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OIL SAND CONVEYING

by

DOUGLAS STEWART BARTH ©

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The undersigned certify that they have read, and recommend to the Faculty of Graduate Studies and Research, for acceptance, a thesis entitled **OIL SAND CONVEYING** submitted by Douglas Stewart Barth in partial fulfilment of the requirements for the degree of Master of Science in Mining Engineering.

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Supervisor

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Date ..DECEMBER 07, 1990...

Dedication

To my wife Richelle and to my parents

for all of their support

and especially their continued

encouragement over the past three years

Abstract

Belt conveyors are one of the most efficient and economical means of transporting bulk solids. Because of the vast amounts of oil sand that must be processed to achieve the required synthetic oil production, belt conveyors are the logical choice to transport the oil sand from the mine to the extraction plant.

Two factors make oil sand conveying unique. The first is the harsh climate of northern Alberta with operating temperatures ranging from minus 40°C to plus 30°C. The second is the stickiness of the bitumen within the oil sand, which causes the oil sand to adhere to the belt and idlers, thereby increasing the power required to drive the belt.

Original designs for the oil sand conveyors were based primarily on the conveyors operating in the European lignite mines. However, once installed, these conveyors did not operate with the efficiency and reliability anticipated. As a result, much of the present design has been the result of trial and error testing after a conveyor was in operation. Defining some of the unknown parameters that make oil sand conveying unique will enable more accurate and reliable design of belt conveyor belt tensions and power requirements.

This thesis investigates two areas in which little research has been published. The first studies examined the effect of temperature on belt bending, oil sand stickiness and density. The second, through field studies, determines a reliable value of the surcharge angle of oil sand, and the factors that influence this value.

Tests were performed to determine the degree to which temperature variations effect the force required to bend oil sand belting, and the density of oil sand. In addition, power measurements were taken on an existing conveyor with the values compared to theoretically calculated

power requirements in order to evaluate the degree to which the oil sand stickiness increases power requirements. Finally, field tests were performed, measuring the value of the surcharge angle of oil sand enabling more accurate calculation of the mass conveying rate.

The results of these field studies can be utilized when calculating theoretical belt tensions and power requirements, leading to more reliable, and accurate conveyor designs.

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List of Symbols

C_s - skirtboard friction factor

C_w - total wrap factor

C_{w1} - primary wrap factor

C_{w2} - secondary wrap factor

d - edge distance

H - distance material is lifted or lowered

H_s - height of material resting against skirtboards

K_t - temperature correction factor

K_{xc} - factor for carry idler rotation resistance

K_{xr} - factor for return idler rotation resistance

K_{yc} - factor for material and carry belt flexure resistance

K_{yr} - factor for return belt flexure resistance

L_b - skirtboard length

L_c - carry side belt length

L_r - return side belt length

Q - conveyor capacity

t - time

T_{am} - force required to accelerate material

T_{bc} - resistance of belt cleaners

T_e - effective tension

T_m - force required to lift material

T_p - primary effective tension

T_{pl} - resistance of plows

T_s - secondary effective tension

T_{sb} - resistance of skirtboards

T_{xc} - frictional resistance of carry idlers

T_{xr} - frictional resistance of return idlers

T_{yc} - carry belt flexure resistance

T_{ym} - material flexure resistance

T_{yr} - return belt flexure resistance

T_o - minimum allowable tail tension

T_1 - maximum carry side belt tension

T_2 - slack side belt tension

T_3 - intermediate belt tension

V - belt velocity

V_o - material feed velocity

W_b - belt unit weight

W_m - material unit weight

α - surcharge angle

β - troughing angle

1. INTRODUCTION

1.1 Objective of Thesis

The objective of this thesis is to better define some of the physical parameters and characteristics of oil sand and oil sand conveying to allow more reliable and accurate prediction of conveyor belt tensions. In conjunction with this, a computer program was developed to enable quicker and more efficient design, thus allowing the engineer to analyze numerous belt conveyor layouts more efficiently.

1.2 Studies Undertaken

Although the use of modern belt conveyors in the mining industry dates back over 50 years, their use in oil sand mines is relatively new (approximately 20 years). This, coupled with the fact that there are only two oil sand mines in operation, has resulted in comparatively little research and development into the operation of oil sand conveyors.

Two areas where data on oil sand conveying are lacking are: 1) the effect of ambient temperature on conveyor operation, and 2) a value for the surcharge angle. Tests were performed to determine the effect that the ambient temperature has on the force required to flex and bend the belting, as well as its effect on the oil sand stickiness. Also, field investigations were carried out to define an accurate value for the surcharge angle and the factors that influence it. Many computer programs are available to aid in the design of belt conveyors. None, however, have the ability to base their calculations on the harsh operating conditions of oil sand conveyors (i.e., extreme cold, unique material physical properties, and specially-designed belting). The programs developed as part of this thesis specifically address these problems.

1.3 Thesis Layout

Chapter 2 contains general information on belt conveyors regarding their main components and a brief history. The advantages and disadvantages of belt conveyor use in the mining industry are addressed, as well as problems that have been encountered in oil sand conveying.

Chapter 3 discusses four widely-used design standards for calculating belt tensions. These are: Conveyor Equipment Manufacturers' Association (CEMA), Goodyear Tire and Rubber Company, Mechanical Handling Engineers' Association (MHEA), and the German Industrial Standards (DIN 22101). Each method is discussed, stressing the differences in recommended procedures, and the subsequent influence on the correctness of the resulting conveyor design.

Chapter 4 briefly discusses three computer software packages that are currently available. The programs that form part of this thesis are then analyzed in detail. Appendix A illustrates the program use with two sample conveyors, their respective input data files and output. Appendix B has documented source code of the subroutines that comprise the programs.

Chapters 5 and 6 relate the findings of the field research. Chapter 5 discusses the effect that the ambient temperature has on belt bending and flexing forces, as well as oil sand stickiness. Chapter 6 contains a summary of the surcharge angle measurements and the findings.

Conclusions, and recommendations that pertain to mining operations, and areas of further research with respect to the effects of ambient temperature on belt conveyor operation in oil sand mines, are presented in Chapters 7 and 8, respectively.

Tables and figures are numbered according to the chapters in which they are contained, with the first character representing their chapter (or appendix). Equations are also presented this way. References are numbered in alphabetical order, and only the reference numbers are used in the thesis text as shown in square brackets.

2. BELT CONVEYORS

The belt conveyor is one of the most common means of transporting bulk solids. Its main component is an endless loop of belt stretched between two pulleys and supported at intervals by idler rollers. Figure 2.1 illustrates a simple troughed belt conveyor and the related terminology.

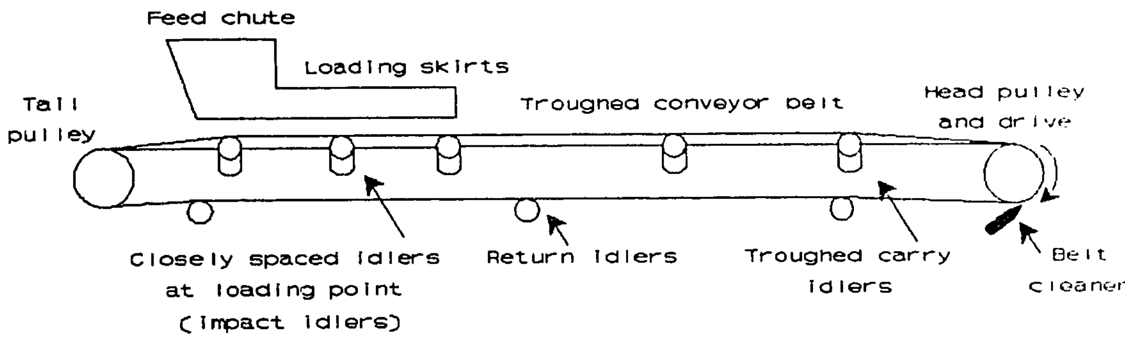

Figure 2.1 Idealized troughed belt conveyor

Transporting bulk materials on belt conveyors dates back to the late 1700's when conveyors were first used to transport grain. These early conveyors used leather, canvas or rubber belting travelling over a flat or troughed wooden bed [7]. One of the earliest large-scale conveyor installations was introduced in 1924 in Pennsylvania. It carried approximately 9000 tonnes per day on a belt made from cotton duck plies covered with natural rubber. Although still relatively primitive, this and other installations proved that belt conveyors could reliably transport large quantities of bulk material over long distances.

Today with high-strength steel cable belts, much larger and longer belts are feasible. A 15-km conveyor with a 900 meter lift in the Selby mine, U.K., is capable of delivering 3200 tonnes of coal per hour at a speed of 8.4 m/s [2]. Conveyors used in the lignite mines of Germany are fed by bucket wheel excavators that produce an average of 240 000 bank cubic meters per day (BCM/day) with peaks of up to 350 000 BCM/day. Another belt conveyor, reported to be one of the largest in the world, is located in Japan and successfully conveys 30 000 tonnes of sand and rock per hour at a speed of 5.3 m/s on a 3 m wide belt.

2.1 Belt Conveyor Applications

The basic advantages of belt conveyors over other means of material transport, such as truck or rail, are as listed below:

Wide range of materials

Belt conveyors are capable of handling a wide variety of materials. The size of material that can be transported is limited by the width of the belt, and can vary from fine chemicals to run of mine ore, coal or pulpwood. Because synthetic rubber can be highly resistant to corrosion and abrasion, conveyors are capable of carrying corrosive and abrasive materials. In addition, friable materials are best transported by conveyors because the smooth ride limits degradation.

Wide range of capacities

Belt conveyors, a continuous mode of bulk material transportation, are capable of transporting relatively small quantities of material of less than 800 t/h to very large quantities of material of over 10 000 t/h.

Adaptability to ground profile

Conveyors are capable of transporting material up inclines (slopes) as high as 30% (limited by material angle of surcharge), allowing them to follow most natural terrains. Covered conveyors may even climb inclines steeper than this.

Reliability

The reliability and availability of belt conveyors is among the highest of bulk material transportation modes. They are capable of operating continuously for extended periods of time without having to stop for maintenance, adverse weather or refuelling.

Minimum labour requirements

The labour required to transport material on a belt conveyor is among the least of any method used for transporting bulk materials. Start-up and shut-down operations of several belt conveyors can be controlled and monitored by one person from a central control room, while a minimum number of personnel are required to inspect the belt conveyors for maintenance requirements.

Low maintenance

Maintenance requirements are considered minimal compared to that of other transportation systems. Extensive support systems (i.e., lubrication trucks, water trucks, road construction and road maintenance) are not required. Most repairs or replacements that are required can be predicted in advance, and scheduled to reduce unwanted downtime.

Low power costs

Conveyors require little power to transport bulk materials, use inexpensive electric power, and are less affected than truck transport by fuel prices. They consume energy only when being used, with little idling time and no empty return trips. In addition, downhill conveyors, if steep enough, will generate power that may be used elsewhere.

Environmentally acceptable

Belt conveyors are environmentally more acceptable than other bulk material transportation methods. Being electrically powered, they produce no emissions and generate relatively little noise. Also, they do not release significant amounts of dust; most dust is easily contained at transfer points.

Mobility / Extensibility

Capabilities that are particularly important to the mining industry are that shiftable conveyors can be laterally moved, extended or shortened to follow the mining face. Belt conveyors can also accept material at multiple feed points, allowing them to be fed by two or more bucketwheels, or feed conveyors, and enabling them to follow the advancement of mining benches.

Safety

Material transportation with belt conveyors is inherently safer than most other methods of material transportation. Fewer personnel are required for operation; therefore, fewer people are exposed to hazards. Also, the driving hazards associated with truck haulage are not present when employing belt conveyor transportation.

Long distance transportation

As a result of the low operating costs for labour and energy, as well as other previously mentioned advantages, the belt conveyor has become an excellent means of transporting materials over long distances.

2.1.1 Disadvantages of belt conveyors

In certain applications, there are distinct disadvantages to using belt conveyors.

Inflexibility

The belt conveyor is an inflexible mode of bulk material transportation. It is a relatively permanent installation that can not be readily moved. Shiftable conveyors may require several days to shift, realign and start up. This results in significant system downtime. Once a conveyor is constructed for a certain application, it must remain at or near its original position.

No horizontal curves

Conveyors must travel in straight lines. Therefore, during construction of belt conveyors, obstacles in a conveyor's line must be removed, or alterations made in the conveyor path. Other modes of transportation such as trucks, slurry pipelines and railways can have a curved path of travel.

High initial capital cost

It should be noted that either large volumes of material must be transported, or the haul distances must be long enough to warrant the need to install a belt conveyor. The capital costs incurred with a conveyor installation are significant, and large volumes or distances must be encountered to realize any appreciable economic benefits.

2.2 Conveyor use in mining

The belt conveyor is becoming more commonplace in the mining industry for use in bulk material transportation. This use has resulted from the need to transport larger quantities of materials over longer distances, more economically, and with a greater degree of reliability.

In surface mining, three types of belt conveyors are used:

a) shiftable conveyors

b) overland conveyors

c) inclined conveyors

Shiftable belt conveyors consist of a series of sections mounted on pontoons. These pontoons are connected by rails which permit the entire conveyor to be moved laterally without dismantling any part of it. The shifting is done by using a tractor fitted with a shifting mechanism which locks onto the head of the shifting rail [16]. The tractor moves the conveyor to the desired location; up to one meter in each pass. Shiftable conveyors are typically used in applications requiring frequent conveyor relocations. The most common use is in open pit mines that utilize continuous mining equipment such as bucket wheels.

