MEAN AND LEAN, CONVEYOR DESIGN FOR THE NINETIES

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The modern design of conveyor belts focused on producing a "Mean and Lean" product, in other words, a conveying system with the minimal capital and operating costs and maximum availability, using the latest technology available is discussed.

Topics reviewed are computer simulations, high speed belts, belt resistances, the use of horizontally curved conveyors, long single flight conveyors (including their dynamic behavior) and steep inclined conveyors. Also the optimization of conveyor stringers, belting and idlers as well as non conventional designs, such as hanging conveyors and return strand transportation are discussed.

Theoretical background is discussed and real life examples are presented.

INTRODUCTION

The depressed bulk commodities market and the worldwide economic recession increase the always present need for economically efficient transport of bulk materials. Traditionally belt conveying has been the best way of transporting bulk materials in distances from a few meters to several kilometers and quantities up to 20,000 tons per hour. However, traditional design methods based on handbooks or "cooking books" are very conservative and tend to produce over-sized systems. This over-sizing is required due to the shallowness of the analysis and the high number of unqualified assumptions. An in depth engineering analysis of a system allows the engineer to reduce the "ignorance" factor and to produce a more efficient system.

During the last couple of decades, belt conveying technology has advanced significantly, mainly due to the use of more sophisticated analysis and design techniques, but also to the use of novel design concepts. This paper intends to give an overview of these two subjects.

BASIC DESIGN PARAMETERS

CAPACITY

In most cases the design capacity of a belt is determined directly from the operational requirements. "X" tons per hour must be transported from "A" to "B". However, in complex plants, the transport needs are subject to other factors, such as surge capacities, events up and down stream, system availability, etc. Up to a few years ago, these multiple variable problems were approached using "rules of thumb", such as, one shift surge capacity or a design capacity 50% higher than the nominal, etc.

When confronting the design of a complex system or a high cost one, the use of modern computer simulation techniques is a must. Up to very recently, simulation was synonymous with long and boring procedures. Whereas modern object orientated software makes the model creation an easy and fast task, with available hardware running complete simulations in a matter of seconds.

As an example, we can mention simulation work done on the Richards Bay Coal Terminal, currently the largest Coal Terminal in the world with a capacity of 53 million tons per year, in order to evaluate the required modifications to the plant for successive capacity expansions. The model had to take into account the almost 150 different routes that the coal can take from rail to ship. The terminal has six rail wagon tipplers, eleven stackers and reclaimers, almost one hundred conveyors and three shiploaders, all of this for eight different users. An additional problem is the high uncertainty on the train and ship arrival times.

The original simulation was written in 1979, when the terminal capacity was 20 million tonnes per year, and used to run on a 8086 PC (XT), with an alphanumeric interface and taking up to eleven hours for a single simulation run. In 1992, the simulations were done on a UNIX work-station, using a graphic interface, and taking a few seconds per simulation run (on a plant more than twice the size).

A computer simulation allows the analysis of the many possible operational conditions, and the determination of the transport capacities required to meet the production or handling goals.

BELT SPEED

The faster a conveyor belt operates, the more material it can transport for a given width. For a set transport requirement, the faster a conveyor moves, the narrower and therefore the cheaper the system can be. Traditionally, belt conveyors operated between 0.5 and 3.5 m/s, with this latest number still being shown in some handbooks as the maximum recommendable speed for a belt. In some applications the top velocity is effectively limited by operation constraints, typically, the need to avoid the degradation of friable products, but in most cases this maximum speed is restricted only by mechanical and structural parameters as well as the inconvenience of loading and unloading a fast belt.

The longer a conveyor, the cheaper it is to operate the system at high speed. When transporting large lumps or abrasive materials an accelerator conveyor is recommendable, feeding the material to the main belt at a fraction of the main belt speed and reducing the wear and tear of the main belt covers. However, if the system is short, the cost of this extra conveyor and loading point eliminates the advantage of a faster belt.

Figure 1 shows a conveyor system transporting R.O.M. coal with 150 mm lumps to a distance of 6 km at a rate of 2000 tons per hour. The conveying speed is 5.8 m/s (1140 f.p.m.) and the belt width 1050 mm (42") with an accelerator belt that operates at a speed of 3.5 m/s (690 f.p.m.) protecting the main belt. After more than two years of operation and several million tons of material transported, no wear is visible on the covers of the main belt. The high speed made necessary the use of variable speed drives that allow a maintenance inspection speed of 1.5 m/s (295 f.p.m.).

