Chapter 29

THE SELECTION AND SIZING OF CONVEYORS AND STACKERS

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ABSTRACT

This paper reviews practices used in the selection and sizing of belt conveyors and stacker systems commonly used in crushing and grinding plant facilities. Historical and modern methods of sizing this equipment are presented. The new methods include: 1) probability theory applications for establishing component reliability, 2) "Monte Carlo" simulation techniques used in evaluating alternative system designs and in establishing service factors and corresponding availability indices of each system, 3) quantifying the system and component operating characteristics using computer assisted modeling tools simulating rigid body and elasto-mechanic behavior for starting and stopping large and/or complex belt conveyors.

INTRODUCTION

It is axiomatic that the engineer design the comminution system for a maximum return on investment. The corollary to this proposition is that the design should maximize the utility of all of its constituent parts within practical limits. The process of selecting the appropriate belt conveyor size and its associated components is again examined from this proposition. A logical method is presented which leads to matching the conveyor size selection to the job requirements. References are made to historical methods where applicable.

BELT WIDTH SELECTION PROCEDURE

The conveyor belt width selection is dependent on many factors. A cataloging and brief description of the major considerations is given:

A. Basic tonnage flow rate (BFR) requirements set by process circuit

B. Material properties

   1. bulk density

   2. surcharge angle

   3. lump size and distribution

C. Flow rate service factor (FRSF) measure above the BFR for peak material loading evaluation are set by:

1. surge required due to feeding system

2. make-up required due to process demand at discharge

3. surge required due to material process variances

D. Belt speed, as influenced by:

1. destruction of transfer station components and ore carrying idlers

2. material degradation at transfer station

3. material flowability through transfer

4. air erosion or environmental factors

5. tracking sensitivity

6. component standardization

7. associated equipment limitations (trippers, et al)

8. wear rate of belt covers

9. sequential stopping time control of conveyors in series

E. Belt edge distance allowance to inhibit material spillage, as influenced by:

1. tracking alignment (wind, idler alignment, structural alignment, load centering error)

2. lump size and percent distribution

3. material surcharge angle

4. idler troughing angle

5. topographic belt line variations

6. transport distance

F. Cross-sectional area capacity as influenced by:

1. load station flow rate correction

2. Belt incline/decline sections

3. Idler trough configuration

Of the six mentioned factors (A through F), only the basic flow rate (A) and material properties (B) are set as prior conditions. Selection of these values is outside the scope of this text.

Flow Rate Service Factor (Item C)

The flow rate service factor (FRSF), is defined to be a multiple of the basic flow rate (BFR). Their product will yield the necessary conveyor system tonnage throughput for a specified time span. The FRSF can alternately be expressed as a measure of the performance variance above the BFR or unity tonnage rate. The BFR can be governed by the material input feed rate to the conveyor, the demand feed rate at the discharge point, or by a combination of the two. The product of the BFR and FRSF is defined as the peak flow rate (PFR). Therefore, to establish the peak conveyor load, the FRSF must be set before the belt size can be selected.

This paper will assume that the FRSF is synonymous with the system availability, such that the measure of loss of production (non-availability) can be made up by increasing the flow rate above the BFR. The FRSF is dependent on the conveyance flow circuit differentiating a single conveyor in a circuit, or groups of conveyors connected in series, parallel, or both. For simplicity, parallel interaction will not be discussed.

An illustration of four basic types of circuits is given in Figure 1. These circuits demonstrate, by example, the qualitative effect of stockpiles or surge bins on the conveyor system sizing. For clarity, the following definitions are assumed:

- The input feed refers to ore material placed on the belt from the mine side.

- The demand feed refers to the ore discharging from the belt to the next process.

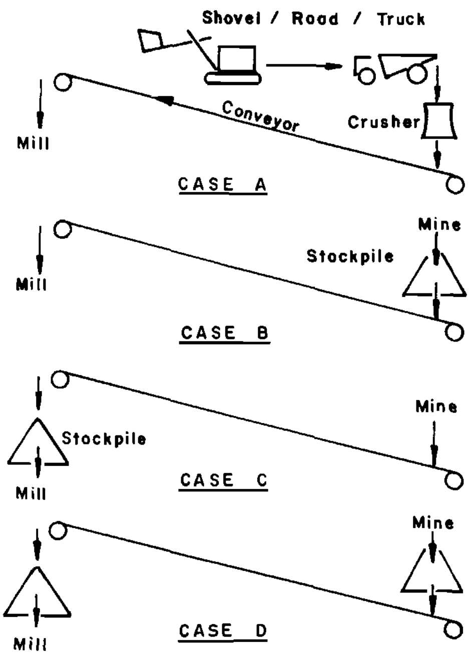

FIGURE I. Basic Conveyor Circuits

Case A is without stockpiles. The input feed, the process demand feed at the discharge, and the conveyor system proper, can each independently have an impact on the conveyor size selection. The availability of material supplied and demanded can cause surges in the transport tonnage.

On the input feed side, crushers, trucks, shovels, roads, weather, etc. can alter the availability of ore. There may be cyclic surges of the input ore, as an example, due to high process rates caused by changes in ore properties of from over blasting.

On the process demand (discharge) feed, the process feed requirements may cycle or be intermittent due to the process or its machinery.

Thus, the net system availability is the product of independent availabilities of input feed, conveyor, and demand feed. The consequent conveyor size may be affected by all three.

The case A circuit has the least likely steady-state flow expectations and greatest flow rate variance. Therefore, it has the largest degree of under-utilization.

Case B is with a stockpile ahead of the receiving conveyor. The input feed from the mine side is partially or fully isolated from the mining activity. Therefore, the conveyor flowrate is dependent on the demand rate from the mill process. The mill feed is dependent only on the availability of the conveyor system.

Case C is with a stockpile at the demand (mill) feed. The input feed variances, as in case A, are dependent on the mine ore availability. Therefore, the mill requirements are dependent on the availability of the mine, the conveyor systems, and the mill stockpile isolation capacity.