Shiftable belt conveyors have been widely used world-wide for decades in the lignite mines. At these mines, the shiftable conveyors are able to transport up to 350 000 BCM/day and, to follow the progress of the bucket wheel excavators.

The overland belt conveyor is a permanent structure. Unlike the shiftable conveyor, it is mounted on anchored supports, and no rails connect the supports. Typical of an overland conveyor application is the overland conveyor at the Obed Mountain Coal Company Limited. This conveyor transports clean, sized and dried hot coal from the coal dryer to the load-out point. The conveyor consists of a single flight 10.73 km long and has a capacity of 640 t/hr. It is 914 mm wide and runs at a speed of 3.9 m/sec. An interesting design aspect of this conveyor is that it is fully covered to protect the dried coal from the surrounding environment and to keep any coal dust from escaping to the atmosphere [15].

Another area in mining where belt conveyors are becoming more widely used is in deep open pit mines. The conveyor suitable for this application is the inclined conveyor. As open pit mines deepen, the haul distances for truck transportation become uneconomical. At present, the break-even point is approximately 200 m, where it becomes more economical o use a belt conveyor to transport material out of the pit. An in-pit crusher may be used in conjunction with the conveyor system to reduce oversized ore. Once the material is out of the pit, it can be fed onto overland conveyors, or any other means of transportation.

2.3 Conveyor use in Oil Sand Mining

Belt conveyors are widely used for transporting material in oil sand mines. To date, only the oil sand is transported via conveyor, but as overburden depths increase, conveyors may also replace the present truck haulage of overburden.

An oil sand plant with a synthetic oil production of 20 700 m^3/day (125 000 barrels/day), requires approximately 245 000 tonnes of feed each day, or 10 200 tonnes per hour. Some forty 154-tonne haul trucks would be required to move this much material assuming two cycles per hr, 24 hr/day, and an 80% availability.

Oil sand mining is an ideal belt conveyor application because:

1. Very high transportation rates can be achieved.
2. Continuous loading/mining is possible.
3. Conveyors can operate in very low temperatures.
4. The material size of the oil sand is small, and therefore, suited for belt conveyors.

2.3.1 Oil sand conveying

Oil sand mining, and the related use of belt conveyors for oil sand transport is relatively new. Lack of experience lead to incorrect assumptions with respect to the physical properties of the oil sand. In addition, the environmental conditions were not fully assessed. With insufficient data regarding the effect of the oil sand properties and the harsh climate, the following major problems have been encountered:

1. Oil sand stuck to both belts and idlers. This greatly added to the belt weight, and reduced its flexibility. It also acted as an

adhesive between the belt and idler rollers, increasing the force required to rotate the idlers. As a result more power was required to drive a belt conveyor used in oil sand transportation service than in other, more conventional applications. Belt cleaning lessened this problem somewhat, but did not eliminate it.

The idler rollers first used were steel shell, which were commonly used elsewhere. The oil sand build-up on the steel shells was not uniform, resulting in different diameters along the length of the rollers. This, in turn, created a differential speed between the belt and parts of the rollers, resulting in strong idler vibrations, and rapidly destroying the idlers. This problem has been somewhat alleviated by replacing the steel shell idlers with "self-cleaning" rubber ring idlers.

2. Other major problems were the result of the cold weather. The cold increased the force required to flex the belt and rotate the idlers (according to CEMA, a temperature of -40 °C increases these resistances three fold). Conveyors never had operated in such extreme temperatures when the oil sand mines started, and the belting was not designed for such cold temperatures.

Another problem resulting from the cold was that as the oil sand built up on the belt and froze to it, the belt and oil sand acted as one solid unit. The impact from belt loading created a crack in the oil sand cover, and the crack propagated through the belt cover [10]. As a result, special cold weather belt covers had to be developed to resist the cold weather, and the impact from the frozen lumps.

After many years, the operators of the existing oil sand mines have adapted to the inherent problems associated with belt conveyor transportation of oil sand. They have learned, through extensive research and much trial and error testing, how to successfully operate belt conveyors in oil sand mines.

3. BELT TENSION CALCULATION METHODS

Four belt conveyor design standards are presently in wide use; defined by the Conveyor Equipment Manufacturer's Association (CEMA) [3], Goodyear Tire and Rubber Company (Goodyear) [7], Mechanical Handling Engineers' Association (MHEA) [10], and the German Industrial Standards (DIN) [4]. Each is discussed below, stressing the differences in recommended procedures and their influence on the correctness of the resulting conveyor design.

3.1 Conveying Rate

The conveying rate of a conveyor is defined by the cross-sectional area of the material being transported, its density, and belt speed. While density and belt speed have the same values regardless of the calculation method, substantial differences exist in the approach recommended for calculation of the cross-sectional area.

3.1.1 Calculation of material cross-sectional area

All of the methods recognize that the cross-sectional area is a function of the troughing angle, β, effective belt width, center roller length, and material surcharge angle, α. The cross-sectional area is divided in two parts; the struck (water level) area, and the surcharge area. All four methods approximate the struck area shape as a trapezoid, and calculate it as such. The surcharge area, however, is calculated in two different ways. CEMA, Goodyear and MHEA assume the shape of the surcharge to be a circular segment (see Figure 3.1), whereas DIN assumes it to be a triangle (see Figure 3.2). The DIN assumption makes calculation of the surcharge area mathematically simpler. To avoid over-estimation of this area, as the actual shape of material is more closely approximated by an arc, DIN chooses the acute angle of the triangle to be two-thirds that of the surcharge angle of the material.

In addition, DIN includes two factors that reflect the material properties and operating conditions which reduce the cross-sectional area. The first is a filling factor intended to compensate for the lumpiness of the conveyed material, the belt edge clearance, non-uniformity of feed and rectilinear run of the belt. The second is a factor that reduces the cross-section of material when conveyed uphill or downhill.

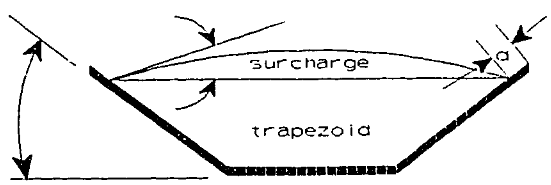

Figure 3.1 CEMA cross-section

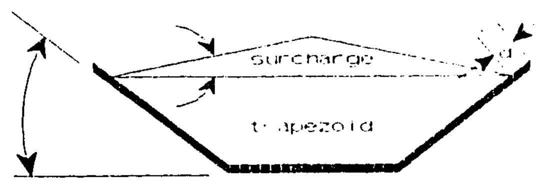

Figure 3.2 DIN cross-section

3.1.2 Edge clearance

The edge clearance, d, is the distance from the belt edge to the material toe on each side of the belt (see Figure 3.1) required to prevent material spillage. Edge clearance is assumed to correspond to belt width. While other methods recommend this approach for all belt widths, DIN assumes that the maximum edge clearance required to prevent spillage is 125 mm, and for belt widths over 2000 mm, recommends it to be constant at this value. This approach appears to be justified for well-aligned and maintained conveyors.

As a result, the material cross-sections of large conveyors, calculated using DIN recommendations, are somewhat larger than those calculated using the other discussed methods. The representative difference is shown in Table 3.1.

Table 3.1 Comparison of effective belt widths

Method	Nominal belt width 1500 mm	2000 mm
	Effective belt width	
CEMA	1288	2180
Goodyear	1288	2180
MHEA	1300	2200
DIN	1300	2250

3.1.3 Material cross-sectional area

CEMA, Goodyear and MHEA produce similar cross-sectional areas, while DIN gives a larger area. However, after applying the reduced surcharge angle, the DIN results are very close to the other three methods (see Table 3.2).

While all discussed methods yield similar cross-sectional areas, the DIN method is more complex, and thus, is more susceptible to errors. In addition, the practice of reducing the cross-sectional area for inclined conveying does not seem to be justifiable in view of the low inclinations involved, except for material with unusually low surcharge angles.

3.2 Effective Belt Tension

Effective belt tension is the minimum tension required to overcome conveyor friction forces, as well as material and belt resistance forces, during the steady operation of a belt conveyor. It is used to determine the drive power requirements, and the drive configuration.

Section 3.7 shows a comparison of effective tension calculations

Table 3.2 Cross sectional area comparison

Method	Nominal belt width 1200 mm	2000mm
	Area, m^2	
CEMA	0.175	0.506
Goodyear	0.183	0.503
MHEA	0.177	0.515
DIN	0.213	0.691
DIN (reduced angle)	0.177	0.517

for a sample conveyor using each of the four methods.

3.2.1 CEMA

CEMA recommends separate calculation of a number of component tensions (unit resistances), and adding these up to derive the effective tension. The belt is broken into upper and lower strands. The unit resistances include frictional resistance from idlers, belt flexure (carry and return), material flexure, gravity forces, pulley bearing resistance, material acceleration force, and accessory resistances such as belt cleaners, plows and skirt boards. Suitable friction factors and a temperature factor are incorporated into the belt and material flexure and idler friction calculations.

Separate calculation of various resistances may reduce the errors involved, because some unit resistances can be calculated rather accurately, thus improving the overall reliability of this method.

3.2.2 Goodyear

Goodyear defines only three components of effective tension; moving parts resistance (belt, idlers and pulleys), material resistance, and gravitational resistance. One composite friction factor accounts for all frictional resistances. It is a function of the idler type and spacing, belt tension, and conveyor structure and maintenance. The correctness of the friction factor determines the accuracy of the total frictional resistance. The "length factor" is a means of compensating for the constant frictional losses, such as belt cleaners and skirtboards [7]. It is utilized by adding between 45m and 145m to the total belt, depending on conveyor alignment and operating conditions.

Considering the general character of friction factors, this method is judged to be less accurate than the CEMA method.

3.2.3 MHEA

MHEA is similar to Goodyear in that it also distinguishes three major components of effective tension. These are frictional resistance of the belt and all moving parts, frictional resistance of the material, and gravitational resistance. MHEA uses two friction factors. The first friction factor accounts for the resistance of the belt and moving parts, while the second accounts for the resistance of the material.

Formulae developed as a function of these friction factors for various conveyor lengths provide reasonably accurate effective tension values. Similar to Goodyear, MHEA uses a corrected conveyor length instead of the actual one to account for fixed resistance. The values of corrected lengths are the result of extensive field studies.

This method's accuracy is comparable to that of Goodyear.

3.2.4 DIN

DIN breaks conveyor resistance into main, secondary, slope and special resistances. Main resistance results from the rotating parts, and moving the belt and material. One friction coefficient is used, common for all main resistances. Secondary resistance is calculated as a function of the main resistance. This is estimated using another friction coefficient, which accounts for frictional resistances between the material and belt at feed points, between material and loading chute skirtboards, belt cleaners, and resistance of the belt bending around pulleys. Slope resistance results from lifting or lowering the material. Special resistances result from camber, or conveyor structure misalignment.

The DIN method is judged to be somewhat more accurate than Goodyear and MHEA. When extensive data detailing values of friction coefficients related to typical mining applications is readily available, the accuracy and credibility of this method is enhanced.

Many German belt conveyor manufacturers use a design method called 'DIN' which is a unit resistance method similar to CEMA. It appears to be as accurate as CEMA, however, information pertaining to this design method is not in public domain.

3.2.5 Limitations of design standards

Use of the CEMA method is preferred when designing a conveyor to be used in a new environment with widely-varying climatic and operating conditions. This method breaks the resistance to movement into unit components, separate for carry and return belts, allowing individual analysis of each resistance. Thus, the influence of the operating environment on the belt tension may be quantified to a greater extent.

DIN, Goodyear and MHEA rely on friction factors which are difficult to quantify without extensive field investigations of existing conveyors. Where results from such investigations are available, the accuracy of the calculations is satisfactory.

A major limitation of the CEMA method, if used for mine conveyors, is that the results are only accurate for belt conveyors with tensions of up to 7258 kg (16000 lbs). For larger conveyors, the friction factors must be modified to be representative of the loads and tensions.

3.3 Belt Traction Forces

Once the required effective tension, T_e, has been calculated, one can define the theoretical slack side tension (pretension), T_2, and tight side tension, T_1 (see Figure 3.3). The wrap factor, C_w, provides the relationship between effective tension and slack side tension. It is a function of the wrap angle of the belt around the drive pulley and the friction coefficient between the belt and the pulley surface. T_2 is defined as:

$$T_2 = T_e * C_w$$

and T_1 as:

$$T_1 = T_2 + T_e$$

For single pulley drives, each of the four methods use the above relationships to define tensions; however, differences exist in the approach to multi-pulley drives. DIN and MHEA consider each pulley individually using the above relationships, while CEMA and Goodyear consider all driving pulleys to be one unit. Ideal tension distribution is assumed between the primary and secondary pulleys (i.e., the ratio of T_1/T_3 equal to T_3'/T_2). (See Figure 3.4). If this condition is met, a

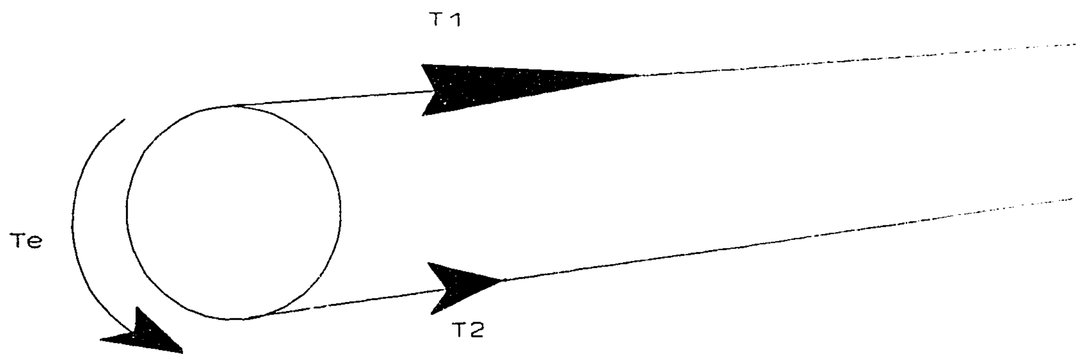

Figure 3.3 Tensions for single pulley drives

minimum amount of slip will occur between the belt and drive pulleys, and the results of the related calculations are reliable. If, however, the horsepower distribution between the drive motors differs, a result of varying pulley speeds, diameters, or other drive characteristics, no ideal tension distribution can be assumed. In this situation, each driving pulley must be treated separately.