The design of this conveyor demanded the simulation of the transient behavior of the belt as well as the analysis of possible resonance problems in the return strand and the supporting structure. The use of a "normal" speed belt would have required a wider belt with a significantly higher capital cost.

Another case where high speed conveying is economically viable is the transport of large volumes of material. When moving more than 8000 m^3 per hour a "normal" speed belt would have to be almost 3 meters wide. Although this is technically possible, the cost per meter of such a system is very high as components are non-standard. Figure 2 shows a conveyor transporting power station coal at a rate of 11000 tons per hour at a speed of 6.3 m/s (1240 f.p.m.). The belt is 2.2 meters wide. The low abrasiveness of the material conveyed and the high cost of a wider belt or a dual system made the fast belt the best economical option.

The author led a feasibility study on the upgrading of a 1800 mm wide conveyor from 5500 tons per hour of power station coal to 11000 tons per hour. The first option was to replace the belt for a new one, 2200 mm wide. However, in order to avoid changing the full conveyor, with the associated down time, a study was done into speeding up the belt to 10 m/s, to handle the extra volumetric capacity needed. The analysis included power requirements, belt tensions, belt covers wear, possible resonant vibrations (of the belt, the idler frames and the structure) and the problems of loading and unloading a belt at such a high speed. The proposed solution included the use of larger diameter idler rolls (229 mm instead of 152 mm), longer loading skirts and specially designed deflection plates. As the current structure had to be replaced due to corrosion problems, the economic advantage of the faster belt disappeared and the wider belt was chosen. However, the basic engineering performed indicated the feasibility of operating a belt at more than 10 m/s, as previously affirmed by Harrison [1].

IDLER DESIGN

The design and arrangement of idlers is one of the main factors determining the friction resistance of a belt system.

Obviously, the type of bearing and the sealing arrangement will determine the rolling resistance of the roll itself (together with the relation between the shaft and roll diameter). Bearings with small clearances, such as taper roll or regular ball bearings are easily affected by penetration of foreign particles. Some sealing arrangements based on lip seals increase the roll resistance due to the lip friction, this extra resistance disappears when the lip is worn and the seal is not effective anymore. This is the reason why high powers are needed to first start-up some new conveyors with badly designed idlers. Also, in the case of poorly designed labyrinth seals the drag force of the grease packed in the seal increases the roll resistance significantly, with a similar effect of high friction forces on start-up. Furthermore, if the labyrinth is not effective preventing dirt penetration, the dirt-grease mixture not only increases the friction forces but also wears the seal away, making the seal useless.

Furthermore, in the presence of high loads, the roll shaft is deflected significantly, and in the case of bearings with little tolerance to misalignment, such as taper roll bearings, the result is a considerable increase in the rolling resistance [2] and early failure. The problem is exacerbated by the unavoidable misalignment between shaft and bearing produced during manufacturing, that can very seldom be reduced consistently below 6′ of arc. This misalignment is the maximum allowable for a taper roll bearing, so any load caused deflection will result on early bearing failure and increased rolling resistance.

The best results (the least rolling resistance) are obtained when using large clearance, deflection tolerant bearings, such as deep groove ball bearings, as recommended by leading manufacturers like SKF [3]. Taper roll bearings are not appropriate for conveyor idler applications.

BELT RESISTANCE

The other two main components of the rolling resistance of a conveyor are the indentation and the flexing resistances.

The indentation resistance is produced by the energy dissipated by the deformation suffered by the belt in the contact point with the roll, as shown on figure 3.

The diameter of the idler roll is inversely proportional to the indentation resistance, i.e., the larger the diameter the smaller the drag force as the deformation of the belt is reduced. In the same way, the higher the belt tension, the lower the resistance, as the angle of contact belt/roll is smaller, and so is the deformation. The load on the belt and the distance between idlers are proportional to the indentation resistance, as they increase the force between roll and belt and as a consequence the deformation and indentation of the belt. All this factors can be evaluated (or at least estimated) using analytical and experimental methods [3,4]. The other important factor in this force is the belt properties, such as cover elastic modulus, hysteresis, rheology (or in vibration terms its damping coefficient) and the belt stiffness.