Case D is with stockpile at both ends of the conveyor. Thus, with proper stockpile sizing the mine and mill process rates can vary from the conveyor design flowrate. The conveyor flowrate will be a steady rate, yielding maximum utilization and minimum cost. The mine, conveyor and mill equipment can each run at their maximum capacities. Excepting for the cost of stockpiles, the economies of scale are optimized for each system independently where the level of availability is balanced against capital expenditures.

Belt Speed (Item D)

In general, the belt speed is maximized within good design practice. This is to realize the significant cost savings from selecting the minimum necessary belt width to meet the flow area requirements. The belt speed selection is dependent through the interaction of the nine given factors.

A number of references give some basic guidelines for the speed limitations corresponding to the material properties, belt width, and ancillary equipment, such as CEMA (1979), Rijsenbrij (1972), Goodyear (1979). The graphs, shown in Figure 2, indicate some empirical limits for general design with respect to points 1 and 3.

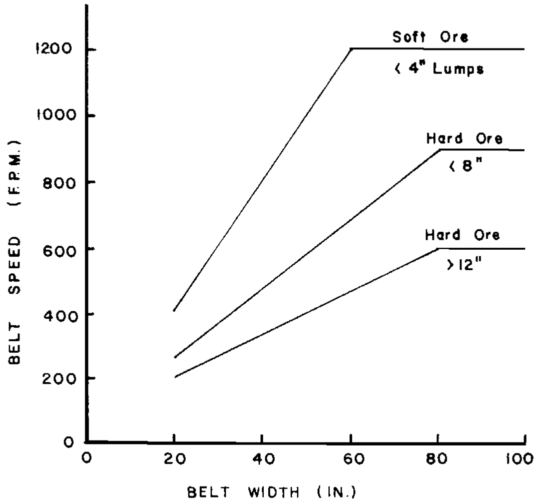

FIGURE 2. Belt Speed Limit vs Width

Transfer Station. The speed selection should take into account maintenance of the transfer station. As the belt speed increases, the damage to transfer station equipment and degradation of the ore increases as the square of the increased speed. Environmental control also becomes more difficult. The design of transfer flow is outside the domain of this paper. A good reference is Colijn (1972).

Trippers. Belts with trippers are speed-limited due to their restricted chute flow path. In hard rock mining, the speed in feet per minute (FPM) is limited to about 10 times the belt width in inches with an upper limit of 700 FPM.

Belt Edge Distance (Item E)

To insure against spillage at the loading station, and to allow for changes in transport geometry, the distance between the belt edge and the material must be evaluated.

Six factors were stated which influence the edge distance selection. A brief discussion of each factor follows:

Tracking. The belt will not perfectly track to its theoretical centerline for various reasons. An allowance is made to accomodate fo these conditions. Various standards, such as CEMA (1979), DIN 22101 (1942), and JIS-B-8805 (1965), reference the minimum edge allowance for tracking as:

$$d1 = 0.055(W) + 0.90 \text{ (in.)} \qquad (1)$$

This industrial standard allowance does not include any consideration for lump containment.

Lumps. To contain lumps at the loading zone and at the discharge, a separate evaluation is required. When material leaves the skirtboard containment, it is in a state of flux. An edge distance allowance is made for large lump containment, as shown in Figure 3, by the following formula:

$$d2 = (SF)\ \left(\frac{\phi}{2}\right)\ \text{TAN} \left[\frac{\alpha + \beta}{2}\right] \qquad (2)$$

where:

SF: slab factor of lump size, normally 1.5
ϕ: rock diameter (in.)
α: surcharge angle (degrees)
β: idler trough angle (degrees)

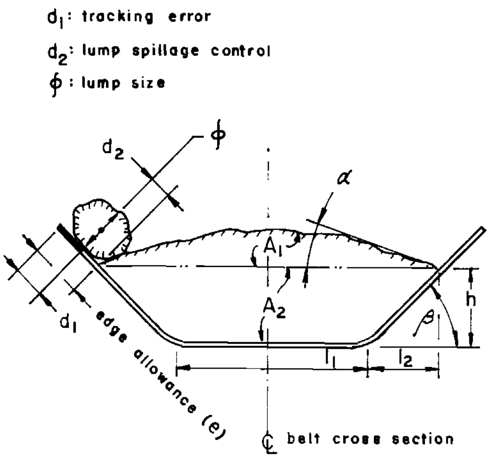

FIGURE 3. Belt Cross Sectional Area & Edge Distance Allowance Dimensional Parameters

An example edge distance calculation is given:

W: belt width 60 inches
ϕ: rock size 8", primary crusher
α: surcharge angle 25 degrees
β: idler trough angle 35 degrees
SF: ore slab factor 1.5

edge = d1 + d2

$$= .055W + .9 + \left[\frac{SF \cdot \phi}{2}\right] \text{TAN} \left[\frac{\alpha + \beta}{2}\right]$$

$$= .055(60) + .9 + \left[\frac{(1.5)(8)}{2}\right] \text{TAN} \left[\frac{20^\circ + 35^\circ}{2}\right]$$

= 7.66 inches

This yields a 72% belt loading based on the CEMA (1979) definition. Historically, engineers have used a fixed value (e.g. 75%) as a basic design criteria without regard to the lump size, belt width, angle of incline/decline, or other factors yet to be discussed. This can lead to significant overdesign or underdesign of the belt width as will be illustrated later. Figure 4 illustrates the variance in loading percentage due only to tracking and lump size allowances.