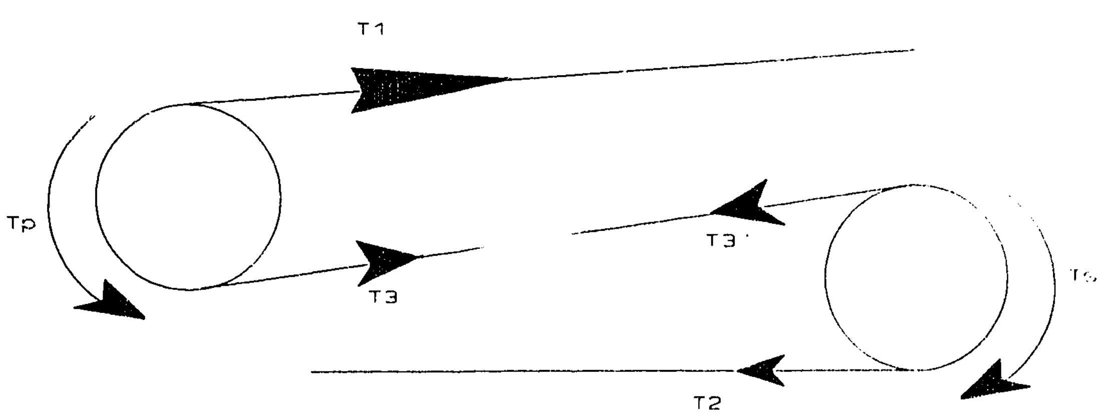

Figure 3.4 Tensions around a dual pulley drive

3.4 Start-up Tension

When a conveyor accelerates during start up, additional tension is induced in the belt. This force is a function of the accelerated mass, (including the conveyed material, belt and rotating parts) and the acceleration rate.

In related calculations, CEMA and MHEA use the equivalent mass of all rotating parts, as opposed to their actual mass. The latter is used by Goodyear and DIN. Using an equivalent mass gives a more accurate value for the forces required to accelerate the conveyor during start-up. The equivalent mass is defined as:

Equivalent mass = $WK^2\ (2\pi \cdot RPM/v)^2$

Where: W is the actual mass of the rotating part

K is the radius of gyration of rotating part

RPM is the rotational speed in revolutions per minute

V is the belt velocity

3.5 Belt Sag

Belt sag depends on belt tension, idler spacing and unit weight of the belt and material. As the sag increases, the power required to drive the belt increases. In addition, a larger horizontal force is induced on the idlers, causing increased wear rates. All four standards calculate belt sag in the same manner. The difference lies in the sag values that are considered acceptable during conveyor operations. CEMA recommends three percent sag, Goodyear and MHEA recommend two percent, while DIN recommends one percent.

The experience gained with large-capacity, high-speed mining conveyors indicates that, in the interest of overall efficiency of conveying, the sag should be limited to between one and one and one-half percent.

3.6 Vertical Curves

When designing a concave curve, the radius of the curve must be large enough to prevent belt lift, belt center overstressing, and edge buckling. The design of a convex curve must prevent center belt buckling and edge overstressing.

The shape of the curve is assumed to be an arc by CEMA, MHEA and DIN. Goodyear assumes the shape to be a parabola which results in slightly larger values for curve radii than the other methods. Not enough data is available to conclude the relative correctness of these approaches.

3.7 Method comparison

A sample conveyor (see Figure 3.5) is used to demonstrate the comparison of effective tension values derived from the four discussed design methods.

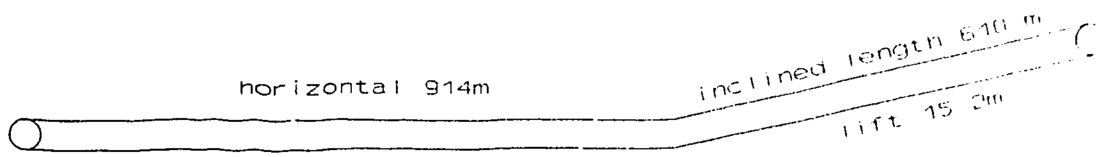

Figure 3.5 Sample conveyor

Belt Specifications

Length	1524 m	(5000 ft)
Belt width	1524 mm	(60 inches)
Belt mass	49.1 kg/m	(33 lb/ft)
Belt speed	5.0 m/sec	(984 ft/min)
Vertical lift	15.2 m	(50 ft)

Trough angle	35 °	
Surcharge angle	20 °	
Material density	1600 kg/m^3	(100 lb/ft^3)
Carry idler mass	46.4 kg	(102.2 lb)
Return idler mass	41.8 kg	(92.2 lb)

Table 3.3 Comparison of effective tension values

Method	Effective tension kN (lbs)	Deviation from overall average (%)
CEMA	230.1 (51838)	-5.6
Goodyear	261.7 (58812)	7.0
DIN	257.4 (57844)	5.3
MHEA	232.7 (51294)	-6.6

The overall average of the four methods was 244.5 kN (54 947 lbs). As Table 3.3 shows, each method deviates less than 10% from the average. The two methods with the largest difference, MHEA and Goodyear are less than 13% apart.

3.8 Discussion of Method Use

In the preceeding sections the major differences between four troughed conveyor design standards were discussed, and the most accurate methods were defined. Overall, the various inequalities between the standards balance themselves out, to a degree (provided that the evaluator has the expertise required to properly assess the belt tensions using each method), with each providing similar final results that seldom differ by more than 10%. Section 3.7 shows a sample conveyor, and compares the belt tensions calculated using each method.

The CEMA standard uses a unit resistance approach, calculating separately all major tension components. As a result, this standard assures the highest accuracy when designing a belt conveyor to operate under relatively unknown conditions, or in an environment with widely-varying operating conditions.

The DIN, MHEA and Goodyear standards do not calculate each tension component separately, but rather use factors that combine the components. These methods are simpler to use, but can produce erratic results, if values of friction factors are not certain. References to existing conveyors, operating in similar conditions, are required to obtain reliable results.

4. COMPUTERIZED CALCULATIONS OF CONVEYOR BELT TENSIONS

To define the belt tensions and the belt conveyor drive power requirements, long and tedious calculations are necessary. Also, many "what if" scenarios must be analyzed to find the best possible design. Evaluation of various belt sizes, idler spacing, load configurations, or even different conveyor profiles, are all part of finding the most efficient and economical design. The time required to perform all of these tasks by hand can be excessive; therefore, developing computer programs to calculate belt tensions, and the power requirements of belt conveyors, enable exploration of alternative designs in a fraction of the time.

4.1 Existing Programs

There are many computer software packages that are available to the engineer to aid in the design of belt conveyors. The three representative packages used were developed by: Creative Engineering Company, Computer Technology Specialists Engineering Limited, and Conveyor Dynamics Incorporated.

4.1.1 Creative Engineering Company

The programs available from this company enable one to design conveyors and estimate related costs. The method of calculation is based specifically on CEMA (2nd edition), using the standard CEMA K_y and K_x factors with no variations. The program allows division of the belt into a maximum of 100 sections, with belt tensions calculated at the endpoints of a particular section. Section ends correspond to belt slope changes, points of change in the amount of material on the belt, or can be defined at any point to allow further evaluations.

The key features of this program are as follows:

Calculation of:

- maximum belt tensions for steady operation and belt accelerating
- power required to drive the belt
- optimum idler spacing; automatically finds the maximum spacing for the worst possible operating conditions
- minimum vertical concave and convex curve radii for operating and accelerating conditions
- deceleration time and the stress induced on the belt during shutdown as well as the amount of material discharged from the belt during shutdown

Analysis of:

- single or dual drive conveyors
- shaft bending and torsion moments for pulleys
- possible harmonic vibration conditions
- various load configurations on different portions of the belt

A cost estimate of the final conveyor design may be prepared after the technical evaluation. The program incorporates cost multipliers to allow for inflation and the use of different currencies.

4.1.2 Computer Technology Specialists Engineering Limited

This program is based strictly on the CEMA method. It is menu driven for easy use, and calculates the following:

- belt tensions

- power and starting torque requirements
- minimum allowable vertical curve radii
- the required counterweight (including winch take-up forces)
- the required backstop (if any)
- acceleration and deceleration times

Two major limitations of this program are that it is able to handle a maximum of only 10 sections (914 m) (3000 ft) maximum length per section, and is unable to evaluate regenerative conveyors.

The program is divided into two parts: "short convey" and a "main convey" program. The short convey part provides the user with estimates of motor power and belt tensions, in order to better choose belt size and strength, idler size and class, etc. The program selects a K_y value from the CEMA tables, depending on the average estimated belt tension of a section, load and inclination. If the total conveyor length is greater than 914 m (3000 ft), the K_y value chosen is 0.016, and if the total mass on the belt is greater than 450 kg/m (300 pounds/foot), the K_y value chosen is 0.018. At this point, the short convey program calculates the required horsepower, motor horsepower, starting horsepower and average horsepower.

The main convey part calculates the final horsepowers, belt tensions, vertical curve radii and maximum motor acceleration. It follows through the standard CEMA progression of finding K_y values. A first estimate of K_y is selected, and the average tension over a particular section is calculated using this K_y value. From the average tension calculated, the program automatically selects a second K_y value. The process is repeated if it is not within 0.01 of the first estimate.

The program calculates values for T_1, T_2 and tail tensions, and then automatically adjusts these values to ensure adequate tension to prevent belt slippage at pulleys, and also ensures that the sag limit is not exceeded.

4.1.3 Conveyor Dynamics Incorporated / Wright Engineers Limited

This is the most extensive and advanced program presently available for belt conveyor design. It calculates the usual power requirements and belt tensions. In addition, it allows analysis of the dynamic stresses induced upon the belt during acceleration and deceleration.

This program is also based on CEMA, and allows the design of conveyors with up to four separate drive pulleys. Similar to the previous programs, this program divides the belt into sections, calculating belt tensions at the end points of each section. From this, calculations are made of the required power to drive the belt, and the vertical curve radii.

The unique feature offered by this program is the dynamic stress analysis performed on the belt for its acceleration and deceleration. During this period, stress waves are created in the belt by the change in belt velocity. The waves usually start at the drive pulley, and propogate through the entire belt. This dynamic stress is added to the static stress, and the combined stresses can over-stress the belt, cause pulley-shell and bearing failure, or pulley-support structure damage [8]. The stress wave travels through the belt and decreases in magnitude with each cycle.

Conveyor Dynamics Inc. uses the theories of wave propagation to predict the change in belt velocity and the stress induced in the belt due to these stress waves. Case studies [14] indicate that the basic theories surrounding the effect of stress waves are correct, however, they are yet to be confirmed by accurate quantitative data.

No further discussion of dynamic stress analysis is included, because the subject is not within the scope of this thesis.

4.2 Programs Developed by the Author

Each of the previous programs were developed to perform calculations on belt conveyors that operate in "normal" conditions. The need arose to develop similar programs that were applicable to design of conveyors that operate in extremely adverse conditions, specifically the conveyors that operate in the oil sand mines of northern Alberta. The described program has proven to be a useful tool for calculating belt tensions. However, it never was intended to be a 'black box'. User input is required to define the location of stations, and select various coefficients. To effectively use this program, the user must have a good knowledge of belt conveyors, and a thorough understanding of how the program works. The program, written in Fortran 77, uses a modified Conveyor Equipment Manufacturer's Association (CEMA) method to calculate the effective tension and the maximum tension along the belt. The belt is divided into sections by the user defining the desired belt layout. The program then calculates the tensions at each station, and visually displays tensions along both the carry and return belts.

Appendix A shows a sample conveyor with data files and full output, while Appendix B provides documented source code.

4.2.1 Belt tension calculations

4.2.1.1 Effective tension, T_e

The effective tension, T_e, is the minimum tension required to overcome conveyor friction and material resistance forces. The program uses a modified CEMA approach for calculation of both the effective tension and the maximum tensions along the belt. Unit resistances include idler friction (carry and return), belt flexure (carry and return), material flexure, gradient resistance, and all accessory resistances such as material acceleration, belt flexure around pulleys around pulleys, scrapers, plows and skirtboards (see Figure 4.1).