The flexing of the belt between idlers produces an energy loss due to the internal friction (or damping) in the belt and in the material transported. The resultant friction force is called the flexing resistance, and is dependant on the actual deflection of the belt. In other words is proportional to belt load and idler spacing and inversely proportional to the belt tension.

CEMA design method takes in account most of the above factors. However, the actual resistance values calculated are greatly conservative. Other design methods , such as ISO and GOOD YEAR apply artificial factors based on historic data. An accurate design must be based on up-to-date research data, applicable to the system to be designed.

In summary, a conveyor designed for low friction resistance must have idlers with large clearance ball bearings, large diameter rolls and with a high average belt tension and minimum sag. Of course, the best design will be determined by the relevant capital and energy cost, and also providing that any potential saving is higher than the cost of a detailed analysis, otherwise, a "cookbook" design and engineering common sense should be applied.

BELT CLASS AND DYNAMIC BEHAVIOR

Traditionally, the required belt class is determined by the rated tension of the selected belt. This nominal tension involves a safety factor of 10 in respect of static ultimate strength in the case of fabric belting (with vulcanized splices). In steelcord belting this factor is usually between 6.7 and 7.5. In the case of aramid fabrics (Kevlar) this factor is highly dependant on the splice and quite variable as the technology is not yet fully developed. However, it is not rare to find enquiry specifications calling for a safety factor of 10, regardless of the type of belting. Probably a result of specifications being copied from project to project, without real updating.

The above factors are all related to the maximum static tension on the belt, as they assume a dynamic or start-up multiplication factor between 1.6 and 2. This factor is related to the additional forces produced by the drive to accelerate the conveyor or the brake (if any) to stop it.

These simple assumptions are not applicable to long and/or high capacity conveyors. In the first place, an incorrect start-up procedure can produce tension multiplying factors of up to seven, causing, as has happened on more than one occasion, the catastrophic failure of the belt. Secondly, the cost of high strength belting makes the use of huge "ignorance" factors uneconomical. Therefore, the analysis and prediction of transient tensions becomes necessary.

The occurrence of elasto-dynamic forces during starting and stopping, over and above the rigid body dynamic forces proportional to the acceleration rate, is related to the excitation of the natural frequencies of the system [5], or as it is called in solid mechanics the generation of stress waves in the belt. These resonant vibrations or tension waves, as one wishes to call them, can be estimated by simplified methods or calculated by finite element analysis, depending on the complexity of the system and the accuracy of the analysis required. Usually the design approach is to avoid the elasto-dynamic problem by proper starting and stopping procedures, or if this is not possible due to design constraints such as power failures, design the belt and associated structures in accordance with the transient forces calculated. In this way the occurrence of catastrophic failures can be avoided, and more economically viable static safety factors, based mainly on splicing and fatigue considerations (about 5 for steelcord) can be used. This approach was followed in the design of the conveyors shown in figures 2 and 4, and no dynamic related problems have been encountered during their operation.

CONVEYOR STRINGERS

The design of the supporting structure of a conveyor belt is not usually considered a critical issue. However, this structure sometimes costs as much as all the mechanical components of a conveyor.

The basic and most repetitive structure in a conveyor is the stringer, and cobsequently will give the best returns from optimization. Traditionally these stringers or tables, as they are sometimes called, have been designed according to structural codes with the main criteria being usually the maximum deflection and aesthetic considerations. An optimal design must be based on dynamic factors as well.

The design of a stringer must be done in such a way that resonance between the rotation of the idlers and the structure is avoided. In this way, conveyors with the same loads per meter running but different speed will have different stiffness requirements for the stringer. The same applies for the transverse vibrations of the belt (flapping).

Once again, the possible saving in structural steel must offset the cost of the detailed dynamic analysis, except in the case of fast belts (over 3,5 m/s) where static design methods are not enough.

ADVANCED APPLICATIONS

LONG, SINGLE FLIGHT CONVEYORS

A cost analysis of a conveyor will show that a large portion of the cost of the system lies in the transfer points. This is applicable both to the capital and the operating cost of the belt. Therefore, a single flight conveyor will in most cases be cheaper to build and operate than a multiple flight system.