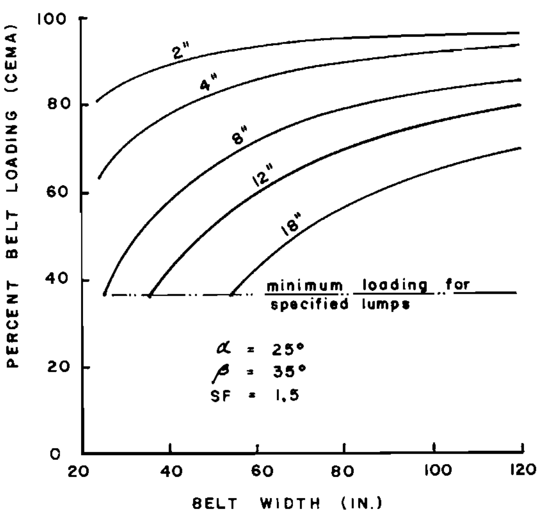

FIGURE 4. Load Percentage vs Width Dependent on Lump Edge Distance & Tracking Error

The size and distribution of lumps should be considered in this evaluation. If the material is being retrieved from a stockpile, the larger lumps tend to segregate to the outer edges and along the bottom of the pile. Thus, a concentrated flow of larger lump material may occur. Engineering judgement is required to tailor this allowance factor to the lump percentage in the flow stream.

The belt width should be no less than three times the largest dimension of the maximum lump size to prevent lumps from pinching between the idler wing rolls.

The lump size and distribution can affect the material surcharge angle in the loading zone. A high percentage of lumps, which measure one-sixth of the skirtboard width or less, can reduce the expected surcharge angle of the published data.

Surcharge Angle. The material surcharge angle is sometimes referred to as the dynamic angle of repose. Agitation of the material as it is loaded onto the belt tends to reduce the angle of repose to the surcharge angle. The surcharge angle is reduced still further by agitation of the material as the belt is transported over the idlers. An empirical (rule-of-thumb) formula which corrects the surcharge angle for overland transport follows:

$$\alpha^{*}(\text{corr.}) = \alpha(\text{baseline}) - (L/K) \qquad (3)$$

where:

- α: the known material surcharge angle in degrees
- L: the conveyor length in feet
- K: varies with belt sag between idlers

K = 1000 ft./deg.
for $\geq 1\%$ average sag, or for multiple changes in the belt line geometry

K = 2000 ft./deg.
for $< 1\%$ average sag

$$\%\ \text{sag} = 12.5\ (Wt)\ (Sl)/T \qquad (4)$$

Sag is defined to be the deflection below belt line, between idlers.

- Wt: Material and belt weight between idlers (per foot)
- SI: idler space (per foot)
- T: belt tension (lbs.)

range $\alpha \leq \alpha^{*} \leq \alpha/2$

Cross-Sectional Area (Item F)

The belt cross-sectional area, reference Figure 3, is calculated for equal length idler rolls by the following formula:

$$\text{Area} = A1 + A2 \quad (\text{in. }) \qquad (5)$$

$$A1 = (l_1 + l_2)(h) \qquad (6)$$

$$A2 = \qquad (7)$$

$$\left[\frac{(l_1 + 2(l_2))^2}{4\ \text{SIN}\,\alpha}\right]\left[\frac{\pi\alpha}{180\ \text{SIN}\,\alpha} - \text{COS}\,\alpha\right]$$

where:

- α: surcharge angle of material (deg.)
- β: idler troughing angle (deg.)
- W: belt width (in.)
- e: edge distance allowance (in.)
- $l_1 = 0.371(W) + 0.25$ (8)
- $l_2 = (.3145(W) - .125 - e)\text{COS}\,\beta$ (9)
- $h = (.3125(W) - .125 - e)\text{SIN}\,\beta$ (10)

Note: for equal length rolls only

Load Station Flow. The acceleration of material at the loading station should be examined for potential spillage as the flow stream exits the skirtboard containment. When the material is first introduced onto the belt, the material velocity does not match the belt velocity. This causes an increase in the flow area per the formula:

$$\text{Area}\ (x) = \frac{(TPH)\ (Ku)}{V(x)\ \rho(x)} \qquad (11)$$

where:

- x: distance from the load point x
- Ku: constant for given units
- V(x): velocity at point x
- $\rho(x)$: apparent material density at point x

Assuming that the flow density (ρ_x) remains constant, for lack of sufficient data, then a relationship can be given which mathematically predicts the material cross-sectional area with respect to a point x. Colijn (1962) expressed this as an "overload ratio (k)".

$$k(x) = \frac{Area(x)}{Area} \quad ; \; k(x) \geqslant 1 \qquad (12)$$

This ratio indicates the multiple above unity the belt will carry at the point (x) evaluated, as shown in Figure 5.

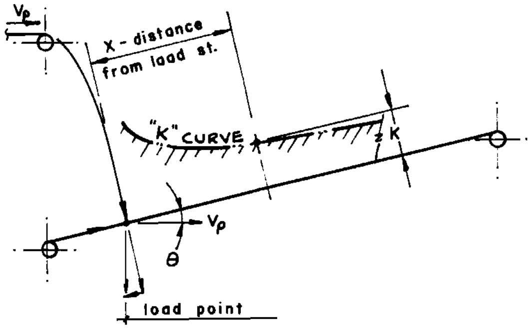

FIGURE 5. Representation of K-Curve on Incline Load Station

The formula for the overload factor k is given in terms of pseudo velocity relationships:

$$k(x) = \frac{Vb - Vc(x)}{Vo + Vl(x) - Vd - Vs(x)} \qquad (13)$$

where:
- k(x): cross-sectional area of ore on belt which indicates multiple of overload above unity
- x: distance from load point
- Vb: belt velocity (fpm)
- Vc(x): cohesive resistance between belt and ore at x
- Vo: initial ore velocity at loading point (x=0)
- Vl(x): chute drop distance impact force assist at x
- Vd: chute drop distance impact force at x
- Vs(x): skirt length drag velocity loss

$$Vc(x) = \frac{3.48\,(C)W(x)}{T} \qquad (14)$$

$$Vl(x) = \frac{(\mu_r \cos\theta - \sin\theta)(x)}{(0.0311)\,(Vb)} \qquad (15)$$

$$Vd = 8 \cdot \sqrt{D} \cdot (\sin\theta) \qquad (16)$$

$$Vs(x) = \frac{\mu_s \rho H^2 (1 - \sin\lambda)(X)^2}{2.5\,(T)\,(1 + \sin\lambda)} \qquad (17)$$

$$Vo = (Vp)(\cos\theta) \qquad (18)$$

- C: cohesion (lbs / sq. ft.)
- T: flowrate, short tons / hour (STPH)
- μ_r: friction coefficient of ore on belt
- θ: belt line incline / decline angle (degrees)
- D: effective vertical chute drop distance (ft.)
- μ_s: friction coefficient of ore on skirt wall
- ρ: material density (lbs./cu. ft.)
- H: material height along skirt (in.)
- λ: material repose angle (degrees)
- Vp: horizontal velocity component at impact point (e.g. previous conveyor belt speed)

The k-value gives an indication of the turbulence in the skirtboard zone. Since most of the belt cover wear occurs at the point of loading, a higher k-value indicates a level of increased wear. The length of skirtboard should be set such that the k-value is unity at the end of the skirtboard (belt and material are at the same speed).