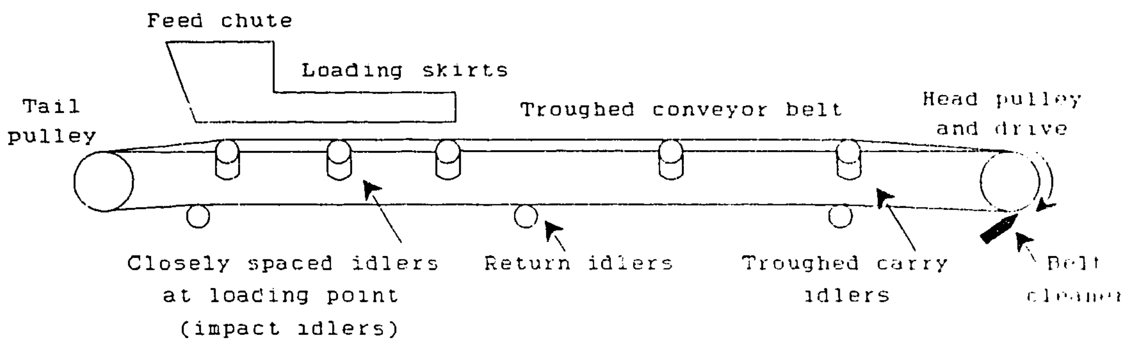

Figure 4.1 Major components of conveyor and material resistance to movement

Calculations of idler friction, belt flexure and material flexure are dependent upon belt length and unit mass, material unit mass and the unit mass of the idler rotating parts. An ambient temperature correction factor is used to compensate for changing belt and idler resistances at low temperatures. The general formulae for these resistances are:

$$T_{xc} = L_c * K_{xc} * K_t$$

$$T_{xr} = L_r * K_{xr} * K_t$$

$$T_{yc} = L_c * K_{yc} * W_b * K_t$$

$$T_{yr} = L_r * K_{yr} * W_b * K_t$$

$$T_{ym} = L_c * K_{yc} * W_m$$

Where:

T_{xc} - frictional resistance of carry idlers in N (lb)

T_{xr} - frictional resistance of return idlers in N (lb)

T_{yc} - carry belt flexure resistance in N (lb)

T_{yr} - return belt flexure resistance in N (lb)

T_{ym} - material flexure resistance in N (lb)

L_c - carry belt length in m (ft)

L_r - return belt length in m (ft)

K_{xc} - factor for carry idler rotation resistance in N/m (lb/ft)

K_{xr} - factor for return idler rotation resistance in N/m (lb/ft)

K_{yc} - factor for material and carry belt flexure resistance

K_{yr} - factor for return belt flexure resistance

K_t - temperature correction factor

W_b - belt unit weight in kg/m (lb/ft)

W_m - material unit weight in kg/m (lb/ft)

The gradient resistance and accessory resistances are defined as follows:

$T_m = H * W_m * 9.807$

$T_{am} = Q * (V - Vo)$

T_{pl} = 0.525 to 0.876 N/mm (3 to 5 lb/inch) of plow width

T_{bc} = 0.525 N/mm (3 lb/inch) of cleaner width

$T_{sb} = (L_b *[C_s * H_s^2 +6]) * 4.45$ (multiply by 4.45 to convert to N)

Where:

T_m - force required to lift material in N (lb)

T_{am} - force required to accelerate material in N (lb)

T_{pl} - resistance of plows in N (lb)

T_{bc} - resistance of belt cleaners in N (lb)

T_{sb} - resistance of skirtboards in N (lb)

H - distance material is lifted or lowered in m (ft)

Q - capacity in kg/sec (stph)

V - belt velocity in m/s (fpm)

V_o - material feed velocity m/s (fpm) t - time to accelerate material (sec)

L_b - skirtboard length (ft)

C_s - skirtboard friction factor

H_s - height of material resting against skirts (in.)

4.2.1.2 Conveyor drive power

Adding all of the resistances applicable over each section of the belt defines the effective tension. Effective tension is used to establish the conveyor drive power as:

$kW = (T_e * V) / 1000$

Where:

T_e - effective tension at the drive pulley in N

V - belt velocity in m/sec

Because the drive efficiency must be accounted for, the actual power requirement is larger. Therefore, the actual power is defined as:

$kW_{act} = kW_{theor}$ / drive efficiency

4.2.1.3 Maximum tension, T_{max}

The force required to lift and lower the belt must be considered when calculating the maximum belt tensions of inclined conveyors. In addition to all the resistances previously described, the extra belt tension from lifting the belt must be added to arrive at the maximum tension. The maximum belt tension does not affect the drive power requirements, but does affect the required belt strength.

4.2.1.4 Modifications to CEMA

Two modifications were made to the standard CEMA tension formulae.

The K_y factor for the carry side is defined differently, and the belt flexure around pulleys is included. The K_y factor is divided into three components; the imprint resistance of idlers into the belt, belt flexure, and material flexure [1]. This allows for more accurate calculations. Each section on the carry side has its own K_y value representative of the load and belt tension conditions of that section. CEMA recommends that for more detailed tension calculations, the belt flexure around pulleys should be included. This is done by taking the belt tension, width and thickness, and pulley diameter values into account as defined in the CEMA handbook 3rd edition. CEMA, however, does not include a temperature factor to compensate for the increased resistance of belt flexure due to cold belting. The program incorporates the temperature factor into the CEMA formula, taking cold weather into account. Tests conducted by the author confirm the correctness of this approach (described in Chapter 5).

4.2.1.5 Program

The program is designed to define the drive power requirements, as well as the belt tension at user defined points along the belt. The latter permits the definition of the maximum belt tension, independent of its location.

Separate calculations are done for each conveyor section, (i.e., the parts of the conveyor between the user-defined stations). The change in tension between stations must be uniform; therefore, stations define slope changes, load changes, or points of additional tension, such as belt cleaners or material acceleration. The effective tension is calculated by summing all resistances that are added or lost between any two subsequent stations.

4.2.1.6 Input

All input into the data files may be in either metric or standard (US) units. Two data files must be supplied by the user. The first file contains general information common to the entire belt:

- belt length (m or ft)
- belt width (mm or in.)
- belt unit mass (kg/m or lb/ft)
- belt velocity (m/s or fpm)
- belt cover thickness (mm or in.)
- allowable sag (percent)
- belt breaking strength (N or lb)
- elevation change (m or ft)
- idler spacing (m or ft)
- idler mass (kg or lb)
- idler diameter (mm or in.)
- idler class (CEMA p. 61, 3rd ed.)
- troughing angle (degrees)
- material density (kg/m^3 or lb/ft^3)
- surcharge angle (degrees)
- cross section derating fraction (0.0 - 1.0)
- number or drive units
- wrap angles (degrees)
- friction coefficients

The second data file defines:

- tail pulley location
- head pulley location
- drive pulley locations
- coordinates of each station

- unit weight of material on the sections expressed as a percent of the full theoretical load
- identification of the section as either a belt or accessory section

No data file is required for the information on conveyor accessories. The data pertaining to the accessories is entered at the program prompt.

4.2.1.7 Tension calculations

An initial estimate of the effective tension is made by calculating its value without dividing the conveyor into sections. This value is then used to estimate the slack side tension, T_2. It is the larger of:

1. The effective tension multiplied by the wrap factor (refer to Section 3.3)
2. The minimum required tail tension less all resistances to the last drive pulley

T_2 can either be calculated in this manner and used as the starting point for maximum tension calculations, or T_2 can be set manually, and be maintained at a constant value. At this stage, tensions at each station along the belt are found by adding the tension at the preceeding station and the resistances over the section under consideration. The calculation progresses around the entire belt, defining the tension at each station, until the final drive is encountered again. In the same manner, a more accurate value of the effective tension is calculated by adding all resistances encountered along the length of the belt.

Upon the completion of the first loop of calculations, a more accurate value of T_2 is derived, and the program repeats this calculation process twice more, for a total of three iterations. The values obtained from this third run are proven to be accurate estimates of the tension at each station. The procedure of using four iterations or more was evaluated, but it was found that the variation between tensions calculated at the stations for the third and fourth iterations varied by less than one percent. Therefore, it was deemed unnecessary to utilize more than three iterations.

The ability exists to vary the amount of material on the belt from section to section. The K_x and K_y factors are adjusted correspondingly to reflect the loads defined by them. With this feature, different load configurations such as regenerative conveyors, conveyors loaded on uphill sections only, or empty conveyors, can be studied to determine their effects on belt tensions. The vertical curves may be designed for using the worst-load configuration.

An ideal tensions distribution is assumed when the belt conveyor has dual drives. For example:

$$\frac{T_p}{T_s} = \frac{1 + CW_2}{CW_1}$$

Where:

T_p - primary effective tension in N (lb)

T_s - secondary effective tension in N (lb)

CW_2 - secondary drive wrap factor

CW_1 - primary drive wrap factor

T_3 is the tension between the primary and secondary drives. T_2 is defined as previously mentioned, while $T_3 = T_1 - T_p$ plus the force required to bend the belt around the drive pulley (see Figure 4.2).

After belt tensions at each station are calculated, the results are plotted to permit easy identification of high-tension or low-tension points. It also enables visual interpretation of tension distribution.

The first point of the X - axis corresponds to the tail pulley, and works

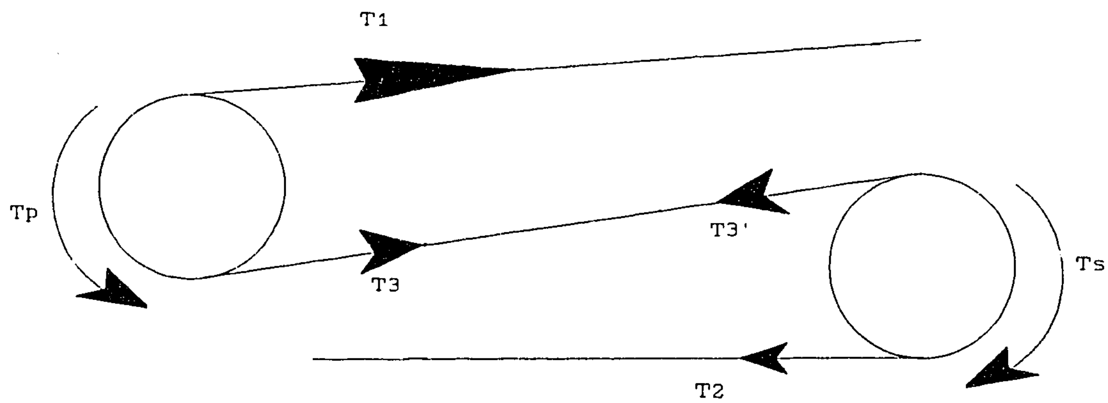

Figure 4.2 Tensions around dual drive pulleys

around the belt. The total tension is presented as the sum of:

1. Resistances from all rotating parts and accessories

2. Material resistances

The software package, PLOT88 [17], is used to plot the data points. Viewing on the video monitor is possible, as well as printing on most printers and plotters. In addition, the plot size may be varied to fit the needs of a particular application.

4.2.1.8 Acceleration forces / Vertical curves

When the belt is accelerated, extra forces are induced in the belt. Evaluation of these forces enables the maximum belt tension to be determined, thus allowing the selection of the proper belt, as well as the definition of the smallest concave vertical curve to ensure that belt lifting does not occur during start-up. Once defined, the acceleration forces can be added to the static forces already existing in the belt to allow for proper belt selection, and reliable vertical curve radii assessment.

Minimum allowable acceleration time must be calculated in order to define the acceleration forces. The accelerating time is limited by the lower of the maximum belt strength, or the available motor torque. Both time limitations must be analyzed, and the larger time of the two sets the minimum allowable accleration time.

The CEMA recommendations are followed to calculate acceleration forces with the program executing the following sequence to calculate the acceleration time limited by belt strength:

1. Find the total equivalent mass of all of the components that must be accelerated (i.e., pulleys (excluding drive pulleys), carry and return belts, carry and return idlers and material)
2. Define the maximum allowable belt tension during start-up:

$T_{avail} = T_{max} - T_e - T_2$

where:

T_{avail} is the tension available for acceleration (N)

T_{max} is the maximum allowable belt tension during start-up (N)

T_e is the required effective tension for belt operation (N)

T_2 is the slack side tension (N)

The minimum allowable acceleration time is:

$$t = \frac{system.equivalent.mass}{T_{avail} * \Delta V}$$

where:

t is in sec

system equivalent mass is in kg (refer to Section 3.4 for definition)

T_{avail} is in N

change in velocity in m/s

A similar procedure is used for definition of the minimum allowable acceleration time as limited by the motor torque:

1. Define the maximum starting torque and calculate the corresponding effective tension produced by this torque

2. Calculate the tension available for acceleration:

$$T_{avail} = T_{e\ avail} - T_{e\ req}$$

where:

T_{avail} is the tension available for acceleration

$T_{e\ avail}$ is the effective tension available for belt acceleration

$T_{e\ req}$ is the effective tension required for belt motion

3. Calculate the acceleration time limited by the motor torque

$$t=\frac{\Delta V*(system.equivalent.mass+drive.equivalent.mass)}{T_{avail}}$$

where:

V is in m/s

equivalent mass is in kg

T_{avail} is in N

The miminum acceleration time is the greater of the two:

1. If the maximum belt strength is the limiting factor, then the extra tension induced on the belt during start-up is:

$$T_{max} - T_e - T_2$$

2. If the maximum belt strength is the limiting factor, then the extra tension induced on the belt during start-up is:

$$T_{e\ avail} - T_{e\ req}$$

Following definition of the extra belt tension created due to belt acceleration, the program calculates the minimum allowable vertical concave curve radii to prevent belt lifting during start-up. The extra t. .ısion is added to the tension already existing at the point of analysis to arrive at the total belt tension at that point. The curve radius is calculated using the standard CEMA formulas for both an empty and full belt.

