cents/kwh

Maximum demand - R14,53/kVA/month

3. AVERAGE TONNAGE PER ANNUM

Year	Tonnage (% of designed tonnage)
1	10
2	20
3	40
4	60
5	80
6	96

4. POWER CONSUMED

YEAR	AC MOTORS PLUS FLUID COUPLINGS		DC MOTORS THYRISTOR FED AND COMPENSATED		DC MOTORS WARD-LEONARD	
	KW	KVA	KW	KVA	KW	KVA *
1	3 293	4 223	619	652	663	7 000
2	3 619	4 553	1 240	1 305	1 326	7 000
3	4 266	5 203	2 482	2 613	2 653	6 693
4	4 932	5 836	3 722	3 918	3 980	6 000
5	4 568	6 437	4 963	5 224	5 306	4 867
6	6 050	6 914	5 866	6 175	6 271	3 537

5. POWER ACCOUNT (ANNUAL) R X 10^3

YEAR	AC			DC (THYR)			DC (W - L)		
	UNIT COST	MAX D	TOTAL	UNIT COST	MAX D	TOTAL	UNIT COST	MAX D **	TOTAL
1	525	736	1 261	99	114	213	106	-1 220	-1 114
2	577	794	1 371	198	228	426	211	-1 220	-1 009
3	680	907	1 587	396	456	852	453	-1 167	- 744
4	786	1 017	1 803	593	683	1 276	634	-1 046	- 412
5	888	1 122	2 010	791	911	1 702	846	- 849	- 3
6	964	1 206	2 170	935	1 077	2 012	1 000	- 616	384

TOTAL FOR 6 YEARS R10,2 x 10^6 R6,5 x 10^6 -R2,9 x 10^6

* Leading kVAr

** Negative amount relates to value of power factor correction available from the synchronous motor.

Difference in power bill

Conventional - DC Thyristor R 3,7 x 10^6
Conventional - Ward-Leonard R13,1 x 10^6

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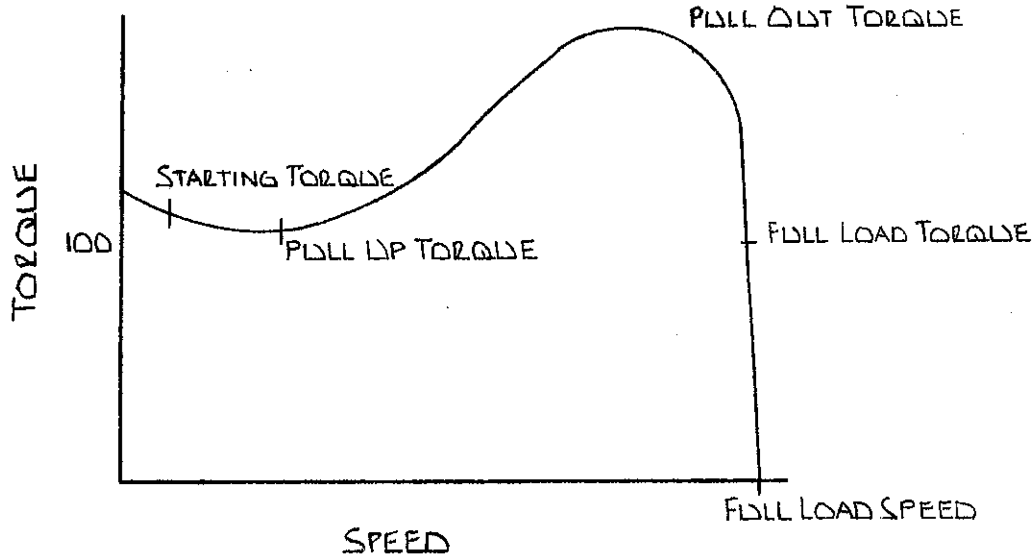