The magnitude of change of k, and the configuration of the turbulence zone ar illustrated in Figure 6.

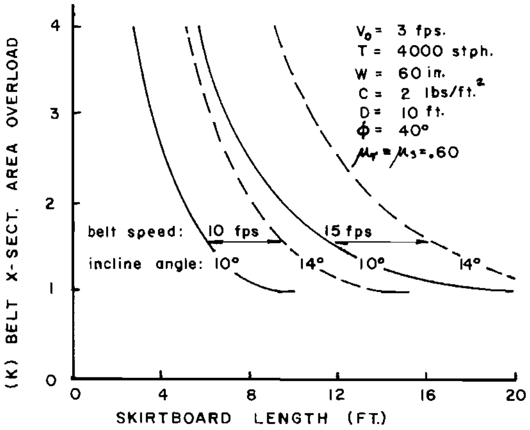

FIGURE 6. Belt Overload Ratio vs Skirtboard Length

Belt Slope. During starting and stopping a conveyor with material on a sloped portion, a determination should be made on the stability of the material. A harsh jolt may fluidize the load. A derivation of the k-factor can be used as a criteria. The material stability criteria is given as follows:

$$\mu_r \leq \mu_a \leq \frac{a}{(g)\ \cos\theta} \qquad (19)$$

$$\leq \frac{FPM}{1932(time)\cos\theta} \qquad (20)$$

where:

$\mu_a = \tan(\theta) - \tan(\alpha)$ (21)
μ_r: friction coefficient of material to belt
α : surcharge angle (deg.)
θ : belt slope angle (deg.)
time: accel/decel time (sec.)
g: acceleration of gravity

This criteria assumes no cohesive resistance between material and belt.

The cross-sectional area, when observed in normal section (Figure 3), must be corrected for an incline or decline portion of the conveyor (CEMA). The area formula given Area = A1 + A2 is corrected as follows:

$$\text{Area} = (A1)\cos\theta + A2 \qquad (22)$$

where:

θ: the belt slope (deg.)

The DIN 22101 makes a correction to the incline belt loading using a different criteria. The area correction factor by DIN 22101 is given by the formula:

$$\text{Area(corr.)} = \frac{\text{Area(baseline)}}{\cos^3\theta} \qquad (23)$$

where:

θ: the belt slope (deg.) $\theta \leq 20°$

Idler Trough Angle. The resultant belt cross-sectional area calculation, as derived from the idler configuration parameters, is dependent on:

- roll configuration or shape factor (length, angles)
- edge distance allowance
- effective surcharge angle

Jilek reviewed the idler trough configurations in depth. His studies included idler configurations from one to five carry rolls with varying roll lengths and troughing angles. His results showed that by altering the lengths of the rolls, for a given troughing angle, there was an optimal area to roll length relationship.

In the standard three-roll idler set, the optimal roll configuration, for most material surcharge angles was found to be:

$$LR = 0.005(\beta) + .075 \quad (24)$$

where:

LR: ratio of center roll length to belt width
β: trough angle (deg.)

This concept is not normally used because of non-standard components and potential belt tracking difficulty due to belt conformance with the trough slope.

SYSTEM AVAILABILTY

It is essential to good design practice to quantify, for economies of scale, the expected system and related sub-system(s) measure of reliability. Reliability is defined as the measure of the probability that a system or its components will perform adequately for an expected time period and for the given operating conditions. Production revenue is directly related to the operating questions of:

1. How long will the process work without interuption?
2. What are likely interuptions?
3. How long will the interruption last?
4. What is the maintenance strategy?
5. What are the production bottlenecks?
6. Can the cost justify elimination of the bottleneck?

It is possible to address these questions through reliability analysis.

Prediction of the system's behavior requires investigation of the constituent parts that influence its performance. Therefore, a reliability measure should be set for all governing elements to predict their respective unscheduled downtimes. The downtime prediction or time to expected failure for each component can be expressed in the form (Tsokos, 1972):

$$F(t) = 1 - R(t) \quad (25)$$

$$= 1 - e^{-t/MTBF} \quad (26)$$

where:

t: denotes contiguous time of operation without failure
F(t): probability of unscheduled shutdown for time t
R(t): probability of reliable operating for time t
MTBF: mean time between failures, =T/f where f is the number of recorded failures in time T

This statistical measure, although not ideal, is commonly used in the prediction of component failure indices where measures of the MTBF have been verified by field data. The expected failure is derived from the exponential probability density function.

The ultimate goal is to predict the system's availability from the known reliability parameters. Specific to the stated reliability function, a given conveyor's availability index can be expressed in this form (Nordman, 1975):

$$AI(\text{system}) = \prod_{i}^{n} \left[AI(\text{component}) \right]_i \quad (27)$$

$$AI(\text{system}) = \prod_{i}^{n} \left[\frac{MTBF}{MTBF + MTTR} \right]_i \quad (28)$$

where:

AI: measure of conveyor availability
i: i-th component

MTTR: mean time to repair the i-th component

Note: component may be synonymous with conveyor assembly.

To make the cost trade-off on the conveyor component availability, a measure of cost effectiveness (CE) is used. This is expressed in the form:

$$CE = \sum_{i}^{n} \left[\frac{VP(An-Ao)}{Cn-Co} \right]_i \tag{29}$$

where:

- CE: cost effectiveness of system where greater than 1.0 indicates the multiple increased level of effectiveness.
- i: i-th component evaluated
- VP: product value
- An: availability index of new equipment
- Ao: availability index of original (baseline) equipment

The CE, as given above, is an estimate in that it does not include the time value of money.