Conveyor designers often get asked questions like: How long can a conveyor be? or What's the maximum length for a conveyor? Like most things in life there is no absolute answer. The most important factors in the maximum feasible length of a conveyor flight are the availability of high tensile strength belts and their cost.

Current technology and production costs result in the optimum length for a single flight conveyor being between 6 and 12 kilometers. This is the point where the lowest cost per meter is generally produced. This is of course a general statement.

In practice almost any conveyor above 2 kilometers uses steelcord belting, as take-up requirements for traditional fabric belting become awkward. For a fabric belt the take-up length required for a 2 km system is between 30 and 60 meters, compared to the 3 to 6 meters required for a steelcord belt.

In underground coal mines, like the ones in the South African coal fields, steelcord belting is seldom used. Due both to the fire resistant requirements of the belting and the problems encountered to perform a hot vulcanized splice underground, in a fiery environment. Another factor is the use of temporary installations, that are repositioned every few months. In this conditions PVC belting with mechanical splices is used. Due to strength limitations, the maximum practical length of a single flight is about 2 km. However, mainly in collecting conveyors, longer belts are sometimes required.

In order to avoid the intermediate transfer stations, a single flight belt can be achieved by using linear or booster drives. A booster drive is formed by an auxiliary belt positioned underneath the main one, that transfers driving power by means of belt to belt friction. In this way the maximum tension in the belt can be reduced as shown on figure 5, which shows the tension profile on a belt with multiple booster drives. The design of this sort of systems is based on research data and operational experience [6].

A potential application for this kind of technology is very long overland conveyors, especially when material degradation must be minimized. One of the reasons is the almost exponential relation between strength and price on steelcord belting. A preliminary study done on a 21 km system, showed a 30% saving between the use of a single flight belt, with 2 booster drives, and a conventional system with 2 flights.

HIGH LIFT AND CAPACITY CONVEYORS

In many applications, for example in-pit crushing operations, it is necessary to lift the material up to several hundred meters in the minimum distance possible. When moving large tonnages, high strength belting and the determination of transient dynamic forces are required.

The largest system of this kind in the world is located in Germany and transports 1800 t/h of R.O.M. coal over a distance of 3745 meters and with a lift of 783 meters. The installed power is 7,2 MW and the belt class 7500 kN/m.

In general, these sort of conveyors do not present many problems on start-up, but they do present problems during stopping. On a power failure or aborted start-up, the tension drops rapidly from the maximum tension to the minimum, and if transient forces are produced, belt tension can become zero. In this condition the belt folds between idlers and snaps up again when the tension recovers, expelling any material that could have been in the belt. If the conditions are right, a non linear coupling of the longitudinal (tension variation) and transversal (flexing) oscillations can occur. If this happens the consequence is almost surely catastrophic failure.

In this sort of belt the prevention of the occurrence of very low or zero tensions in transient conditions is of paramount importance. If this not possible due to design constraints, a dynamic simulation must be performed and the conditions necessary to avoid non-linear vibrations found and implemented.

DOWN HILL CONVEYORS

Steep incline downhill conveyors are often regenerative, in other words they generate energy when loaded. These types of conveyors are quite common in mountain mining areas like Chile, where the material is often taken from the mountain to the beneficiation plant downhill. Figure 6 shows one of such systems.

A downhill conveyor must have enough power to drive the empty belt and enough braking capacity to withhold the loaded belt. The main safety feature of such a conveyor is a failsafe braking system, able to stop the conveyor in a power failure and overload situation. A runaway conveyor will often destroy the drive due to centrifugal forces and sometimes the drive and tail station.

Braking must be applied in a fashion such that no transient forces are created or the system designed to withstand them. Sometimes variable speed drives are used to ramp up and down the belt speed and the brake is reserved for locking off the standing belt and in case of power failure.

When designing a system like this, the conservative position is to use a realistically low friction factor for the calculations. Otherwise the regenerative force can be underestimated and the system under designed.

HORIZONTALLY CURVED CONVEYORS

Very often, due to geographic conditions, it is not possible to install a single flight straight conveyor system. Therefore, it is advantageous to curve conveyors and save thereby on earthworks, bridges, transfer stations, and so on.