FIG 1

TORQUE SPEED CURVE OF AC MOTOR

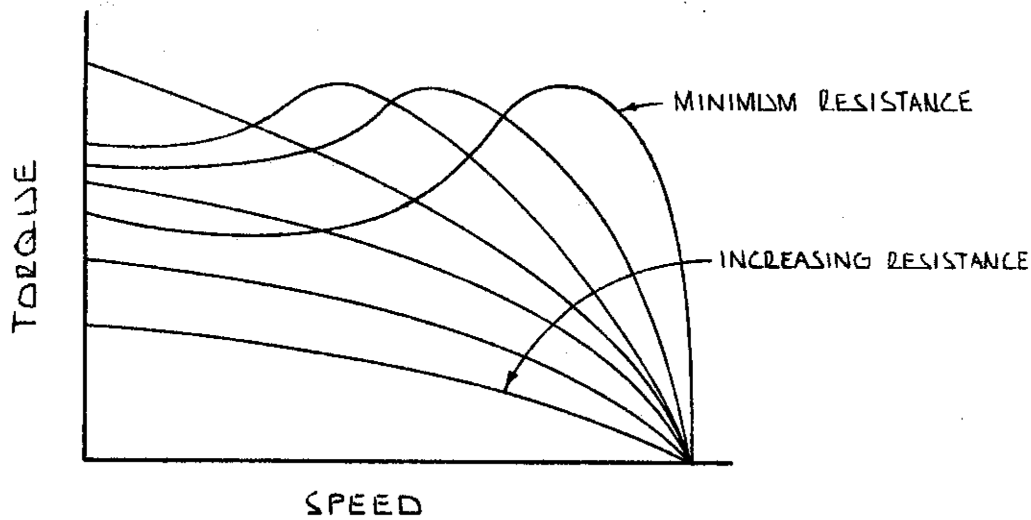


FIG. 2

TORQUE SPEED CURVE OF AN A.C. SLIP RING MOTOR

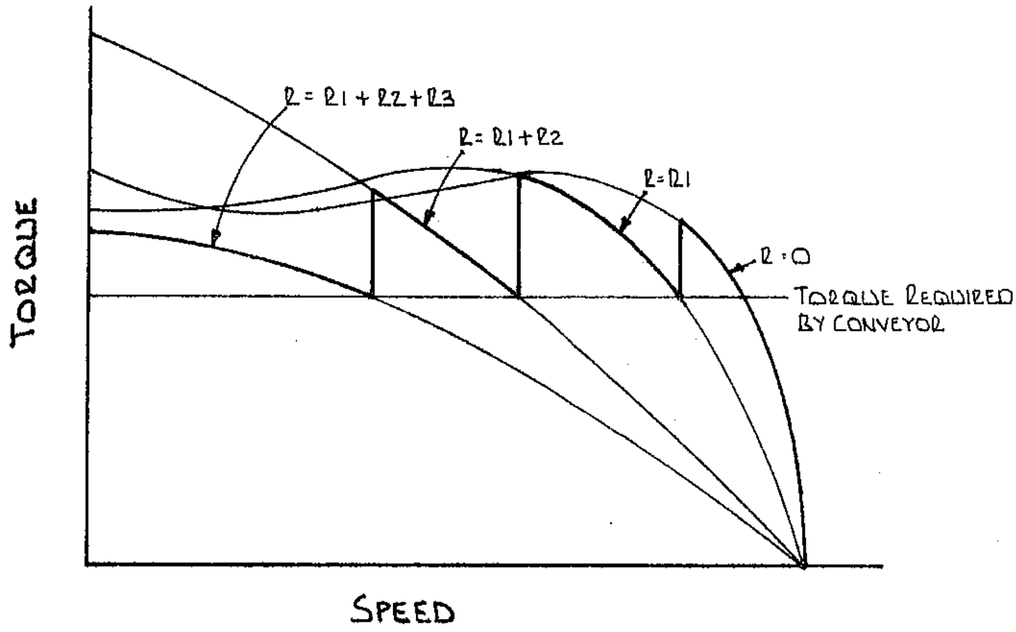


FIG 3