Conveyor components and system effects that are typically a part of the conveyor system availability circuit are governed by the following:

- Electrical lines
- Feeder transformers to motor
- Motor contactors
- Motors (bearings, insulation, winding)
- Motor controls (fluid coupling, et al)
- Gearboxes and auxiliaries (bearings, gearing)
- Drive couplings
- Pulleys (shaft, rim, end disk)
- Pillow block bearings
- Takeup assembly
- Instrumentation
- Chutes, plugging
- Holdbacks/brakes
- Belt repair (rips)
- Belt splices
- Power outages
- Weather shutdowns

Information for electrical MTBF and MTTR data is referenced in published data of IEEE Committee Report (1974), and Yu (1978). The MTBF on reducers and pillow block bearing can be estimated from the L-10 life hours. The MTBF of bearings is typically given as approximately four times the L-10 life (Shigley, 1963). Therefore, by example, the MTBF of a two-stage reduction gearbox with two identical bearings per shaft would be:

$$MTBF = \left[\frac{2}{\frac{1}{L1} + \frac{1}{L2} + \frac{1}{L3}} \right] \tag{30}$$

where:

- L1: L-10 life hours of the low speed shaft bearings
- L2: L-10 life hours of the intermediate shaft bearings
- L3: L-10 life hours of the high speed shaft bearings

INTRODUCTION TO MONTE CARLO SIMULATION METHODS

Each production center should be designed for the most cost effective plant flow rate (CEFR). The FRSF for the plant is made up of the individual CEFR values for all process centers. It is somewhat complex to make an evaluation of the best CEFR at each station when all of the plant facilities are interacting. Therefore, modern methods are employed today with the use of computers, to formulate such manners of investigation.

Historically, the FRSF has been selected through experience factors. Typically, the FRSF has been set to a value of 1.25 times the BFR. Lack of understanding of the system interactions traditionally leads to an overdesigned rule-of-thumb FRSF. Overdesigned equipment and stockpile sizes are specified to compensate for potential production loss due to adverse weather conditions and/or prolonged equipment downtime. The breakeven economic tradeoff factors of risk capital versus production losses are normally not quantified. A modern method of quantifying the FRSF, including the interaction of equipment performance, stockpile sizes, operating constraints, and seasonal weather, variations is through the use of stochastic simulation of the process. Principles of these techniques have been referenced in the articles of Cruz et al (1981), Lakey (1979) and Sajkiewicz. The techniques of Monte Carlo simulation are well suited to investigating the stochastic behavior of alternative system concepts.

The Monte Carlo simulation technique is patterned after its namesake place of chance. With a spin of the wheel of chance, you either observe the event (you win) or you don't (you lose). The wheel of numbers represents a random set of occurences. Each occurence represents an observation in time. By setting the availability factors (random numbers on the wheel) that correspond to the expectation of continued production of each constituent event (shovel, truck, crusher, conveyor, weather, power, et al) their interaction can be simulated.

This is accomplished by sequentially spinning the wheel for each successive function in the process circuit, applying production constraints where applicable. For a win, the production continues. For a loss (equipment failure, etc.), the production stops at the given event and any consequences broadcast by the event until time corrects the shutdown (equipment is repaired, the storm passes, et al).

Monte Carlo modeling can integrate most of the major activities of the process. Maintenance programs can be identified in terms of manpower, equipment, and parts inventory. As historical records are tabulated, ongoing simulation can be used to improve future expectation for plant planning and prospects for expansion. Capital expenditure can be directed to those items identified to be production bottlenecks with the greatest net return on investment. It is the object of the simulation study to evaluate alternative concepts such that a process configuration can be found which yields the greatest overall performance.

COMPUTER AIDED DESIGN (CAD)

Today, the engineer is faced with more complex systems. He is required to make decisions with far greater cost implications as the transport systems increase in size and scope. Unfortunately, the time allocated to engineer such systems has not increased proportionately, Therefore, more engineers have turned to computer systems for design assistance.

One such program is presented here in Figure 7 and 8 . The program is tradenamed BELTSTAT. It is used for detailed analysis of belt and motor sizing, idler, pulley, controls and brake selections, and other features necessary to good conveyor design.

BELTSTAT is patterned after the CEMA engineering methods. The conveyor is analyzed for all operating states, such as:

1. Fully or partially loaded belts

2. Summer and winter conditions

3. Running, starting, and stopping

4. Multiple drive placements

BELTSTAT evaluates the conveyor belt as an in-elastic body (non-stretching).

Although an in-elastic analysis is sufficient for the majority of conveyors designed, it is sometimes desirable to study the starting and stopping transient phases. Beltflex is a computer-assisted design tool which models the elasto-mechanical transient behavior during starting and stopping. Its value is in predicting excessively high or low transient forces in the conveyor system which can overload components or cause other harmful effects. It is primarily used as the final step in refining or validating the design, for chosen control concepts to start and stop the system. A typical analysis is illustrated in Figure 9 for general reference.

Many of the factors discussed in this paper are now utilized in computer aided design. All aspects of the conveyor can be studied to maximize its overall utility.