4.2.1.9 Deceleration

The deceleration time of a given belt conveyor is calculated for the full retarding force of the brakes. The kinetic energy of the moving conveyor must by defined to calculate the stopping time. This is done by using the equivalent mass of the entire conveyor system as shown:

$$KE=(Systemequivalentmass)*\frac{(\Delta V^2)}{2}$$

The retarding force can be defined:

$$R=kW*Efficiency*\frac{1000}{V}$$

Now, the program calculates the deceleration time of the conveyor as follows:

$$t=\frac{KE*2*R}{V}$$

From the deceleration time, one can calculate the volume and weight of material that will be discharged from the belt during deceleration.

4.2.1.10 Output

The program produces an output file divided into five sections.

1. Tabulated data related to the carry side of the belt
2. Shows the tension at each station, as well as a breakdown of individual components that contribute to the tension at that particular station. In addition, the program displays the effective, tight-side, slack-side, and minimum-tail tensions.
3. Echoing of the input data to the output file

4. Acceleration/vertical curve analysis

5. Deceleration calculations

In addition to the numerical output, a plot of the tensions along the belt length is provided as an optional feature. Appendix A shows a sample belt conveyor with data files, as well as typical output.

4.2.2 Comparison of theoretical and actual power requirements

The relative accuracy of the theoretical power requirements obtained using the computer programs presented in this thesis were determined by comparing the results with those calculated using each of the four design methods described in Chapter 3, and with the actual power requirements obtained from measurements performed on an existing conveyor installation.

The sample conveyor is defined by the following parameters:

Length:	1151 m
Vertical lift:	17.8 m
Material mass:	225.9 kg/m (79% of full theoretical load)
Belt mass:	81.8 kg/m
Belt width:	1524 mm
Belt speed:	5.4 m/sec
Carry idler spacing:	1.22 m
Return idler spacing:	3.05 m
Ambient temperature:	-34°C
Motor efficiency:	95%

In order to eliminate the effect that the oil sand stickiness (refer to Section 5.2) has on the power required to drive a belt conveyor, the results used for the power requirements of the actual conveyor were taken from a test performed at -34°C. Each of the five theoretical values was then calculated for a temperature of -34°C. In addition, the actual conveyor had an average load estimated at 79% of its full theoretical load. This estimate was based on the material height measured along the entire length of the running conveyor, and a corresponding surcharge angle of 10°. Therefore, the load used in calculating the theoretical power requirements was also 79% of the full theoretical load, or 226.9 kg/m.

As illustrated in Table 4.1, the power requirements calculated using the programs in this thesis estimated the actual power requirements more closely than the other four design methods. This this close estimate is a result of the utilization of the K_y friction factors based on the field studies by Behrends et al [1], which are more representative of the loads and belt tensions of large conveyors, and the use of the unit resistance approach.

Table 4.1 Comparison of theoretical and actual power requirements

Design method	Effective tension (kN)	Power (kW)
Thesis programs	245	1393
CEMA	210	1194
DIN	231	1314
Goodyear	199	1134
MHEA	206	1170
Actual conveyor	257	1463

5. EFFECT OF TEMPERATURE ON OIL SAND CONVEYING

The ambient temperature that a belt conveyor functions in affects the power that is required to operate it. In all belt conveyor installations, both the idler rotational resistance, and the belt flexure resistance increase with decreasing temperature. In addition to these effects, the oil sand stickiness and density vary with temperature, making the effect of temperature on oil sand conveying unique.

As discussed in Chapters 3 and 4, CEMA uses three factors to account for the major resistances; K_x, K_y, and K_t. The K_x factor represents the resistance of the idlers to rotation, as well as the sliding resistance between the belt and idler rollers. The force required to rotate the idlers arises from bearing and grease resistance. The bearing resistance is a function of the load on the idlers, and does not vary much between installations because most idlers have anti-friction bearings. The resistance from the grease, however, increases with decreasing ambient temperature. The sliding resistance between the belt and idlers, caused from the belt not travelling exactly perpendicular to the idlers, will be higher in oil sand conveying than in most other types of conveying because of the sticky nature of the oil sand.

The K_y factor is used to calculate the resistance of the belt and material as they flex over the idlers, as well as the resistance of the idler imprint into the belt. It is a function of the belt tension, load, and idler spacing. In addition to these three components, a fourth can be added for oil sand conveying. Due to the oil sands' sticky nature it is not possible to keep the belt and idlers free of oil sand. The result is a build-up on the belt and idlers. This increases both the force required to flex the belt and the imprint resistance between the belt and idlers, as well as the force required to overcome the belt and idlers glueing together.

The temperature correction factor, K_t, is used in conjunction with the K_x and K_y factors to compensate for the effect that the ambient temperature has on the operation of the conveyor. It is a dimensionless multiplying factor that increases the calculated value of belt tensions to account for the increased resistances that can be expected due to low temperatures [3]. The CEMA recommended values for K_t are based on typical belts, grease, and conveyor installations. Therefore, the CEMA values are not suitable for use in oil sand conveyor design.

The following sections examine the effect of temperature on belt flexure, the stickiness of oil sand, and the density of oil sand.

5.1 Belt Flexure Tests

The rubber in the belting used for oil sand conveying has a different chemical composition than most other types of belting. Due to the cold weather, and the peculiar properties of the oil sand, belting has had to be specifically developed for use in oil sand conveying. With the oil sand belt being constructed of different rubber than most conventional belts, the forces required to bend and flex the belt change. As a result, tests were conducted to quantify the forces required to flex this type of belting over a range of temperatures, calculate modified K_t values, and relate these findings to the CEMA recommended K_t values. Two methods were used to estimate the actual resistance to belt flexure; belt bending and belt flexure around pulleys.

5.1.1 Belt bending

5.1.1.1 Procedure

The belt bending tests depended on the free bending of a belt sample over a 180° range. One end of the belt sample was secured and the force required to bend the belt to a pre-defined position was measured (see Figure 5.1). The force required to bend the belt at room temperature was used as a base value, and the forces required to bend the belt at lower temperatures were compared to this base value. The ratio of the force required at lower temperatures to the force required at room temperature was considered to be the modified K_t value (as applied to the belt flexure resistance only).

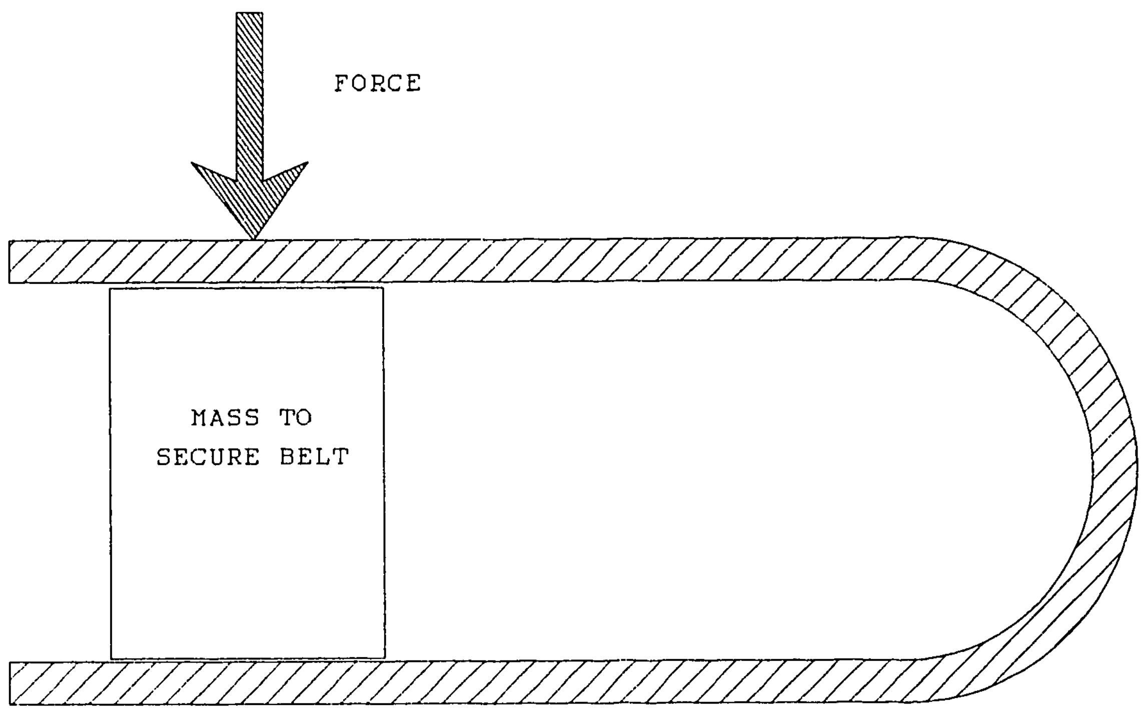

Figure 5.1 Sketch of belt bending test

The belt bending tests were carried out on four belt samples, each taken from an ST4000 belt. Two samples were approximately 50-mm wide and included three cables, while the two other samples were approximately 75-mm wide and included five cables (see Figure 5.2).

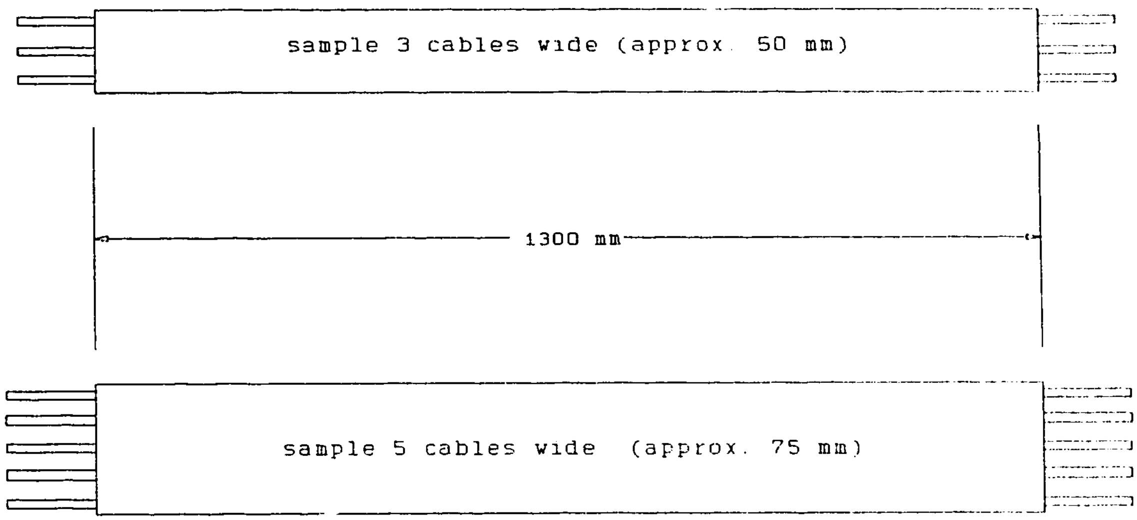

Figure 5.2 Sketch of belt samples

5.1.1.2 Results

Forty measurements were taken from each belt sample at temperatures of +5°C to -20°C at 5°C intervals. Table 5.1 shows the calculated K_t values for each of the belt samples.

The overall averages for the measured K_t values differ significantly from the recommended CEMA values, as illustrated in Table 5.1. The change in the force required to bend the oil sand belt can be readily detected from +22°C to +5°C (20% increase), whereas a temperature of approximately -15°C must be present before the same 20% change in force is recognized by CEMA. However, as the temperatures decrease further, the measured K_t factors appear to approach those recommended by CEMA.

Similar results for the effect of temperature on the turning resistance of large conveyor idlers were reported by Golosinski and

Cholewa [6]. If the turning resistances of the idlers are translated into K_t values, they would be approximately 1.24, 1.45, 1.61, and 1.83 for the temperatures of 5 °C, -10 °C, -20 °C, and -30 °C, respectively. The first three K_t values for the turning resistance of the idlers compare very closely with the K_t values measured for the belt bending. Assuming that this trend continues, and knowing that the value of 1.83 for the turning

Table 5.1 Comparison of measured and CEMA K_t values

Temperature range (°C)	Measure K_t overall average values	CEMA recommended values
22	1.0	1.0
5 to 1	1.20	1.0
0 to -4	1.30	1.05
-5 to -9	1.38	1.10
-10 to -14	1.49	1.15
-15 to -20	1.57	1.30
-21 to -25	1.70 (estimate)	1.50
-26 to -30	not measured	1.80
-31 to -35	not measured	2.20
-36 to -40	not measured	2.70
< -40	not measured	3.0

resistance of the idler roller at -30 °C is approximately equal to the CEMA recommended value at the same temperature. Therefore, one can assume the convergence of the measured K_t values and the CEMA K_t values. Based on these results, it may be concluded that the belting used in oil sand mining is affected more by lower temperatures than most other types of belting. As a result, when calculating the power requirements and belt tensions of oil sand conveyors, the K_t values in Table 5.1 should be used as follows: Utilize the measured values up to -25 °C, and the CEMA recommended values for temperatures of -26 °C and lower (see Figure 5.3).

Figure 5.3 graphically illustrates the difference between the power requried to drive a belt conveyor using the CEMA recommended K_t values, the measured K_t values, and the actual power required to drive the belt conveyor. As the graph shows, the power calculated using the measured K_t values is much closer to a "real life" situation, confirming that the measured values are better to use in design of an oil sand conveyor, while the CEMA recommended values underestimate the required belt bending forces (the discrepancy between the actual and theoretical powers may be attributed to the oil sand's "stickiness", and is explained further in Section 5.2).