TORQUE PULSES THROUGH RESISTOR SWITCHING

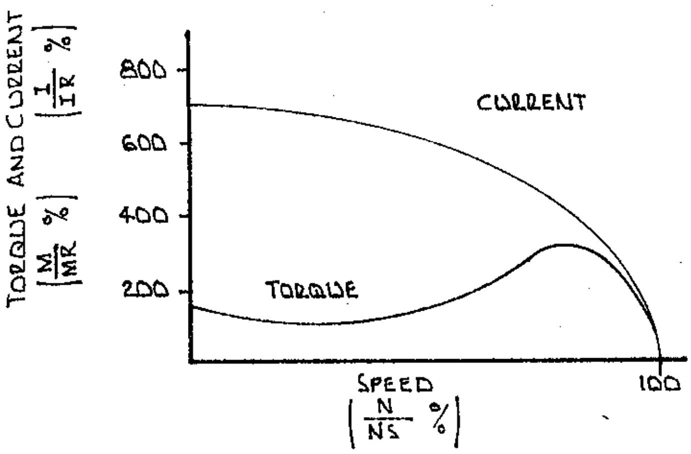


FIG. 4

CURRENT & TORQUE VERSUS SPEED FOR AN AC MOTOR

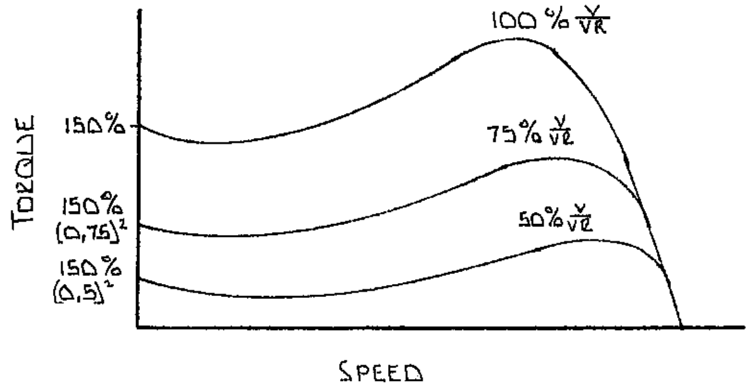


FIG 6