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JOB NUMBER : SME-AIME PRESENTATION	968 N. MADISON AV. PASADENA, CA. 91104	DATE: 04-FEB-82
CONVEYOR NO. : 1A	BELTSTAT 2.5 : CONVEYOR DESIGN PROGRAM	RUN NO: 3

REMARKS: CASE 2: FULLY LOADED, FULL FRICTION, 15 SECOND START, 5 SECOND BRAKE, HEAD DRIVE, TAIL BRAKE

MATERIAL SPECIFICATIONS

1. MATERIAL CONVEYED	ALUVIAL	4" x 0"
2. DESIGN TONNAGE	2000.00	STPH
3. DESIGN BELT X-SECTIONAL LOADING ALLOW	100.00	PCT
4. BULK DENSITY	100.00	LBS/CU FT
5. SURCHARGE ANGLE	20.00	DEG
6. LUMP SIZE AND PERCENTAGE	4.00	IN X 60.00 PCT
7. LUMP SHAPE FACTOR	1.40	
8. CHUTE DROP DISTANCE	6.00	FT
9. ABRASIVE INDEX	MODERATE	
10. ENVIRONMENTAL CONDITION .(CEMA INDEX)	MODERATE	
11. MAINTENANCE CONDITION ...(CEMA INDEX)	FAIR	
12. HOURS IN SERVICE PER DAY............	24.00	HRS
13. TEMPERATURE RANGE	25.00	TO 100.00 DEG F

BELT SPECIFICATIONS

1. WIDTH	42.000	IN (18.9 /LUMPS)
2. SPEED	617.727	FPM
3. TYPE OF BELT CARCASS STEEL/FABRIC ...	FABRIC	
4. TENSION RATING	350.000	PIW
5. TENSION UTILIZATION RUNNING	98.675	PERCENT OF RATED
ACCEL/DECEL	112.700	PERCENT OF RATED
6. WEIGHT	12.992	LBS
7. COVER THICKNESS	0.187	X 0.062 IN
8. ELASTICITY	840.002	LBSX1000
9. SAG ALLOWABLE ON CARRY SIDE, PCT	1.500	PCT
10. EDGE DISTANCE / BED DEPTH...........	4.942	IN/ 7.242 IN
11. X-SECTIONAL AREA AVAIL (100% OF CEMA)	196.064	SQ IN
12. X-SECTIONAL AREA AVAIL (NO EDGE DIST)	281.385	SQ IN
13. X-SECTIONAL AREA UTILIZED (CEMA)	155.408	SQ IN
14. X-SECTIONAL LOADING PERCENTAGE (CEMA)	79.264	PCT
15. IMPACT FORCE FROM LUMPS	33.600	LB-FT
16. TAPE LENGTH (NOT INCL SPLICE LENGTH)	1761.499	FT

IDLER AND ANCILLARY SPECIFICATIONS

	CARRY	RETURN
1. IDLER SERIES	D6	C6
A. TROUGH ANGLE	35.00 DEG	
B. DIAMETER	6.00 IN	6.00
C. CORRECTED LOAD CAPACITY	879.66 LBS	114.85
D. APPLIED LOAD AT MAX SPACING	725.49 LBS	129.92
E. ROTATING WEIGHT	49.20 LBS	43.30
F. SEAL DRAG (Ai)	1.50 LBS	0.50
G. NUMBER OF IDLERS (APPROX)	149.	84.
2. (KY) TROUGH SHAPE MULTIPLIER	1.075	1.000
3. (KY/KX) CORRECTION (REGENERATION) ...	1.000	
4. BREAKAWAY FRICTION MULTIPLIER	1.500	
5. (KT) TEMPERATURE ADJUSTMENT (KY/KX)	1.026	
6. IDLER SEAL CORRECTION (REGENERATION)	1.000	
7. SKIRTBOARD FRICTION FACTOR	0.100	
WIDTH	28.000 IN	
MAXIMUM MATERIAL HEIGHT ..	2.216 IN	

MOTOR / REDUCER / BRAKE SPECIFICATIONS

1. LOCATION OF DRIVE / BRAKE UNITS	9	20	0
2. MOTOR NAMEPLATE HORSEPOWER	125.0	0.0	0.0
RUNNING HORSEPOWER	119.1	0.0	0.0
3. POWER RATIO	1.000	0.000	0.000
4. MOTOR SYNCHRONOUS RPM	1800.0	78.3	0.0
RUNNING RPM	1752.5	78.3	0.0
5. BREAKAWAY TORQUE (PCT FULL-LOAD-TORQUE)	105.285	0.000	0.000
6. STARTING TORQUE LIMIT (PCT FULL-LD-TQ)	138.149	0.000	0.000
7. DRIVE INERTIA AT MOTOR ......(LB-FT-SQ)	33.3	0.0	0.0
8. DRIVE EFFICIENCY	0.950	0.950	0.000
9. DRIVE WRAP ANGLE ....(DEGREES)	200.000	190.000	0.000
10. DRIVE FRICTION FACTOR, RUNNING	0.350	0.350	0.000
ACCEL/DEC	0.400	0.400	0.000
11. GEARBOX RATIO	27.852	1.000	0.000
12. BRAKE TORQUE LOW-SPEED ....(FT-LBS)	0.0	3303.3	0.0
13. BRAKE ENERGY ABSORBED ....(HP-SECS)	0.0	123.7	0.0

14. ACCELERATION	TIME	15.000 SEC	TRAVEL:	77.22
15. DRIFT	TIME	7.111 SEC	TRAVEL:	36.61
16. BRAKING	TIME	5.000 SEC	TRAVEL:	25.74

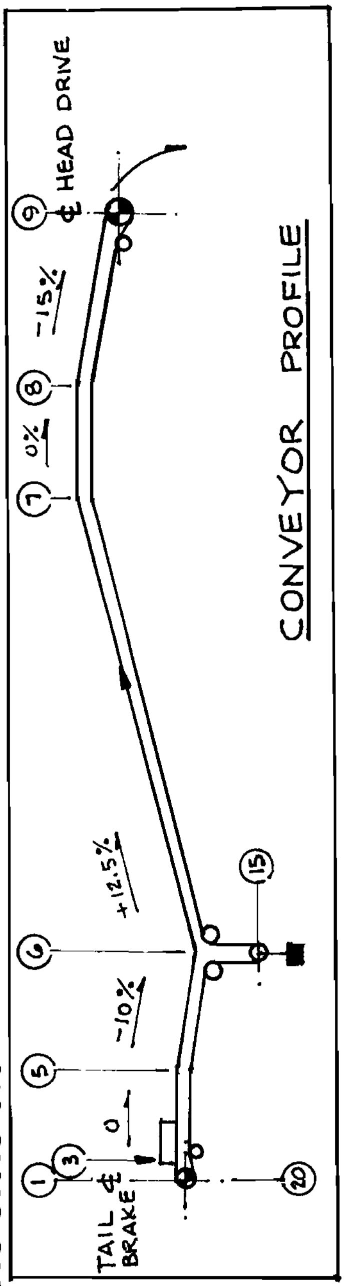

FIGURE 7.