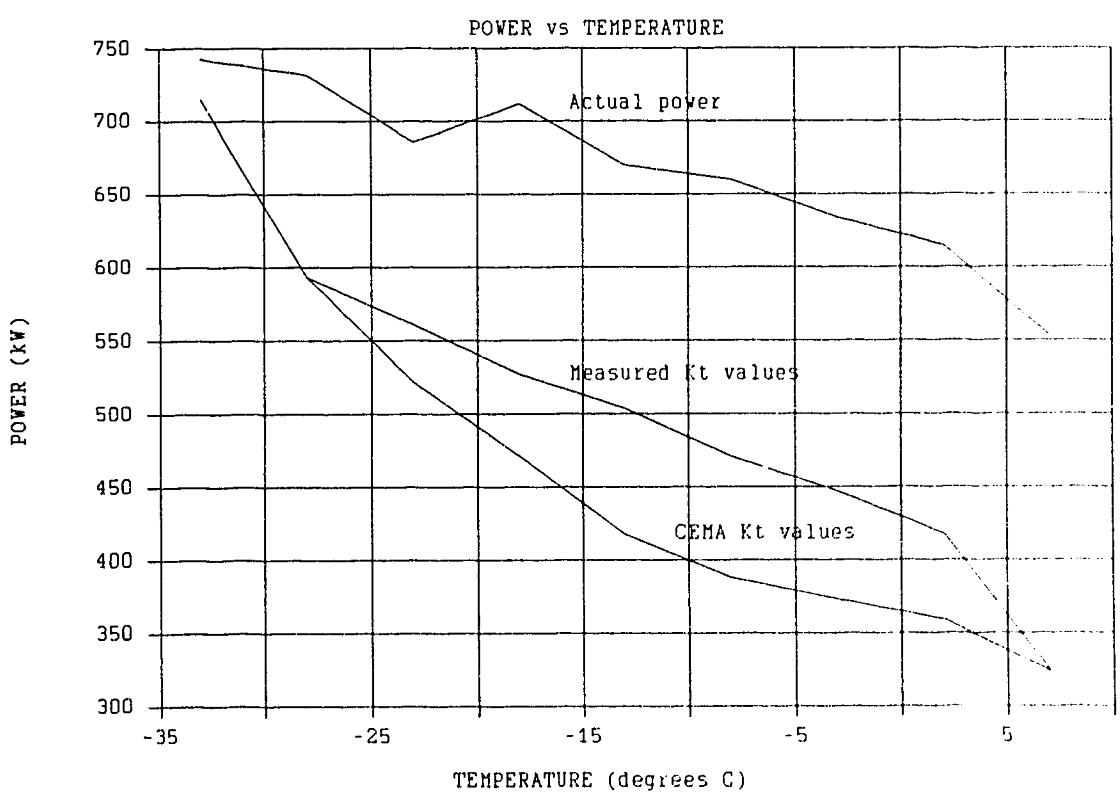

Figure 5.3 Plot of theoretical and actual power requirements

5.1.2 Belt flexure around pulleys

The force required to flex belting around a pulley is a function of the pulley diameter and the belt stiffness. The smaller the pulley diameter and stiffer the belt, the greater the force required to flex the belt around the pulley.

Although the force required to flex belting around a pulley is small in comparison with the total force required to drive the belt, defining it accurately may enhance the overall accuracy and reliability of belt conveyor tension estimates. Therefore, tests were carried out to confirm that the CEMA recommended values (at room temperature) applied to belting used for oil sand conveyors. Investigations for freezing temperatures were beyond the scope of this work.

5.1.2.1 Procedure

The apparatus used to measure the belt flexure resistance during bending around pulleys is shown in Figure 5.4. A wooden pulley was mounted on a shaft, and this, together with the bearing assembly, was attached to a stand. To measure the belt flexure resistance, a 75-mm wide belt sample (similar to those shown in Figure 5.2) was wrapped around the pulley. A 392-N weight was attached to one end of the belt to provide tension sufficient to keep the belt in constant 180° contact with the pulley. A strain gauge, with a resolution of 0.1-N, was attached to the other end of the belt to measure the forces required to flex the belt and overcome any minor resistances such as bearing resistances.

The apparatus was set-up with a 392-N weight attached to one end of the belt with the weight resting on the ground. The other end of the belt was slowly pulled approximately 20 cm downwards, and then let back up. This procedure was repeated five to eight times with the strain gauge recording the force required to pull the belt down each time.

The force required to flex the belt at room temperature was used as a base, and all of the recordings at temperatures lower than this were compared to the base force to detect any increase in force.

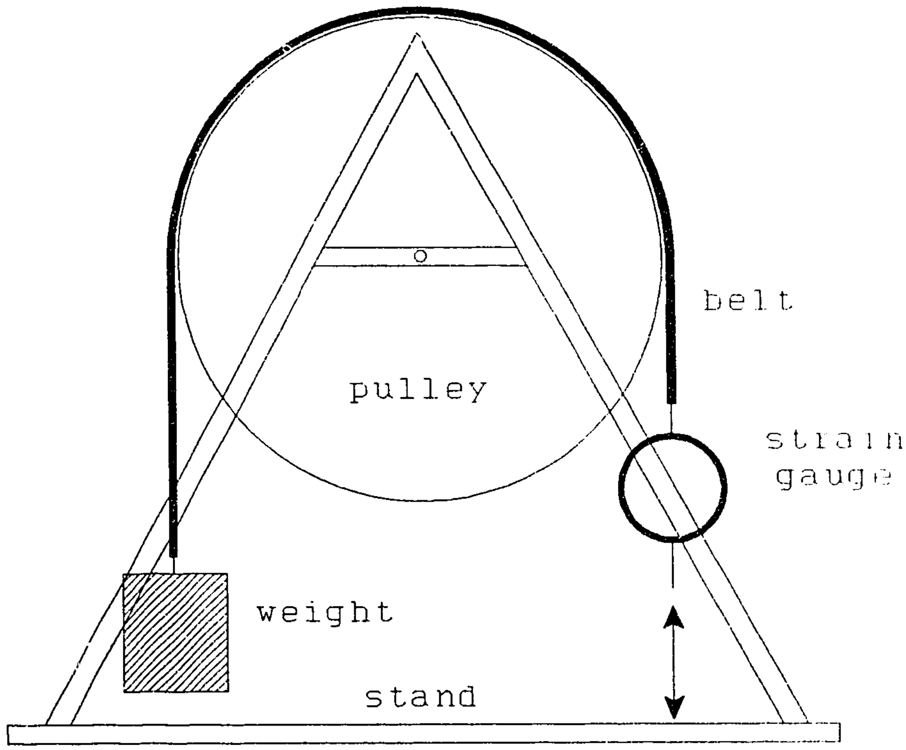

Figure 5.4 Belt flexure apparatus

5.1.2.2 Calculation of flexure resistance

When using this technique to define the belt flexure resistance of belting around pulleys, resistances other than the belt flexure resistance must be accounted for, or eliminated, to islolate the belt flexure resistance. These resistances are the bearing resistance, and the changing resistance caused by the changing mass of belt on each side of the rotating pulley. The bearing resistance can be defined by measuring the force required to rotate only the pulley. The belt mass, however, must be eliminated from any equations used to calculate the belt flexure resistance.

The following equations show how the belt mass is eliminated from the equations used to calculate the belt flexure resistance (Figure 5.5 illustrates the forces on a diagram).

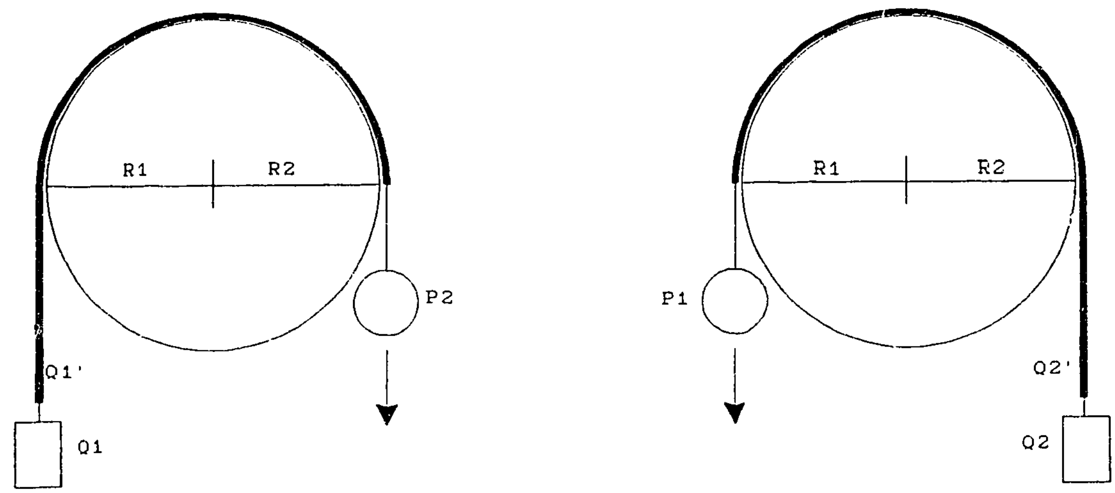

Figure 5.5 Sketch of forces around pulley

$$-(Q_1 + Q_1')R_1 + (P_2 + Q_2')R_2 - M_o - M_B = 0 \qquad (1)$$

$$-(P_1 + Q_1')R_1 + (Q_2 + Q_2')R_2 - M_o - M_B = 0 \qquad (2)$$

Adding (1) + (2)

$$-(Q_1 + P_1 + 2Q_1')R_1 + (Q_2 + P_2 + 2Q_2')R_2 = 0 \qquad (3)$$

Rearranging (3)

$$Q_1'R_1 + Q_2'R_2 = (Q_1 + P_1)R_1/2 - (Q_2 + P_2)R_2/2 \qquad (4)$$

$$-(Q_1 + Q_1')R_1 + (Q_2 + Q_2' + B)R_2 - M_o - M_B = 0 \qquad (5)$$

Rearranging (5)

$$M_B = (Q_2 + B)R_2 - Q_1R_1 - M_o + (Q_2'R_2 - Q_1'R_1) \qquad (6)$$

Substituting (4) into (6)

$$M_B = (Q_2 + B)R_2 - Q_1R_1 - M_o + (Q_1 + P_1)R_1/2 - (Q_2 + P_2)R_2/2 \qquad (7)$$

Where:

P_1 - total weight required to move belt to side 1 (N)

P_2 - total weight required to move belt to side 2 (N)

Q_1 - weight attached to side 1 (N)

Q_2 - weight attached to side 2 (N)

Q_1' - weight of belt on side 1 (N)

Q_2' - weight of belt on side 2 (N)

R_1 - moment arm of side 1 (m)

R_2 - moment arm of side 2 (m)

B - measured force by strain gauge (N)

M_o - bearing resistance (N•m)

M_B - belt flexure resistance (N•m)

5.1.2.3 Results

The tests were conducted in room temperature with a pulley diameter of 0.902-m, and pulley bearing resistance of 0.1-N. The average force measured by the strain gauge over eight tests was 24.3 N, with the average belt flexure resistance being 7.4 N. Substituting into equation 5:

$$M_B = (391.5 + 24.3)(0.452) - (383.6)(0.450) - (0.1)(0.452) + (383.6 + 374.2)(0.450)/2 - (391.5 + 415.8)(0.452)/2$$

$$M_B = 3.4 \text{ N•m}$$

Converting to force:

$$M_B = 3.4 \text{ N•m} / 0.452 \text{ m}$$

$$M_B = 7.5 \text{ N}$$

This value can be compared to the value obtained by using the CEMA formula to calculate the force required to bend a belt around a pulley. The belt sample used had a thickness of 0.033 m, was 0.075 m wide, and had a tension of 392.4 N. Substituting into the CEMA formula gives a value of 8.4 N, or 12% higher.

The student 't' test can be used to determine if the measured results are adequate:

$$t = (\bar{x} - \mu_o) * \frac{\sqrt{n}}{s}$$

where: $\bar{x}$ is the sample mean

μ_o is the expected value

s is the sample standard deviation

n is the sample size

$t = (7.5 - 8.4) * \sqrt{8} / 0.9$

$t = -2.8$

At a level of significance of $\alpha = 0.005$, where t = 3.355, the null hypothesis cannot be rejected, and the measured value of 7.5 N is accepted. As a result of these investigations, it can be concluded that the CEMA recommended values for the belt flexure resistance around pulleys can be applied to oil sand conveyors.

The belt stiffness increases as the ambient temperature decreases. It follows that the belt flexure resistance should also increase in lower temperatures. To define the required force to flex a frozen belt and compare it to the room temperature results, all other test conditions must be identical. The most important factor that must remain the same is the arc of contact between the belt and pulley. As the arc angle decreases, so does the force required to flex the belt. A much larger weight is required to keep the frozen belt in 180° contact with the pulley than is needed for the room temperature belt. The apparatus used for the room temperature investigations was not capable of supporting such large weights, and, as a result, further tests defining belt flexure resistances in colder temperatures were impossible. Another study of this nature, however, may be undertaken in the future to better determine the effect of ambient temperature on the belt flexure resistance of oil sand belting.

5.2 Oil Sand Stickiness

The property of oil sand that makes it such a unique material to convey is its stickiness. Oil sand adheres to the belt, idlers and pulleys, increasing the power required to drive the belt conveyor. Figure 5.3 illustrates the difference between the theoretical power required to operate an empty belt conveyor, and the actual power required to operate that same belt conveyor. At ambient temperatures above five °C, the theoretical power for the empty belt is 325 kW. The actual recorded power for the same belt is 550 kW, or 225 kW greater. As the temperature decreases, the gap between theoretical and actual powers decreases slightly, but is still substantial (at -15 °C, theoretical power is 520 kW and actual power is 680 kW, 160 kW greater).