AC MOTOR TORQUE IN RELATION TO VOLTAGE

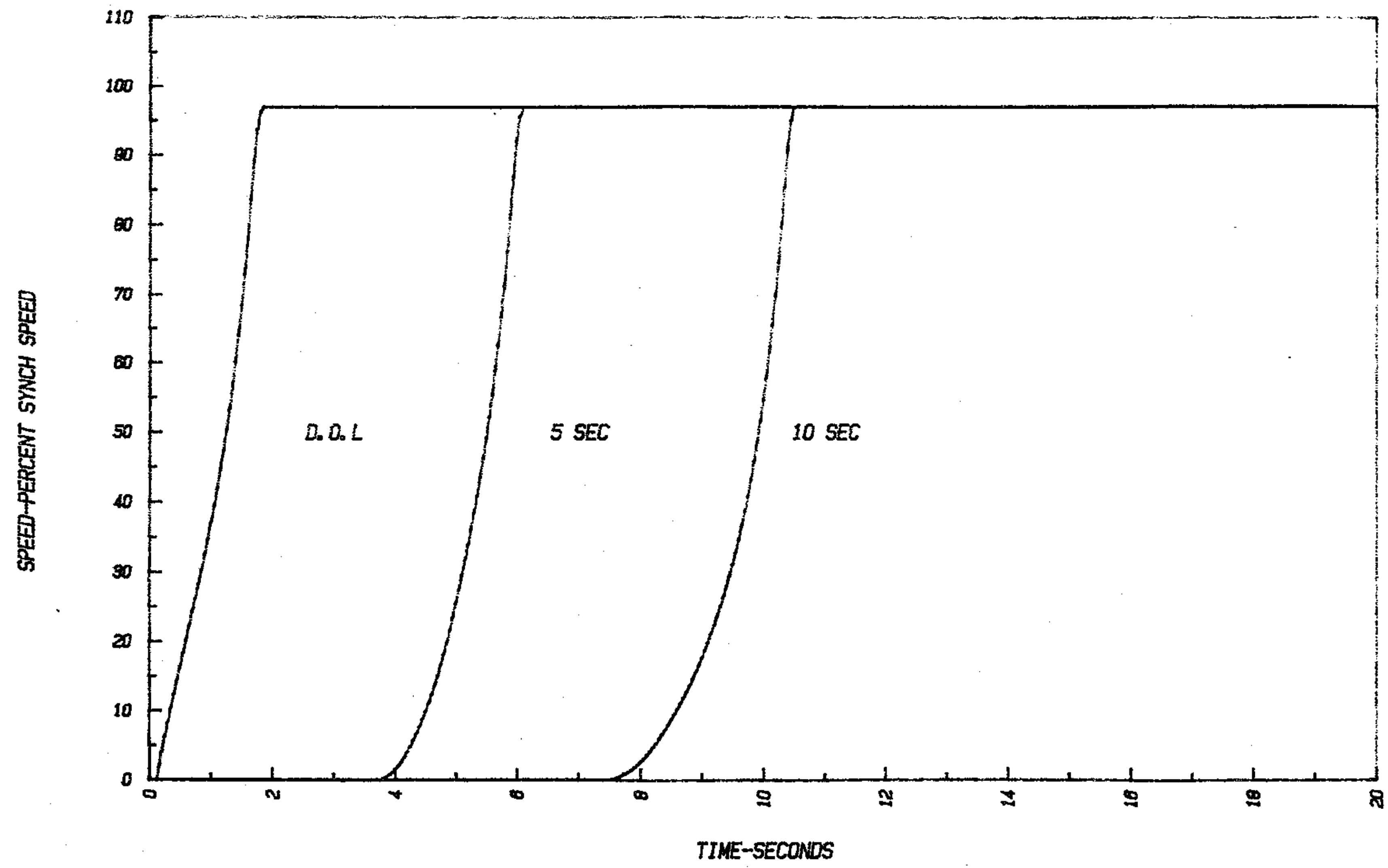


FIGURE 7a-SPEED/TIME CURVES-LIGHT LOAD

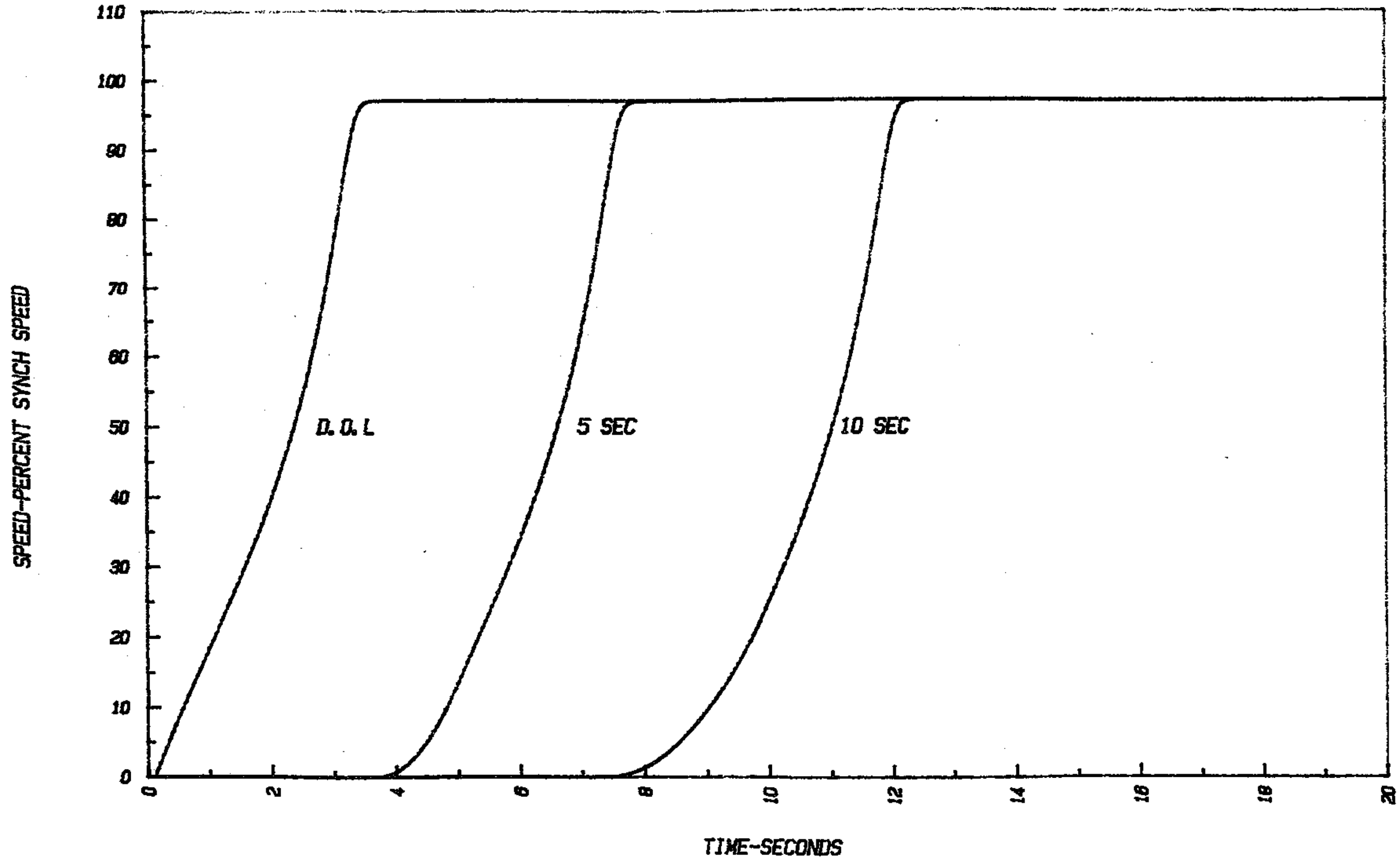


FIGURE 7b-SPEED/TIME CURVES-AVERAGE LOAD