CLIENT : SAMPLE RUN
JOB NUMBER : SME-AIME PRESENTATION
CONVEYOR NO. : 1A

C O N V E Y O R   D Y N A M I C S ,   I N C.
968 N. MADISON AV. PASADENA, CA. 91104
BELTSTAT 2.5 : CONVEYOR DESIGN PROGRAM

PAGE NO: 2
DATE: 04-FEB-82
RUN NO: 3

REMARKS: CASE 2: FULLY LOADED, FULL FRICTION, 15 SECOND START, 5 SECOND BRAKE, HEAD DRIVE, TAIL BRAKE

FLITE NO	LENGTH (FT)	HEIGHT (FT)	IDLER SPG (FT)	SAG TEN (LBS)	KY	KX	WM (LB/FT)	WT (LB/FT)	MASSES (SLUGS)	LOADING (PCT)
1	10.0	0.0	10.00	1032.7	0.0336	0.1591	0.00	12.99	6.	0.0
2	5.0	0.0	2.50	1082.7	0.0171	0.6091	0.00	12.99	5.	0.0
3	10.0	0.0	1.25	1259.5	0.0181	1.2846	107.92	120.91	50.	100.0
4	75.0	0.0	6.00	6045.7	0.0351	0.3346	107.92	120.91	301.	100.0
5	100.0	-10.0	6.00	6045.7	0.0354	0.3346	107.92	120.91	403.	100.0
6	400.0	50.0	6.00	6045.7	0.0290	0.3346	107.92	120.91	1616.	100.0
7	100.0	0.0	6.00	6045.7	0.0223	0.3346	107.92	120.91	401.	100.0
8	150.0	22.5	6.00	6045.7	0.0239	0.3346	107.92	120.91	608.	100.0
9	3.6	-3.0	0.00	6045.7	0.0150	0.0000	0.00	12.99	405.	0.0
10	5.0	3.0	0.00	0.0	0.0150	0.0000	0.00	12.99	2.	0.0
11	145.0	21.8	10.00	406.0	0.0150	0.0591	0.00	12.99	82.	0.0
12	100.0	0.0	10.00	406.0	0.0150	0.0591	0.00	12.99	54.	0.0
13	398.0	-50.0	10.00	406.0	0.0150	0.0591	0.00	12.99	216.	0.0
14	0.0	-20.0	0.00	406.0	0.0150	0.0000	0.00	12.99	18.	0.0
15	4.7	0.0	0.00	0.0	0.0150	0.0000	0.00	12.99	15.	0.0
16	0.0	20.0	0.00	0.0	0.0150	0.0000	0.00	12.99	8.	0.0
17	93.5	10.0	10.00	406.0	0.0150	0.0591	0.00	12.99	63.	0.0
18	95.0	0.0	10.00	406.0	0.0150	0.0591	0.00	12.99	51.	0.0
19	5.0	-1.8	0.00	406.0	0.0150	0.0000	0.00	12.99	7.	0.0
20	3.0	2.5	0.00	0.0	0.0150	0.0000	0.00	12.99	12.	0.0

STATION PT.	ITEM	TENSION SPECIFICATIONS: RUNNING (LBS)	BREAK'WY (LBS)	ACCEL. (LBS)	BRAKE (LBS)	DRIFT (LBS)	CON WT (LBS)	MISC. DRAG (LBS)	CURVE RADIUS (FT)	PULLEY DIAMETER (IN)	ESTIMATED SHAFT (IN)
1	TAIL	6672.	6853.	6824.	9039.	6491.	0.	0.0	0.	0.	0.000
2	SKRTBDS	6678.	6863.	6835.	9034.	6490.	0.	0.0	0.	0.	0.000
3	LOAD ST	6603.	6370.	6843.	9028.	6487.	0.	207.3	0.	0.	0.000
4		6926.	7224.	7120.	9168.	6658.	0.	69.8	0.	0.	0.000
5	CONVX R	7340.	7704.	7741.	3964.	6637.	0.	0.0	51.	0.	0.000
6	CONCV R	6597.	7003.	7274.	7391.	5311.	0.	0.0	612.	0.	0.000
7	CONVX R	14200.	14753.	15937.	11666.	10574.	0.	0.0	749.	0.	0.000
8	CONVX R	14505.	15090.	16567.	11145.	10298.	0.	0.0	1923.	0.	0.000
9	HEAD DR	12276.	12910.	14755.	7663.	7188.	637.	0.0	0.	37.	5.512
10		6222.	6039.	5967.	6989.	6761.	0.	126.0	0.	0.	0.000
11	BEND P	6333.	6269.	6134.	7150.	6924.	116.	31.0	0.	16.	3.253
12		6740.	6655.	6543.	7332.	7156.	0.	0.0	0.	0.	0.000
13		6766.	6694.	6606.	7247.	7104.	0.	0.0	0.	0.	0.000
14	BEND P	6221.	6202.	6209.	6258.	6247.	314.	34.9	0.	30.	4.529
15	TAKE-UP	6000.	6000.	6000.	6000.	6000.	417.	36.3	0.	36.	4.964
16		6037.	6056.	6047.	6007.	6016.	0.	0.0	0.	0.	0.000
17	BEND P	6301.	6322.	6317.	6254.	6268.	316.	35.1	0.	30.	4.543
18		6492.	6543.	6551.	6315.	6367.	0.	0.0	0.	0.	0.000
19	BEND P	6517.	6580.	6611.	6234.	6318.	167.	17.4	0.	20.	2.752
20	DRIVE 2	6512.	6585.	6612.	6215.	6303.	321.	126.0	0.	30.	5.050

AISI 1045 STEEL

SUMMARY	RUNNING	BREAK'WY	ACCEL.	BRAKE	DRIFT
(TE1)	6016.	6334.	8750.	636.	389.
(TE2)	0.	-45.	-53.	-2664.	-29.
(TE3)	0.	0.	0.	0.	0.
LIFT FORCE	1889.				
FRICT FORCE	3443.				
MISC FORCE	684.				
WR FACTR (1)	3.393	4.040	4.040	4.040	
T1/T2 (DR 1)	1.973	2.138	2.473	1.097	1.063
WR FACTR (2)	3.192	3.768	3.768	3.768	
T1/T2 (DR 2)	1.024	1.041	1.032	1.454	1.030
TOTAL MASS	4323.				