The first problem to be addressed is the difference in tension between the inner and outer edge of the belt, as the outer edge follows a longer trajectory and tends to elongate more. If the tension in the outer edge is too high, damage can be produced to the belt carcass. If the inner edge tension is too low or zero, the belt will sag excessively.

The second problem is the existence of a radial force towards the inner curve, that will cause the belt to drift off the inner curve rolls. Various systems have been proposed to control this drifting. Probably the simplest one is using Garland type idlers. By lateral shift of the idler configuration an equilibrium between radial and gravitational forces can be obtained. A similar idea uses rigid idler frames, which are pivoted above themselves. The third method is to employ standard idler frames which are essentially superelevated on the inner edge of the curve and horizontally cambered and twisted. This method has proven to be superior to the Garland and pivoting type due to the mechanical sensitivity of these two first systems. With the right test data, all the relevant frictional, tensional and gravitational forces present in a conveyor curve can be simulated in a computer program, and a satisfactory design produced. This was the methodology used to successfully design the conveyors on figures 2 and 4, as well as a curved conveyor that transports 6000 t/h at 5.8 m/s [7].

NON CONVENTIONAL DESIGNS

HANGING CONVEYORS

In underground mines it is not unusual to hang a conveyor belt from the roof instead of using the footwall to mount it on top. This configuration facilitates cleaning underneath the belt and eliminates the need for supporting legs. In some cases of deep hard rock mining, like the South African Gold Mines, the cost of the civil works required to mount a conveyor can be significant, and much higher than installing roof bolts to hang the belt from.

Totally suspended systems, with garland idlers hanging from a cable stringer that hangs from the roof, are used in coal mines. In hard rock mining, it is more appropriate to hang a conventional continuous steel stringer due to the large loads involved.

LOW CAPACITY BELTS

Figure 7 shows a conveyor designed to handle 180 t/h of gold ore at 2000 meters depth. The belt is 580 meters long and has a vertical lift of 157 meters. Due to the maximum lump size (300 mm), the width of the belt was chosen as 1050 mm (42"). The conveyor is loaded by a loading conveyor fed by orepasses.

Traditionally, due to the problems to control accurately the feed, the conveyor would have been designed according to volumetric capacity and the belt speed set accordingly. However, in this case that would have resulted in a high class belt, large pulleys and a huge high reduction drive, all of this impossible to handle underground, without mentioning the high associated cost. It was decided instead to load the belt intermittently, utilizing only a fraction of the volumetric capacity. A power transducer on the drive motor activates an alarm when the maximum load has been reached in the belt. The operator stops the loading conveyor, waits for the main belt to clear up and starts loading again. If the belt keeps on being loaded, an overload switch trips the belt. In the event of the alarm and the overload switch not working, the integrity of the system is protected by the fact that belt and structures can take the full stall torque of the drive. In this way, the capital cost of the system was reduced significantly and a fully workable system was given to the client.

RETURN STRAND FEED

In most mining and industrial plants raw materials are brought in, while product and waste are sent out, often using the same route. Conveyor belts can be used to transport material in two directions. Historically, this have been done by using reversible belts. The disadvantage of this system being that material cannot be transported in both directions simultaneously. An alternative is to use the return strand of the belt for loading purposes also, in this way, material can be transported in both directions simultaneously.

Figure 7 shows a new conveyor belt feeding the return strand of an existing one. In this case the client had an existing raw material import conveyor and required a product export conveyor system. A new conveyor was built to collect the material from the product storage building and used to feed the return strand of the existing overland conveyor. At the other end a transfer station and an unloading conveyor were built. Due to the terrain elevation, the conveyor can transport material in both directions with virtually no extra power needed. The cost of the project was 40% of the cost of a system with a new overland belt.

CONCLUSIONS

The capital and operational cost of a conveyor system can be substantially reduced by means of proper analysis and the use of novel ideas and techniques. This kind of cost reduction will make a contribution towards maintaining the competitiveness of the conveyor belt technology and towards the economic mining of raw materials

REFERENCES

[1] Harrison, A.,Hayes, J.W.,Roberts, A.W.,"The feasibility of high speed narrow belt conveyors for bulk solids handling",Mechanical Engineering Transactions, Number 3,1982.

[2] Greune, A.,Hager, M.,"The energy saving design of belt conveyors",Bulk Solids Handling,August 1990.