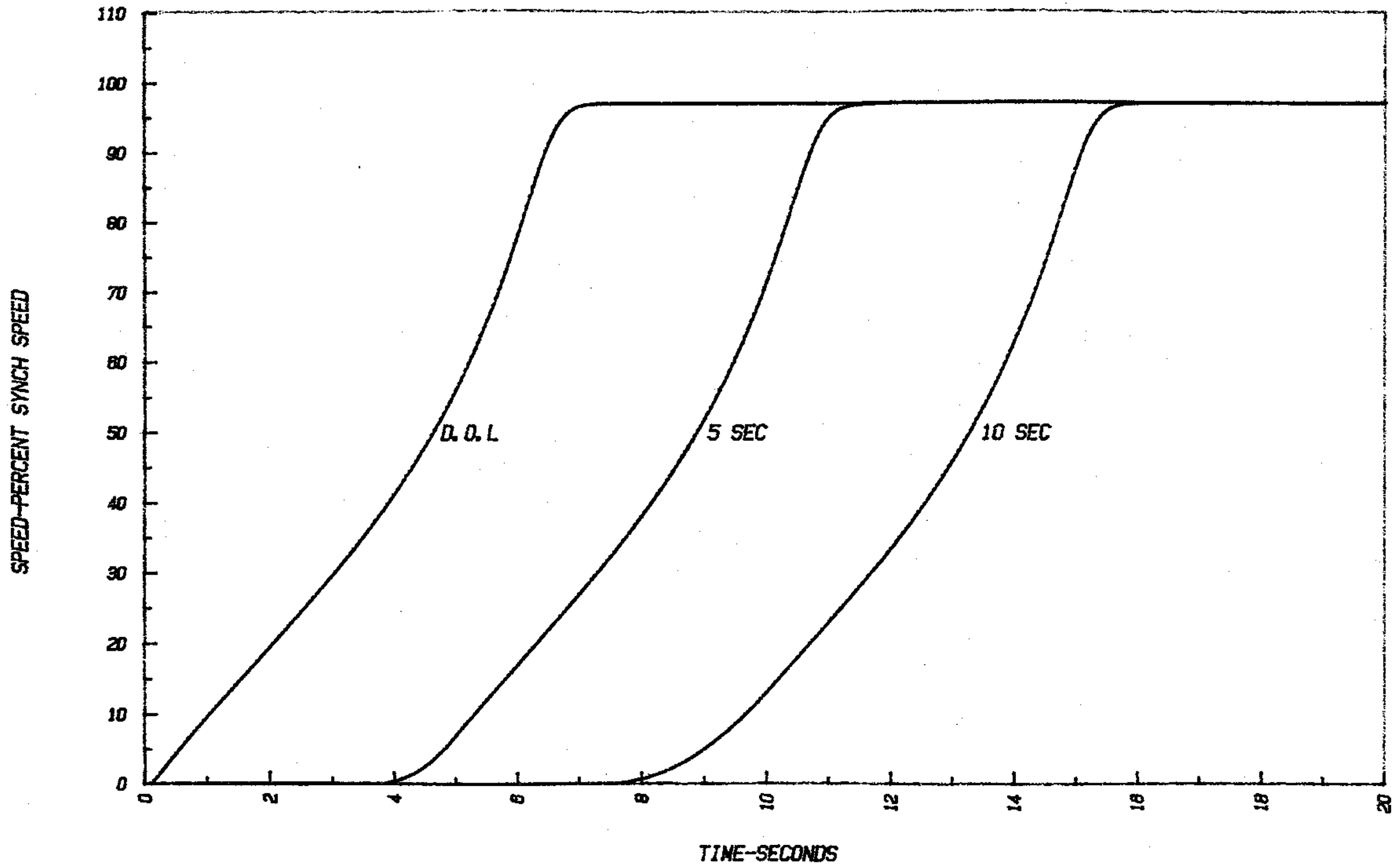


FIGURE 7c-SPEED/TIME CURVES-HEAVY LOAD

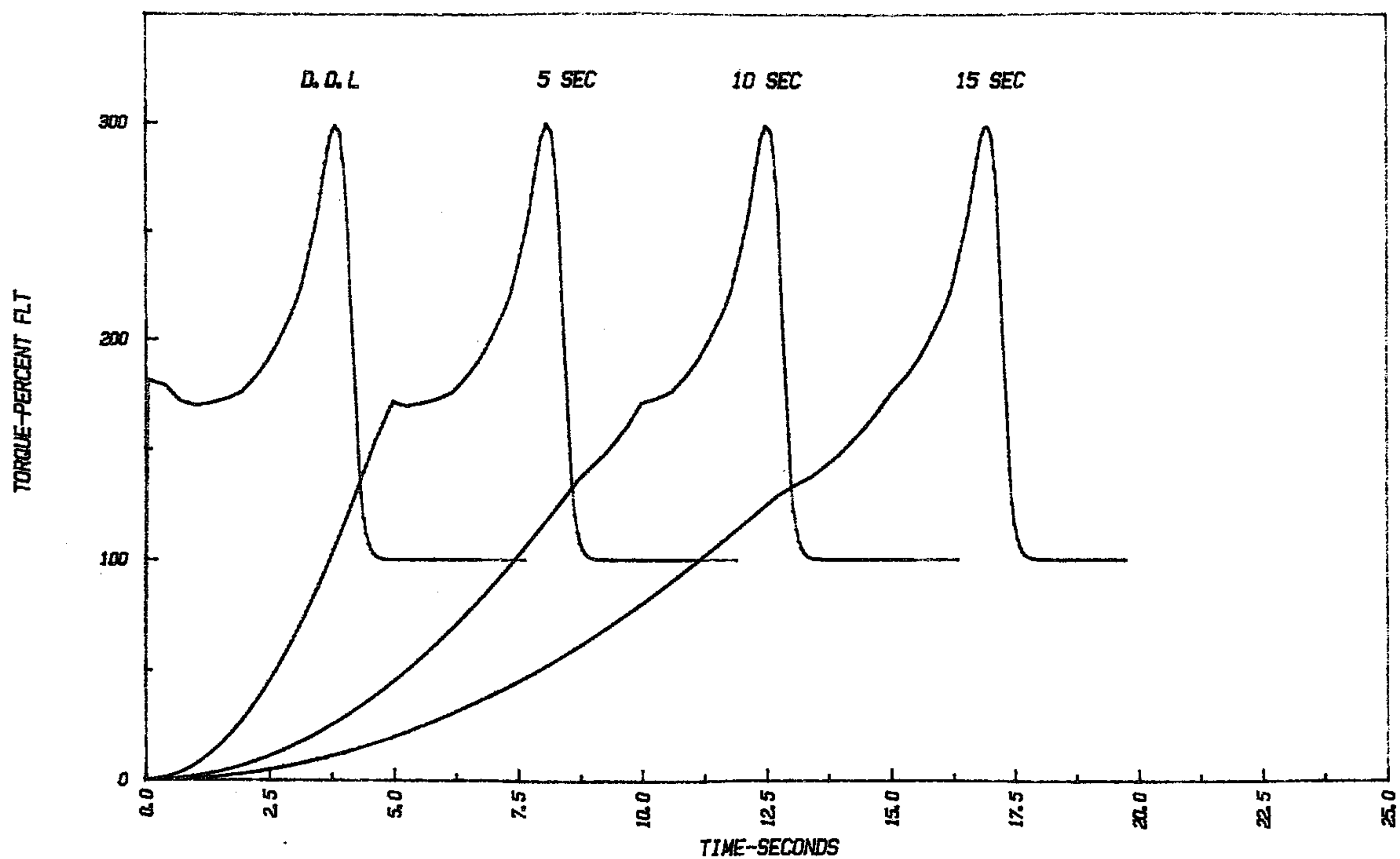


FIGURE 8-TORQUE/TIME CURVES-SOFT START

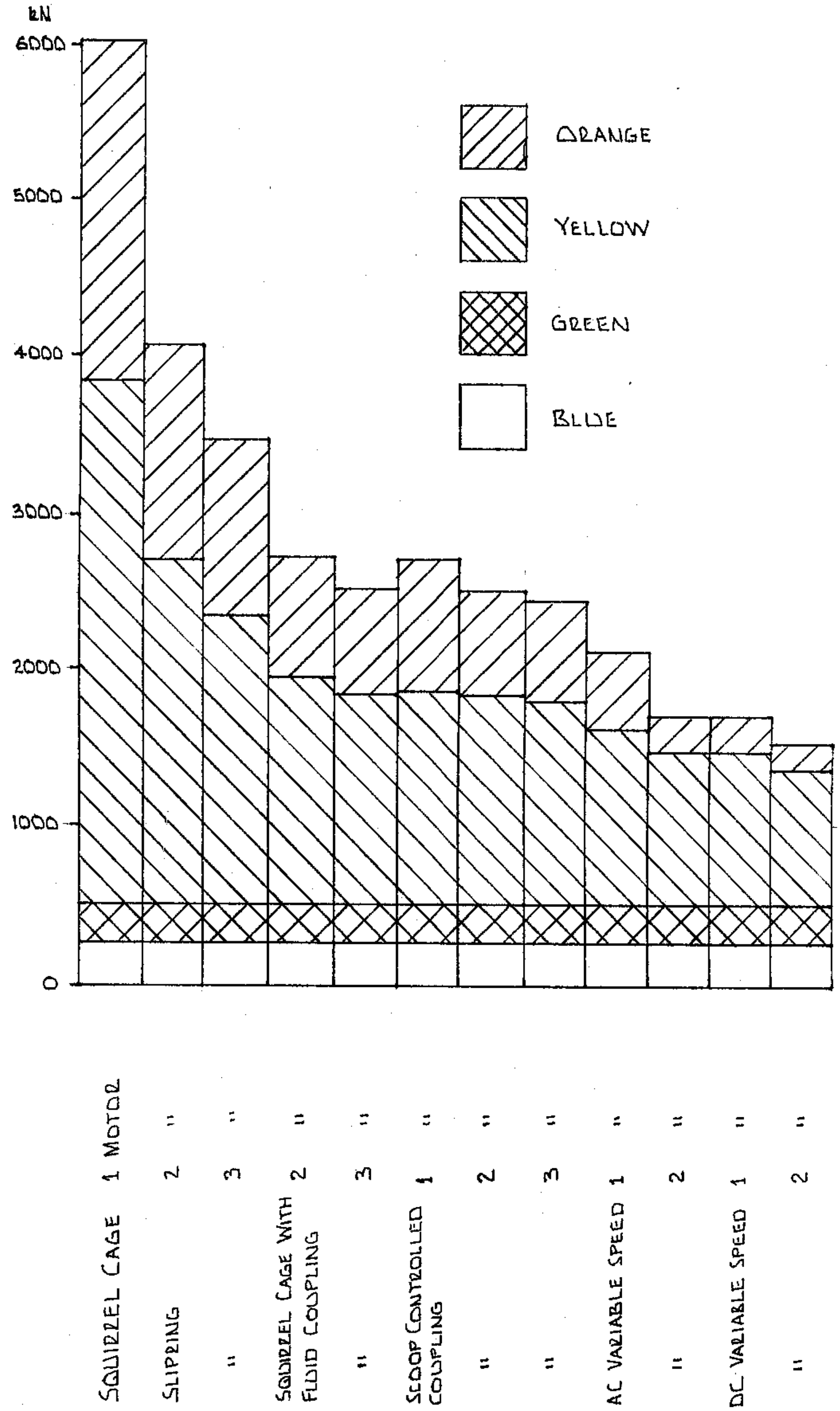