DRIVE PULLEY DRAG:

DRIVE 1: 45.54 LBS

DRIVE 2: 45.47 LBS

COUNTERWEIGHT SPECIFICATIONS (DIMENSIONS IN FEET):					
SPLICE ALLOWANCE	0.00				
THERMAL TRAVEL	0.00				
CREEP ALLOWANCE	1.41				
TENSION TRAVEL:	1.07	1.23	1.40	1.50	0.43
TOTAL TRAVEL:	2.47	2.64	2.81	2.91	1.84

FIGURE 8

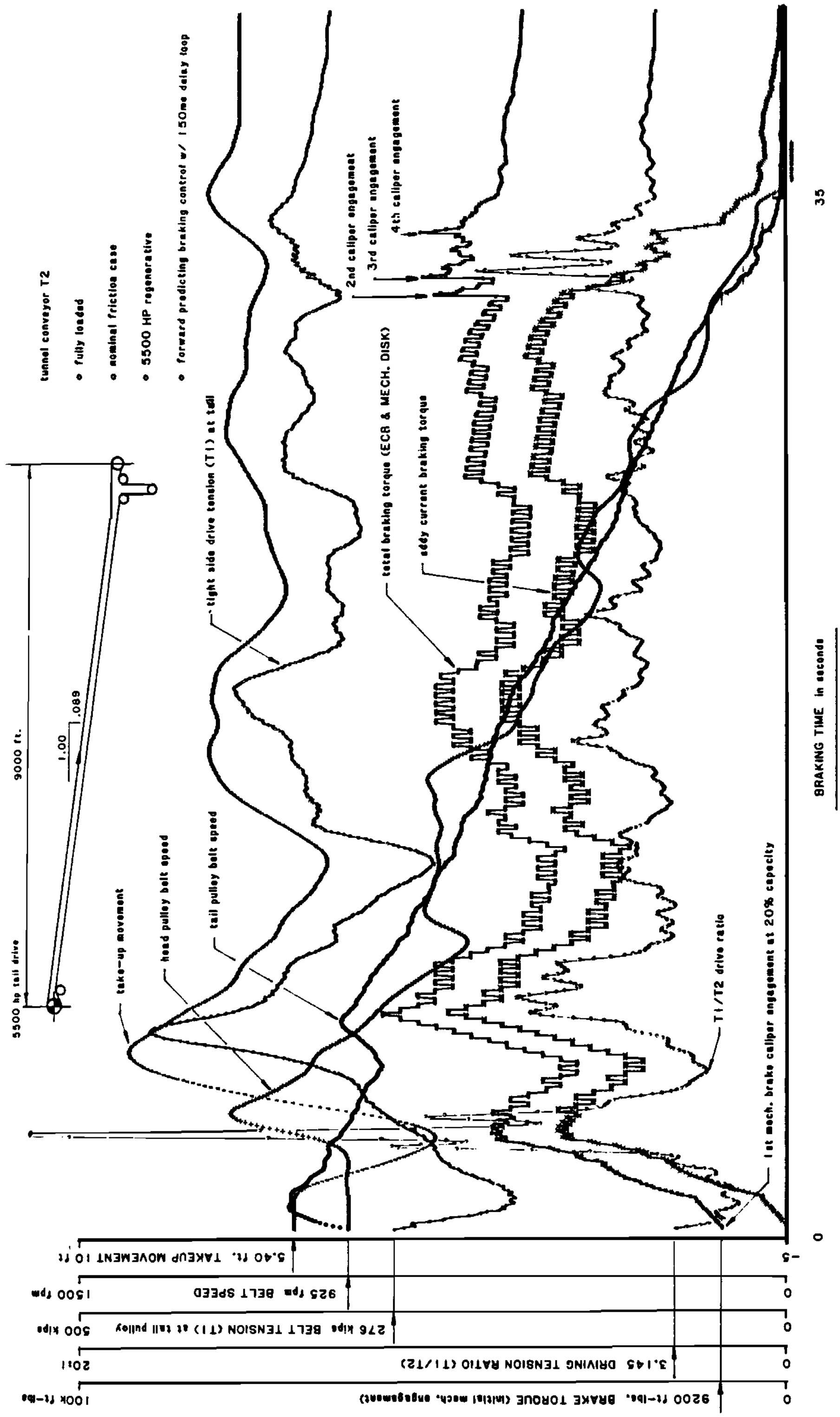
tunnel conveyor T2
• fully loaded
• nominal friction case
• 5500 HP regenerative
• forward predicting braking control w/ 150ms daisy loop
5500 hp tail drive
9000 ft.
1.00
.089
take-up movement
head pulley belt speed
tail pulley belt speed
tight side drive tension (T1) at tail
total braking torque (ECB & MECH. DISK)
eddy current braking torque
2nd caliper engagement
3rd caliper engagement
4th caliper engagement
T1/T2 drive ratio
1st mech. brake caliper engagement at 20% capacity
BRAKING TIME in seconds
0
35
BELTFLEX ELASTIC-TRANSIENT ANALYSIS April 8 82
-5 5.40 ft. TAKEUP MOVEMENT 10 ft
0 925 fpm BELT SPEED 1500 fpm
0 276 kips BELT TENSION (T1) at tail pulley 500 kips
0 3.145 DRIVING TENSION RATIO (T1/T2) 20:1
0 9200 ft-lbs. BRAKE TORQUE (initial mech. engagement) 100K ft-lbs

FIGURE 9.