[3] Stewart-Lord, M.,"Rolling bearing characteristics for maximising the life of conveyor idler rolls",International Materials Handling Conference 1991.

[4] Spaans, C.,"The calculation of the main resistance of belt conveyors",Bulk Solids Handling,November 1991.

[5] Zamorano, S.P.,"Dynamic analysis of conveyor belts",The South African Mechanical Engineer, September 1991.

[6] Küng, W.,"Booster drive for Twistdraai Colliery",Design report, 1989.

[7] Zamorano, S.P.,"The largest capacity horizontally curved conveyor in the world",Technology S.A.,January-February 1992.

Figure 1, Koorfontein Overland Conveyor, South Africa

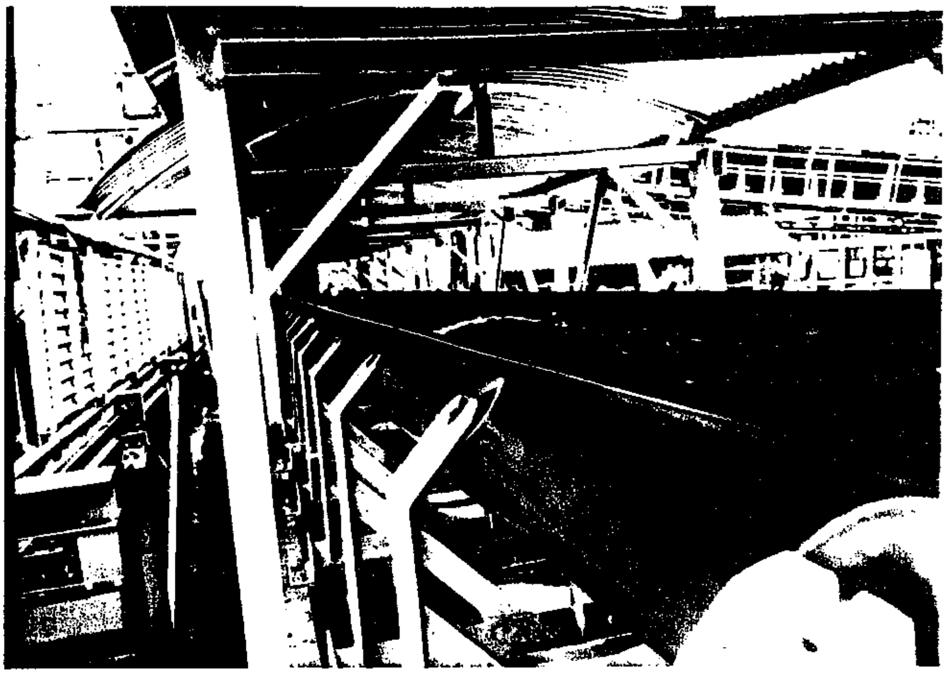

Figure 2, 11000 t/h conveyor, Richards Bay, South Africa

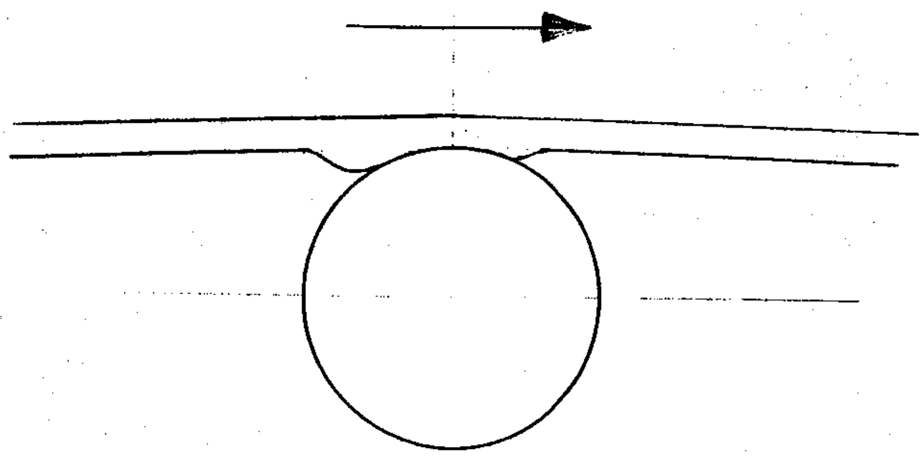

Figure 3, Belt indentation