FIG 9

TOTAL BELT TENSION FOR DIFFERENT DRIVE SYSTEMS

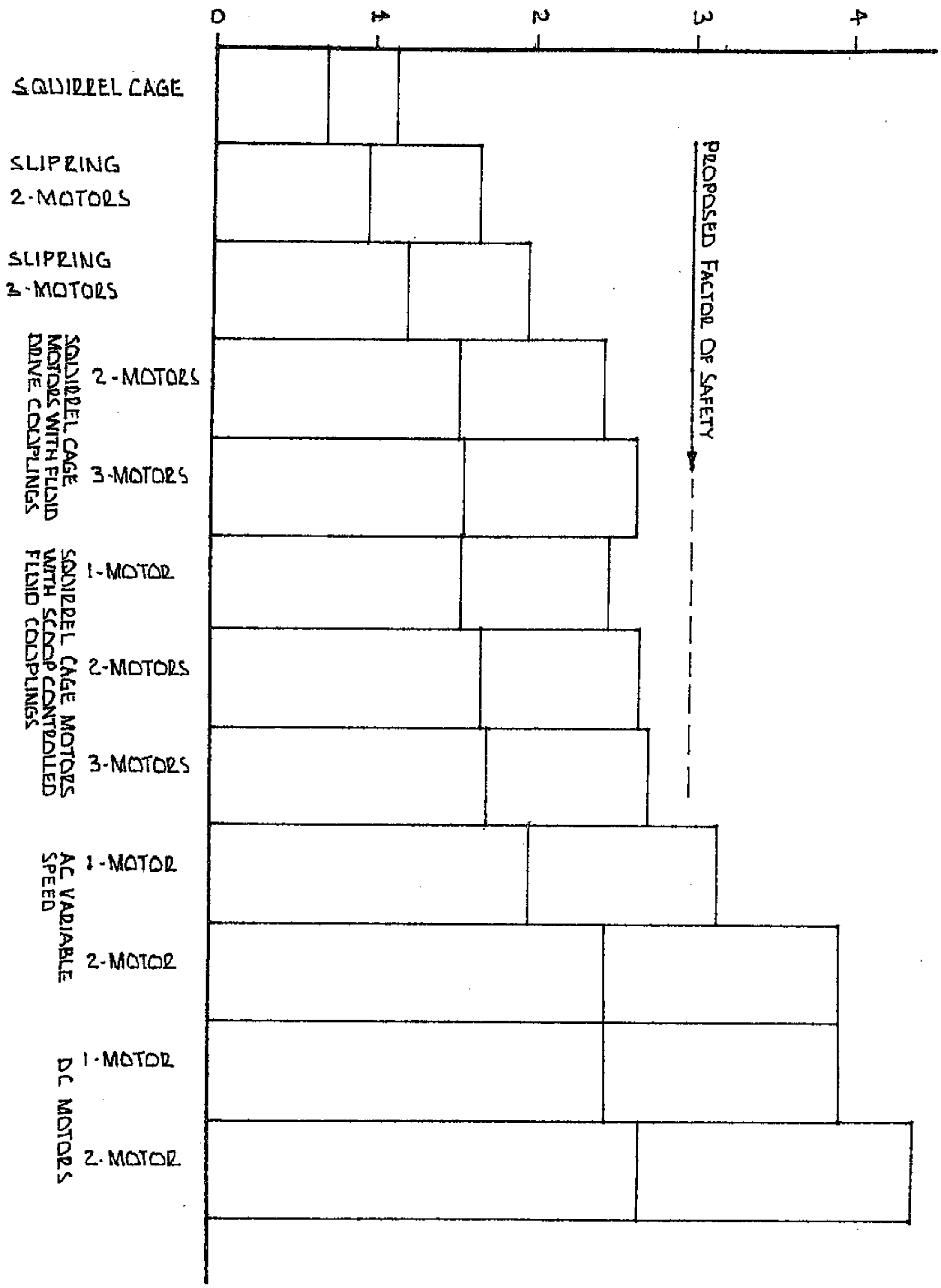


FIG.10

FACTOR OF SAFETY
(CASE 1 - 90% GEARBOX EFFICIENCY PLUS 10% ALLOWANCE ON MOTOR)