No accurate explanation exists for this phenomenon, but it appears that oil sand stickiness results in large power demands. The most probable explanation is that the oil sand sticks to the belt, idlers and pulleys, and causes the following:

- the belt and idler weights are substantially increased
- the oil sand acts as a glueing agent between the belt and idlers and pulleys
- the imprint of the idlers into the belt is deeper, increasing imprint resistance
- oil sand stuck on the belt acts as part of the belt, increasing belt flexure resistance

Each of these consequences are likely to occur when the oil sand sticks to the belt and idlers, thus increasing the power required to drive the belt conveyor.

As the temperature decreases, the power increases which is required to flex the belt and turn the idlers. As explained in Section 5.0, a temperature correction factor, K_t, is used to compensate for these power demands. Theoretically, the power demands should continually increase as the ambient temperature decreases, but as shown in Figure 5.3, it can be seen that the actual power requirements start to decrease when the ambient temperature is in the range from -18 °C to -23 °C. Once the temperature reaches about -25 °C, however, the power demands starts to increase again. Investigations by mine operators have shown similar results when the rolling resistance of idlers is considered. In this case, the power demand peaks in the -15 °C to -20 °C range, then decreases as the ambient temperature decreases.

The most logical explanation for this decrease in power requirements at approximately -20 °C appears to be that this is the temperature at which the oil sand becomes brittle and falls off of the belting, idlers and pulleys. As a result, the adhesive nature of the oil sand is completely lost, resulting in a decreased power demand. Referring to Figure 5.3, however, the difference between the theoretical and actual power requirements at -23 °C is still approximately 130 kW. This difference may be explained by the fact that, when frozen to the belt, the oil sand acts as part of the belt. Its stickiness may be lost, but the extra oil sand weight is added to the belt, and flexure of the belt is more difficult. As a result, more power is required to drive the belt conveyor than if the belt had no oil sand permanently frozen to it.

5.3 Oil Sand Density

The density of oil sand is used in conjunction with the material cross-section and belt speed to determine the mass conveying rate of a belt conveyor. Therefore, it is imperative that an accurate value for the density of oil sand be known, and how the ambient temperature affects the density.

The amount and size of the lumps within the conveyed oil sand both increase with decreasing ambient temperature. With this phenomenon, it should follow that the density should decrease. Tests performed on the density of oil sand at various temperatures confirm this phenomenon.

For sake of simplicity, the difference in density is distinguished between frozen and non-frozen oil sand (i.e., winter density and summer density). The average winter density was found to be 1186 kg/m^3, while the summer average is 1284 kg/m^3.

A two sample t test, similar to that described in Section 6.5.2, was used to test for a significant difference between the summer and winter densities. At a 0.001 level of significance, and with 29 degrees of freedom, the tablulated t value is 2.756. The calculated t value is 8.636. Because 8.636 is greater than 2.756, the two densities are considered significantly different.

The summer density is 8.3% higher than the winter density, which would lead one to conclude that the mass conveying rate is also 8.3% higher in the summer than in the winter. This, however, is not the case, because the lumpiness of the oil sand in the winter creates a greater surcharge angle of the oil sand, thereby increasing the volumetric rate, and compensating for the lower winter density. As a result, the summer values for both the density and surcharge angle should be used for design purposes. This concept is discussed in further detail in Section 6.4.

6. OIL SAND SURCHARGE ANGLE MEASUREMENTS

The surcharge angle is the angle above the horizontal which the surface of the material describes while on a moving conveyor (see figure 6.1). It is sometimes referred to as the "dynamic" angle of repose.

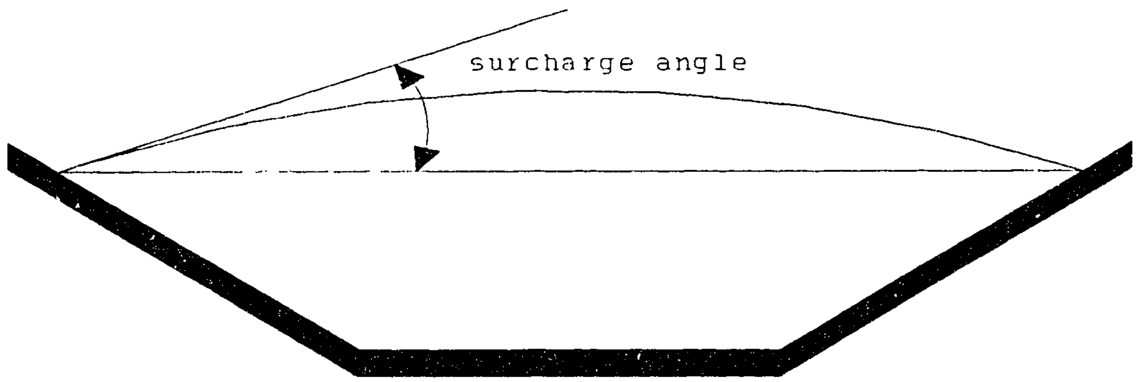

Figure 6.1 Diagram illustrating surcharge angle

The surcharge angle affects the amount of material on the belt, and thus its accurate value must be known for conveyor design purposes. For a given belt conveyor the surcharge angle, belt width, troughing angle, and belt speed define the volumetric capacity of the conveyor, and, in connection with the density of the conveyed material, it defines the drive power requirements, maximum conveyor lengths and belt tensions. For example, a 1524-mm wide belt, with a 35° troughing angle, and a 10° surcharge angle has a cross-sectional area of 0.226 sq m. The surcharge angle of oil sand has been estimated as high as 20° [12], and this same belt with a 20° surcharge angle has an area of 0.267 sq m, an increase of 18.1%. Thus, the volume of material being transported can be significantly affected by relatively small changes in the surcharge angle.

6.1 What Affects Oil Sand Surcharge Angle

The surcharge angle of oil sand has been estimated to be between 10° and 20°. As shown above, this range of the surcharge angle results in unacceptably wide differences when estimating the volumetric capacity of a belt conveyor. The surcharge angle of oil sand is affected by both the oil sand's physical properties, and the disturbances encountered while being transported on the belt. The following parameters may influence its actual value:

- oil sand grade
- ambient temperature
- oil sand lumpiness
- moisture content of the oil sand
- distance travelled on belt
- conveyor inclination
- idler spacing
- belt tension
- belt sag

Of these, the oil sand grade, the ambient temperature, and the distance the oil sand travels on the belt, are considered to be by far the most significant, and were therefore investigated in detail. The other factors were found to be of lesser importance.

6.1.1 Oil sand grade

The bitumen content of the oil sand affects its adhesive forces. It is commonly assumed that the higher the bitumen content, the higher the surcharge angle. However, when the oil sand freezes, the bitumen loses its adhesive properties, and, as a result, its content should have no effect on the surcharge angle. For practical purposes, three grade ranges of bitumen content are distinguished: below 9% is considered to have low bitument content, between 9% and 11% medium content, and greater than 11% high bitument content.

6.1.2 Ambient temperature

The ambient temperature indirectly affects the surcharge angle by altering the lump size of the oil sand. When the oil sand freezes its lumpiness increases, and, as a result, the surcharge angle increases. This change, however, does not necessarily mean that the actual amount of material being transported by the conveyor will increase, because the density of frozen oil sand is less than that of unfrozen oil sand. The frozen oil sand has many voids which decreases its overall density, and offsets the increased surcharge angle to some extent.

6.1.3 Distance travelled on belt

The conveyor belt and material is lifted and lowered as it travels over the idlers, resulting in numerous small impacts on them. Once the oil sand has been loaded onto the belt, its surcharge angle starts to decline because of these disturbances encountered while the belt is in motion. The farther the belt travels, the lower the surcharge angle becomes, until it stabilizes at a certain specific value.

6.2 Oil Sand Surcharge Angle Investigations

A protractor type device (shown in Figure 6.2) was used to evaluate the oil sand surcharge angle. The protractor portion had a 15-cm radius, large enough to provide accurate resolution when reading the angle from the perimeter. The arms were attached to the protractor with a bolt and wing nut. This allowed for easy positioning of the arms on the oil sand,

and quick tightening of the nut to secure the arms in place. The arms of the device were extended to the edge of the material and positioned

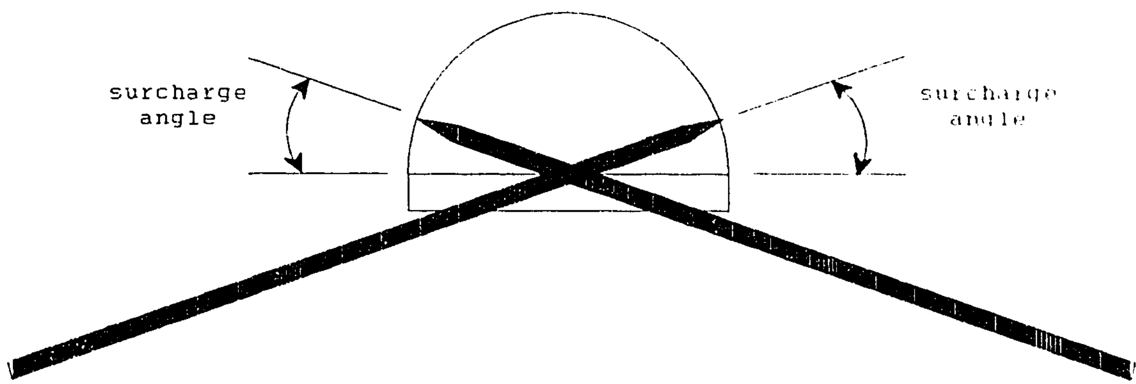

Figure 6.2 Surcharge angle measuring device

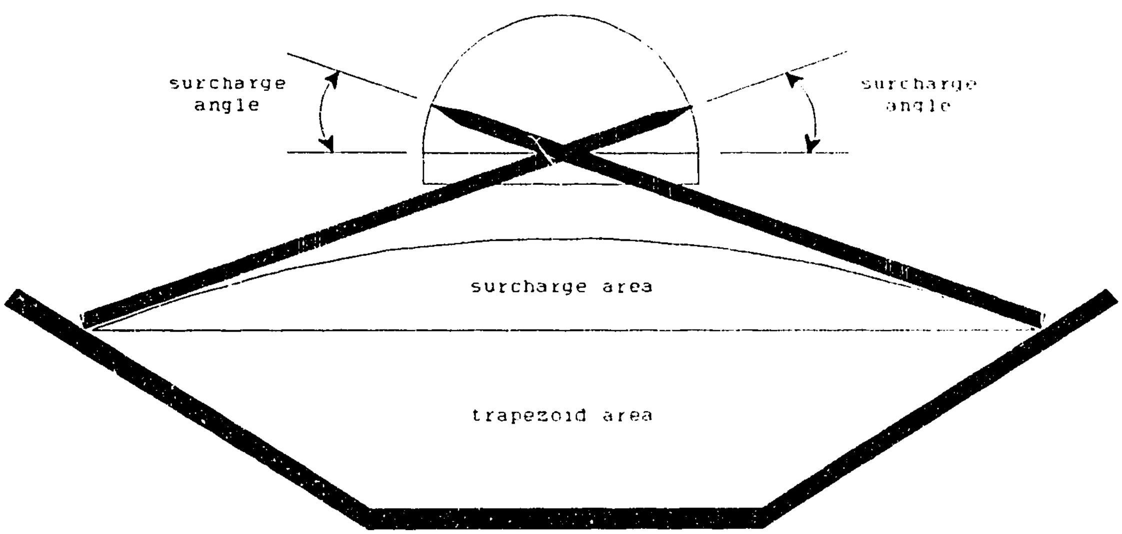

Figure 6.3 Positioning of measuring device on oil sand

tangent to it, while being perpendicular to the belt length. The surcharge angle of each side of the material was then read off of the protractor. Assuming that the surcharge angles on each side of the belt were equal, the two readings were averaged, and the result was assumed to be the actual surcharge angle. Figure 6.3 illustrates the positioning of the device while being used to measure the surcharge angles.

6.3 Field Measurements

The surcharge angles were measured on 1524-mm wide belt conveyors at Suncor. Measurements were taken when the conveyor had stopped running, and was either partially- or fully-loaded. Several measurements were taken at approximately half-meter intervals at both the head and tail stations of the conveyor each time it was stopped. The average of these readings were then calculated.

6.3.1 Summer measurements

Altogether, 225 measurements were taken during the summer. These are summarized in Table 6.1 which shows mean values of surcharge angles by location and bitumen content. The ambient temperature range during the investigation was between 10 °C and 25 °C.

From these results, it can be concluded that the distance that the oil sand travels on the belt has a very significant effect on the surcharge angle (refer to Section 6.5.3). The total change in surcharge angle from the load point to 1500 meters of travel was 1.7°, 2.0° and 2.3° for low-, medium- and high-grade oil sand, respectively.

Student t tests comparing two means were used to test the difference between the average surcharge angle for different bitumen contents. These tests, shown in Section 6.5, indicate that there is no statistical evidence to prove that the bitumen content affects the surcharge angle of oil sand (see Section 6.5).

Results from the measurements recorded at the load point allow the conclusion that 10° is a conservative estimate of the oil sand surcharge

Table 6.1 Statistics on summer surcharge angles

Bitumen content	Dist. from load point (m)	Mean surch. angle (°)	Standard deviation (°)	Sample size	Coeff. of variation (%)
Low	0	9.2	1.14	27	12.4
	1500	7.5	0.77	16	10.3
Medium	0	9.6	1.33	56	13.9
	1500	7.6	1.32	87	17.4
High	0	9.4	1.02	27	10.9
	1500	7.1	1.04	12	14.6
All	0	9.4	1.22	110	13.0
	1500	7.5	1.24	115	16.5

angle in above-freezing temperatures. The angle decreases to 7.5° as the oil sand travels on the conveyor.