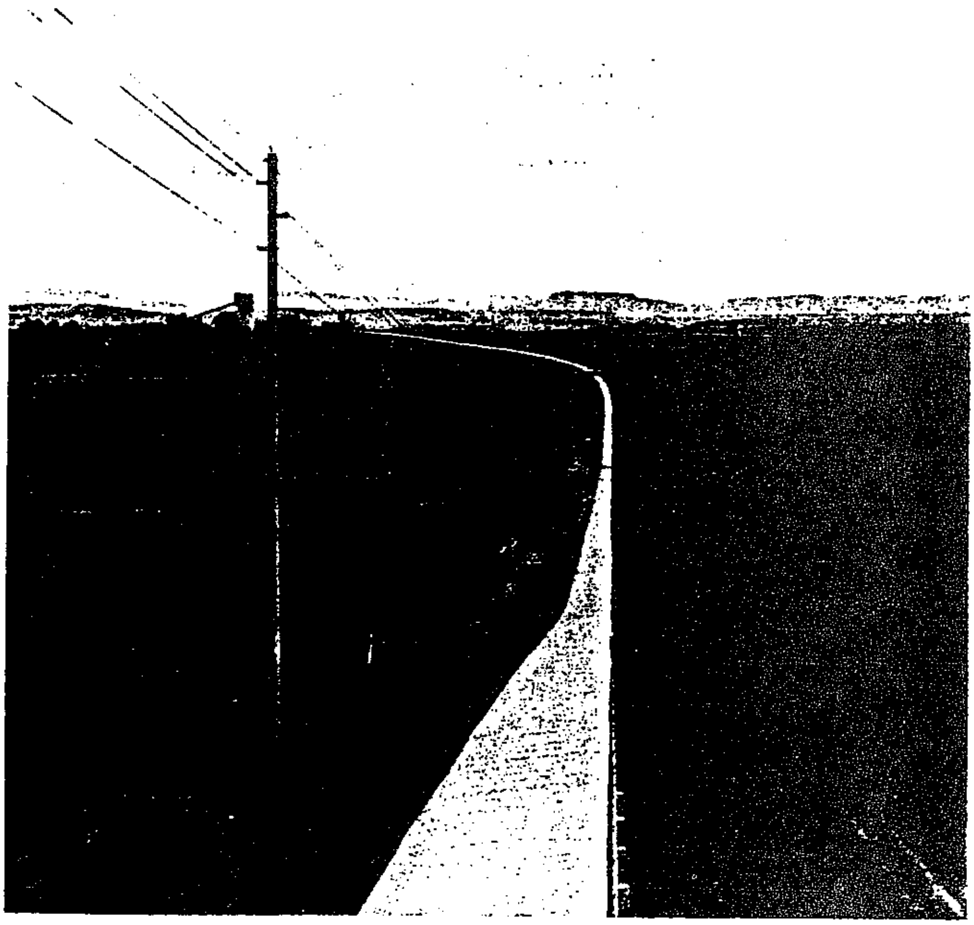

Figure 4, Majuba Overland conveyor, South Africa

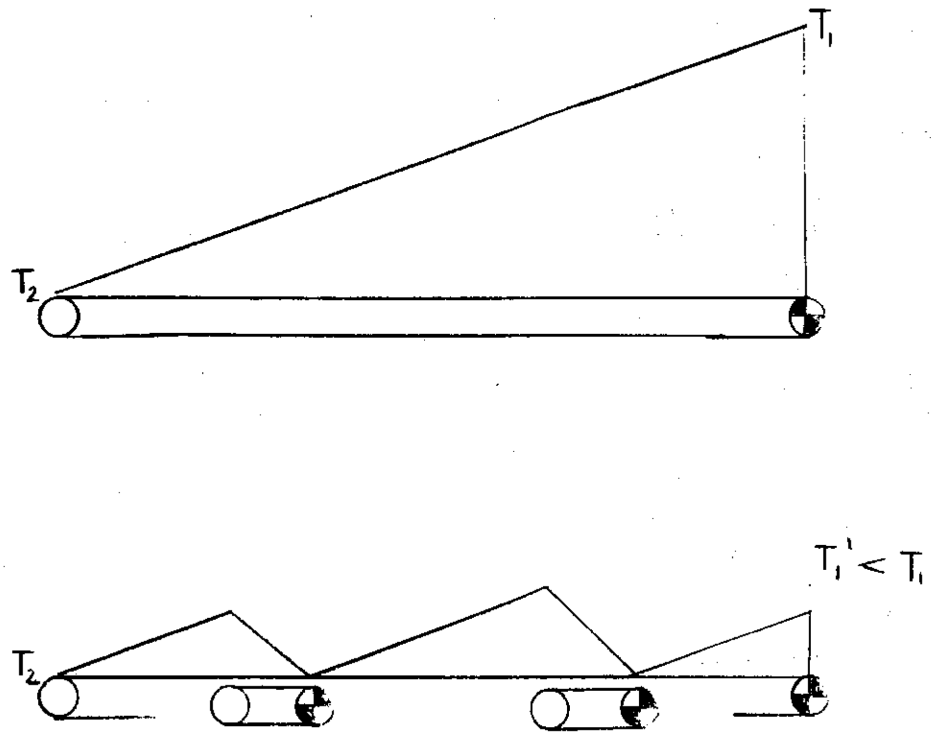

Figure 5, Tension distribution with booster drives

Figure 6, Heap Leach Conveyors, Tres Cruces, Chile

Figure 7, Incline conveyor, Kinross Gold Mine, South Africa

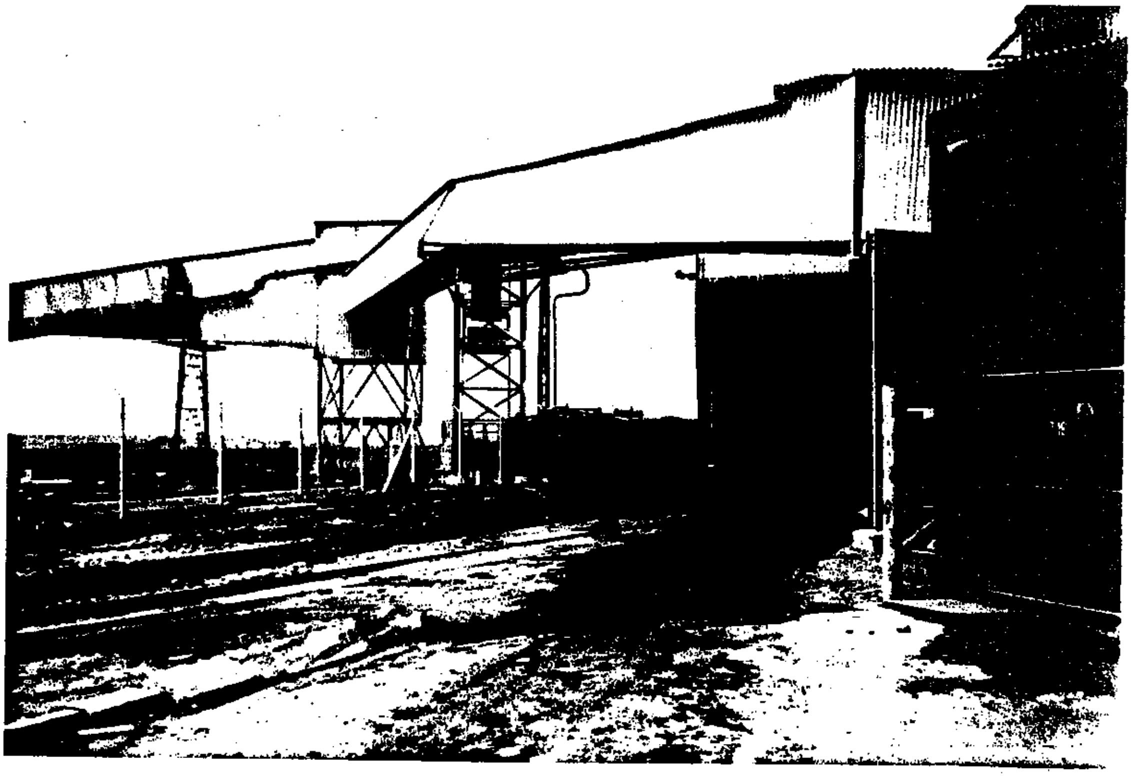

Figure 8, Return strand feed, I.O.F., South Africa