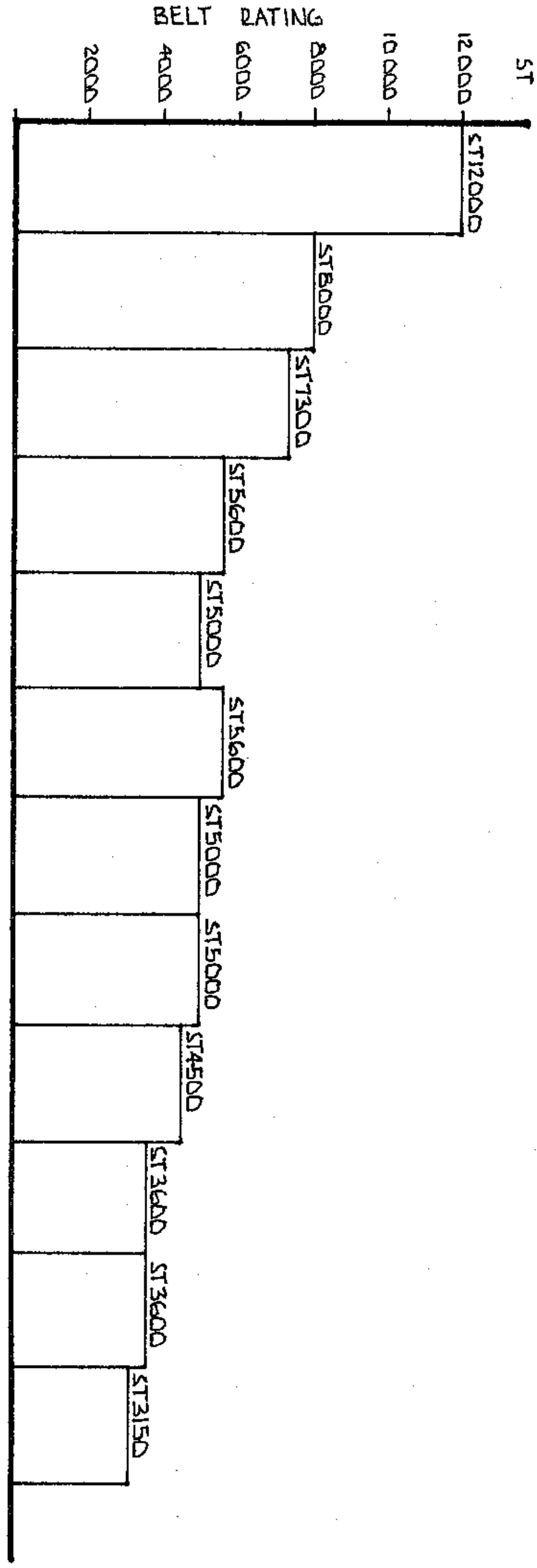


FIG 11

BELT RATING FOR F.O.S. OF 3 ON MAXIMUM TENSION

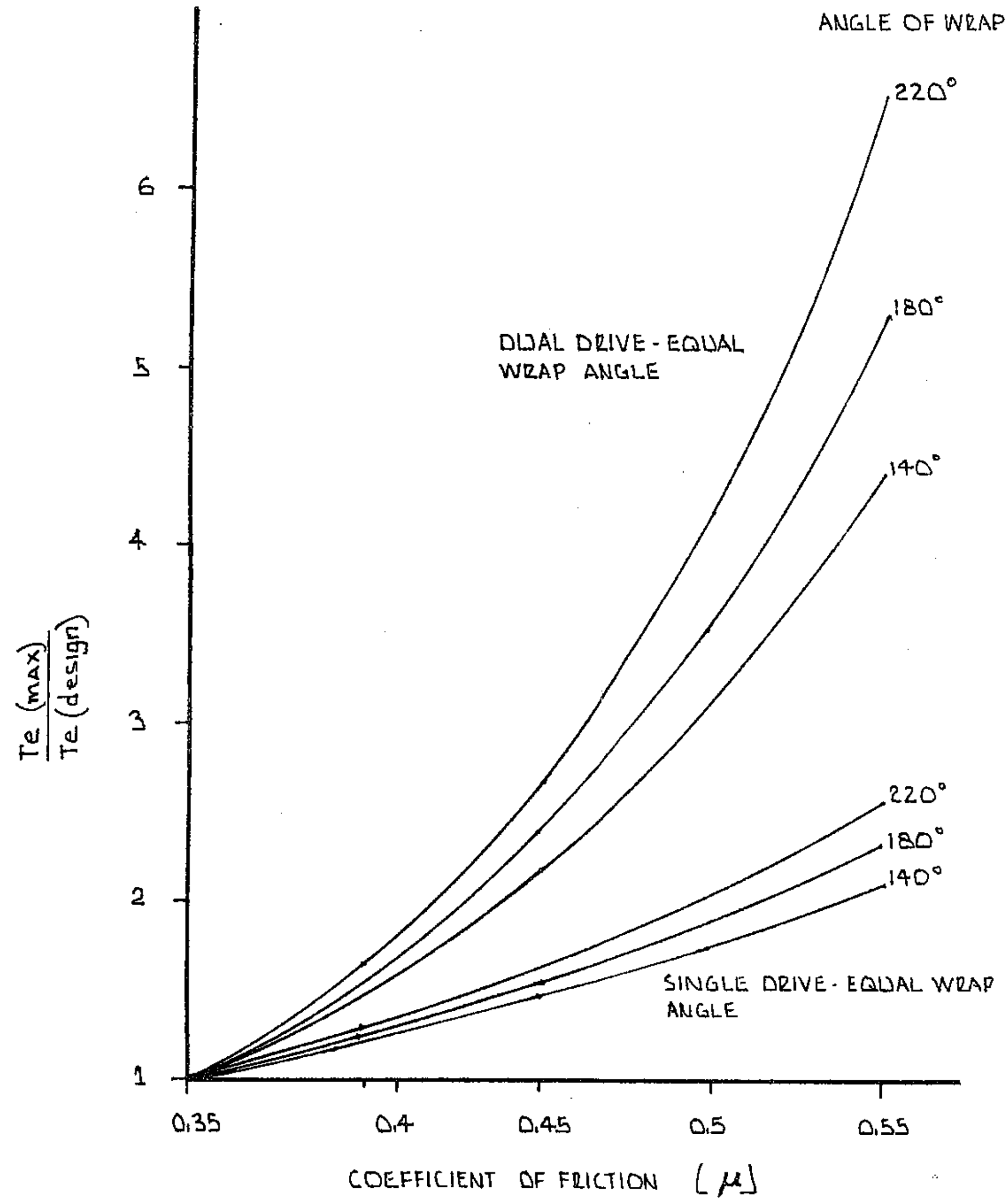


FIG 12

EFFECTIVE TENSION AS A FUNCTION OF COEFFICIENT OF FRICTION

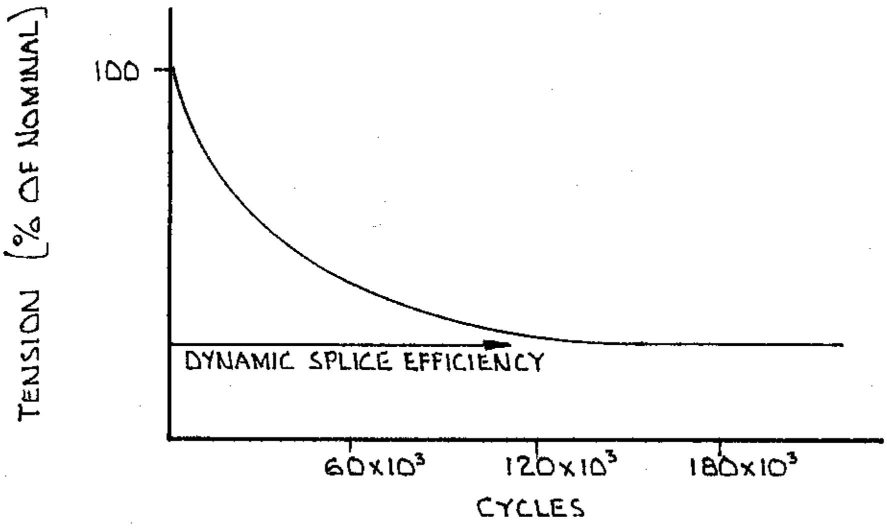


FIG 13

DYNAMIC SPLICE EFFICIENCY

$$FWA = \frac{1}{n}\left(\sum_{a=1}^{n} T_{ave}\ \theta_a\right)$$

WHERE T_{ave} IS AVERAGE TENSION ON PULLEY

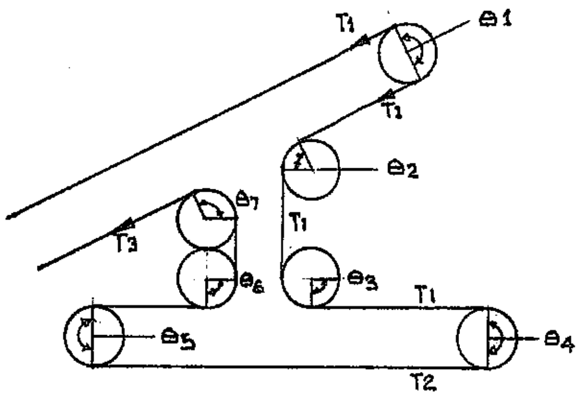
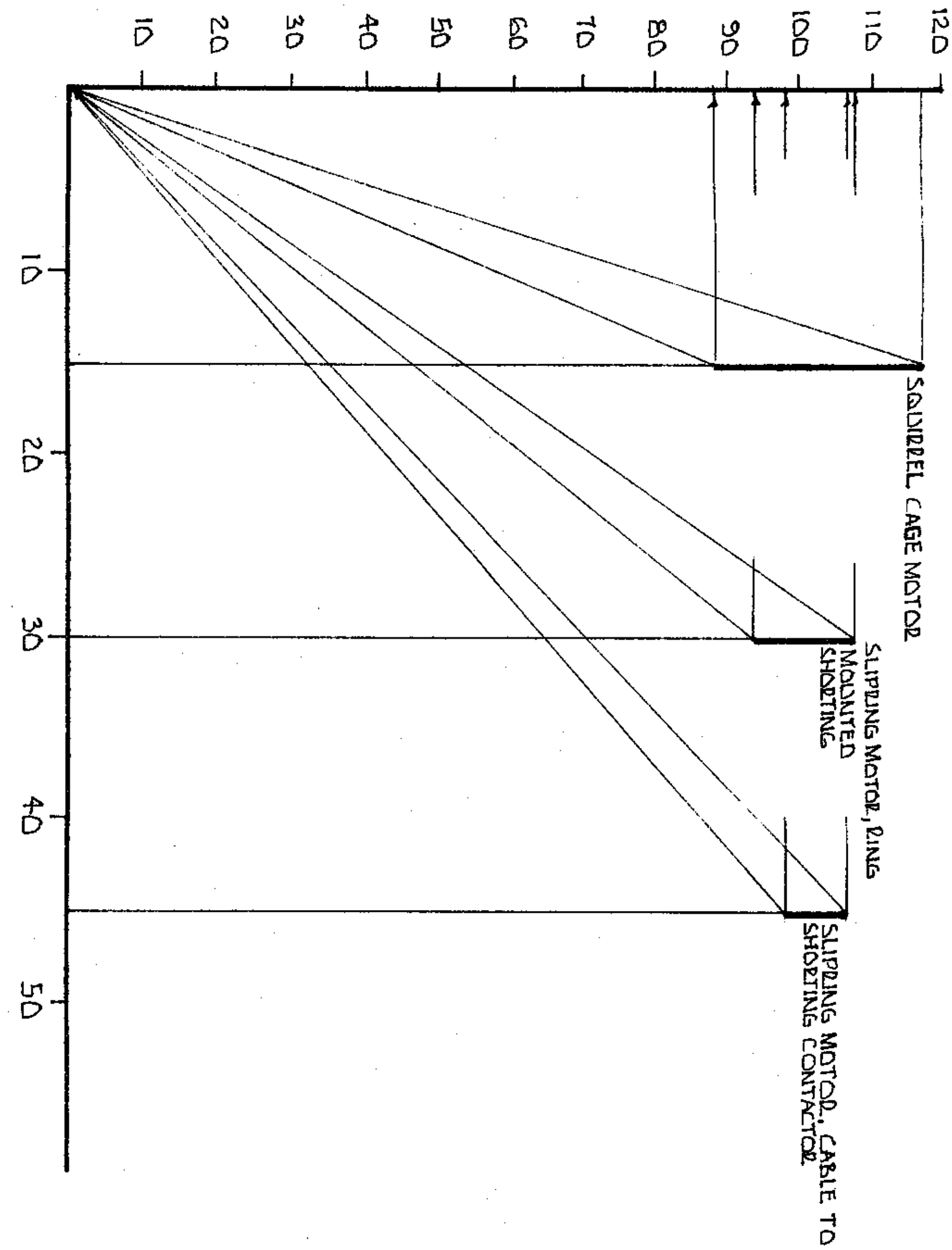


FIG 14

CALCULATION OF TENSION WRAP ANGLE FACTOR



<u>FIG 15</u>
LOAD SHARING WITH AC MOTORS