6.3.2 Winter measurements

A total of 64 surcharge angle readings were taken during the winter. The effect of the bitumen content on the surcharge angle in the winter was not analyzed because bitumen freezes in the winter, and loses its stickiness, and, the summer tests showed that bitumen content has little, or no effect on surcharge angle. Table 6.2 shows the measured mean surcharge angles by location. The ambient temperature range during the winter measurements was -5°C to -25°C.

6.4 Comparison of Winter and Summer Surcharge Angle Values

The distance that the oil sand travels on the belt conveyor has a similar affect on both the frozen and unfrozen surcharge angles. This is illustrated by the fact that the change in the summer and winter surcharge

Table 6.2 Statistics on winter surcharge angles

Distance from load point (m)	Mean surcharge angle (°)	Standard deviation (°)	Sample size	Coefficient of variation (%)
0	13.8	1.32	30	9.6
1500	12.2	1.27	34	10.4

angles (from the head station to the tail station) were 1.9 and 1.6°, respectively.

It can be seen that the surcharge angles are about four degrees greater in the winter than in the summer. This appears to result from the larger size of the lumps of oil sand in the winter. However, with the surcharge area containing larger lumps in the winter, there are also larger voids. These voids reduce the density of the material contained within the belt, and subsequently reduce the mass conveying rate. The significance of the larger surcharge angle in the winter can be assessed by comparing the calculated mass of oil sand on a belt conveyor using winter data and summer data.

Studies performed at the University of Alberta show that the density of oil sand is temperature-dependent, averaging 1284 kg/m^3 in the summer and 1186 kg/m^3 in the winter. The mass per linear meter is defined by multiplying the cross-sectional area of the oil sand on the belt by the density of the oil sand. Assuming a 1524-mm wide belt, and standard CEMA edge distances, the cross-sectional areas are 0.223 m^2 in the summer and 0.241 m^2 in the winter (assuming 9.4° and 13.8° surcharge angles in the summer, and winter, respectively). From these values, the calculated linear mass of oil sand is 286.3 kg/m in the summer, and 285.8 kg/m in the winter, or less than a 0.2% difference.

This allows the conclusion that the mass conveying rates are virtually identical in both the summer and the winter, and that summer data can be used in conveyor design.

6.5 Evaluation of Results

Once the raw data on surcharge angles was collected, statistical analyses were performed to provide estimates of the mean surcharge angle, its variability and the confidence of the results. Standard statistical equations were used as follows:

$$\bar{x} = \frac{\sum_{i=1}^{n} x_i}{n}$$

Where $\bar{x}$ = sample mean

x_i = each individual measured value

n = number of measurements

The standard deviation, s, is a measure of the absolute variation within a given sample. It is defined by the formula:

$$s = \sqrt{\sum_{i=1}^{n} \frac{(x_i-\bar{x})^2}{n-1}}$$

To compare the variation between several sets of data (as between data sets of different bitumen content or position), a measure of the relative variation was used. The coefficient of variation, V, gives the standard deviation of a data set as a percentage of its mean.

$$V = \frac{s * 100}{\bar{x}}$$

If the coefficient of variation of a particular data set is less than that of another data sets, then the measurements from the first data set are considered to be more precise.

Interval estimates are intervals that, with a particular degree of certainty, contain the sample mean. They are useful, because point estimates cannot be expected to coincide with the quantities they are intended to estimate [13]. Confidence intervals for each data set were calculated from the surcharge angle measurements.

6.5.1 Surcharge angle means and confidence intervals

As a sample calculation, the summer measurement of low bitumen content at the load point with a 99% confidence interval is shown.

Sample size, $n = 27$

Standard deviation, $s = 1.14$

Sample mean, $\bar{x} = 9.2$

$t_{\alpha/2} = 2.779$ for $n - 1 = 26$ degrees of freedom

$$\bar{x} - t_{\alpha/2} * \frac{s}{\sqrt{n}} < \mu < \bar{x} + t_{\alpha/2} * \frac{s}{\sqrt{n}}$$

Substituting into the above formula:

$$9.2 - 2.779 * \frac{1.14}{\sqrt{27}} < \mu < 9.2 + 2.779 * \frac{1.14}{\sqrt{27}}$$

The confidence interval is: (8.6, 9.8)

This means that the mean surcharge angle for low bitumen content oil sand at the load point is between 8.6° and 9.8° 99% of the time. The confidence intervals for the summer surcharge angles, by bitumen content and distance from the load point, as well as all of the summer measurements, (regardless of bitumen content), and all of the winter measurements, are shown in Table 6.3.

6.5.2 Effect of bitumen content on surcharge angles

Another check was made to determine if the bitumen content has any

Table 6.3 Confidence intervals for surcharge angles

Bitumen content	Distance from load point (m)	Confidence interval (99%) (°)
Low	0	(8.6,9.8)
	1500	(6.9,8.1)
Medium	0	(9.1,10.1)
	1500	(7.2,8.0)
High	0	(8.9,10.0)
	1500	(6.2,8.0)
All	0	(9.1,9.7)
	1500	(7.2,7.8)
Winter	0	(13.2,14.4)
	1500	(11.6,12.8)

effect on the surcharge angle. This was done by performing a two-sample t test comparing each pair of mean surcharge angles. However, the two-sample t test cannot be used when there is a substantial difference between the two respective population variances. Therefore, where the variates of the surcharge angle populations appeared to be quite different, they were tested for equality using the F statistic. The two-sample t test was performed if the variances were considered equal by the F test. The low-and medium-grade oil sand at the load point was analyzed as an example.

Low bitumen content: sample size = 27

sample variance = 1.30

Medium bitumen content: sample size = 56

sample variance = 1.77

Solution:

1. Null hypothesis: $\sigma_1^2 = \sigma_2^2$

   Alternative hypothesis: $\sigma_1^2 \neq = \sigma_2^2$

2. Level of significance: $\alpha = 0.02$

3. Criterion: reject null hypothesis if $F > F_{0.01}$ with 26 and 55 degrees of freedom, or $F > 2.03$.

4. $F = 1.33/1.14 = 1.17$

5. Decision: since $F = 1.17$ does not exceed 2.03, the null hypothesis cannot be rejected, and the variances can be considered equal.

Table 6.4 shows the comparisons of the F values for each of the six possible pairings. From the table it can be seen that the variances from each pairing are considered to be equal.

Table 6.4 Comparison of F statistics

Distance from load point (m)	Pairing (bitumen content)	F value (calculated)	F value (from tables)
0	Low-Medium	1.17	2.35
	Low-High	1.25	2.56
	Medium-High	1.70	2.35
1500	Low-Medium	2.94	3.53
	Low-High	1.82	3.74

With proof that no significant difference exists between each sample pair, the two-sample t test was performed. The formula used to estimate the value for t is:

$$t = \frac{\overline{X_1} - \overline{X_2}}{\sqrt{(n_1 - 1) * s_1^2 + (n_2 - 1) * s_2^2}} * \sqrt{\frac{n_1 * n_2 \, (n_1 + n_2 - 2)}{n_1 + n_2}}$$

where $\bar{x}_i$ are the sample means

n_i are the sample sizes

s_i^2 are the sample variances

Once again a sample calculation for the low - medium bitumen content pairing at the load point is used as an example:

1. Null hypothesis: $\mu_1 - \mu_2 = 0$

   Alternative hypothesis: $\mu_1 - \mu_2 \neq 0$

2. Level of significance: $\alpha = 0.01$

3. Criterion: reject null hypothesis if t is less than -2.576 or if t is greater than 2.576, where 2.576 is the value of $t_{0.005}$ for an infinite number of degrees of freedom.

4. Calculations: $\bar{x} = 9.2$ $\bar{x} = 9.6$

   $s_1^2 = 1.31$ $s_2^2 = 1.76$

   $t = -1.343$

5. Decision: since $t = -1.343$ is not less than -2.576, the null hypothesis cannot be rejected and the surcharge angles of the low and medium grade oil sand at the load point can be considered equal.

Table 6.5 compares all of the bitumen content pairings. From these findings, it can be concluded that bitumen content has no significant effect on the surcharge angle of oil sand.

6.5.3 Effect of distance travelled on surcharge angles

Tests were performed to determine whether the distance that the oil sand travels on the belt has a significant effect on the surcharge angle. Two sample t tests (as those illustrated in Section 6.5.2) were performed, comparing the equality of the surcharge angle of oil sand at the load and discharge points for both summer and winter values. Because the sample

Table 6.5 Effect of bitumen content on surcharge angles

Distance from load point (m)	Pairing (bitumen content)	t (calculated)	t (from tables)
0	Low-Medium	-1.343	2.576
	Low-High	-0.676	2.576
	Medium-High	0.69	2.576
1500	Low-Medium	-0.293	2.576
	Low-High	1.169	2.779
	Medium-High	1.255	2.576

variances between the respective pairs are virtually identical, using the F statistic to test for variance equality is not required. As illustrated in Table 6.6, at a 0.001 level of significance, and an infinite number of degrees of freedom, the distance travelled does has a significant effect on the surcharge angle of oil sand.

Table 6.6 Effect of distance travelled on surcharge angle

Summer/ Winter	t (calculated)	t (from tables)
Summer	11.580	2.576
Winter	4.938	2.576

6.6 Significance of an Accurate Surcharge Angle

As stated in Section 6.1, the surcharge angle of oil sand may be related to many factors. Three factors, the bitumen content, ambient temperature, and the distance travelled were felt to be the most significant, and were subsequently investigated.

Field investigations have shown that the bitumen content has no significant effect on the value of the surcharge angle. Therefore, the recorded surcharge angles of all three ranges of bitumen content from summer measurements can be grouped to determine the surcharge angle of oil sand. The average surcharge angle calculated from summer data was 9.4° at the load point and 7.5° at the discharge point. The 99% confidence intervals were (9.1°, 9.7°) and (7.2°, 9.7°), respectively. A statistical proof is shown in Section 6.5.2 and results are tabulated in Table 6.5.

As suspected, the ambient temperature does affect the surcharge angle of oil sand. It does so by increasing the lump size, thereby raising the surcharge angle. However, this does not necessarily mean that the amount of material transported by the belt conveyor is more in the winter than in the summer. The larger lumps create larger void spaces which reduce the amount of material on the belt. One can only assume that the increased surcharge angle in the winter months compensates for the increased voids created by the frozen lumps. This assumption allows the use of the summer results for design purposes. The values for winter surcharge angles were 13.8° at the load point, and 12.2° at the discharge point. The 99% confidence intervals were (13.2°, 14.4°) and (11.6°, 12.8°), respectively.

The results of these investigations allow two important conveying parameters to be quantified. The first is the cross-sectional area of the material on the belt. The second is the maximum amount of material that can be put on the belt before spillage occurs near the discharge point.

6.6.1 Cross-sectional area

In order to provide accurate estimates of belt conveyor capacity, belt tensions and power requirements, the surcharge angle of oil sand must be known to determine the amount of material that can be transported by a belt conveyor. It was illustrated in Section 6.0 how a slight discrepancy of 10° in the surcharge angle significantly affects the theoretical cross-sectional area of the material. Therefore, it is imperative to know the accurate value. This value must not be under-estimated, because the belt conveyor may not have enough power, or the belt may not be strong enough to transport the oil sand. Conversely, it must not be over-estimated. As a result, a cost-efficient design can be achieved by knowing an accurate value for the surcharge angle.

The results from the field investigations showed that the surcharge angle at the load point during summer was 9.4°. For design purposes, a surcharge angle of 10° may be safely assumed for oil sand conveying to provide a slight safety factor in the belt tension calculations.

6.6.2 Material spillage

When material spills off of the edge of a belt conveyor, it may trip the conveyor off causing unnecessary downtime. Damage can occur to the idlers and belt, and require additional clean-up. Also, material spillage poses a threat to the safety of personnel working near a conveyor installation.

When the material is loaded onto the belt, it forms the approximate shape of an arc with a given surcharge angle. While travelling on the belt, it is subjected to many vibrations, and the material starts to slump, causing the surcharge angle to lower. Subsequently, the material moves towards the edge of the belt. Figure 6.4 illustrates this phenomenon.

Knowing the surcharge angle at both the load and discharge points allows calculation of the maximum amount of material that can be placed on the belt before spillage occurs at the discharge point (i.e., the minimum edge distance at the load point). The cross-sectional area at the discharge point must be calculated to do this, (assuming the material has slumped to the edge of the belt with a surcharge angle of 7.5°). The corresponding dimensions of the cross-section at the load point (using a surcharge angle of 9.4°) can then be determined once this area is known. Figure 6.5 illustrates this concept.

From Figure 6.5 it can be seen that the trapezoidal area is larger at the discharge point than at the load point, because there is no edge distance. Because the cross-sectional areas at both points must be equal, the area increase in the trapezoidal area must be offset by a decrease in the surcharge area. As Figure 6.5 shows, the surcharge area at the discharge point is less than at the load point.

A 1524-mm wide belt is used as an example. Assuming a completely full belt at the discharge point, and a final surcharge angle of 7.5°, the calculated cross-sectional area is 0.3041 m^2. The corresponding area at the load point with a surcharge angle of 9.4° possesses an edge distance of 1.2 cm. This is substantially less than the CEMA recommended edge distance of 10.6 cm. However, if the extreme recorded surcharge angles are used, (i.e., 13.5° at the load point, and 5° at the discharge point), then the calculated edge distance is 4.3 cm. This is still much less than the CEMA recommended edge distance. One point to consider, is that

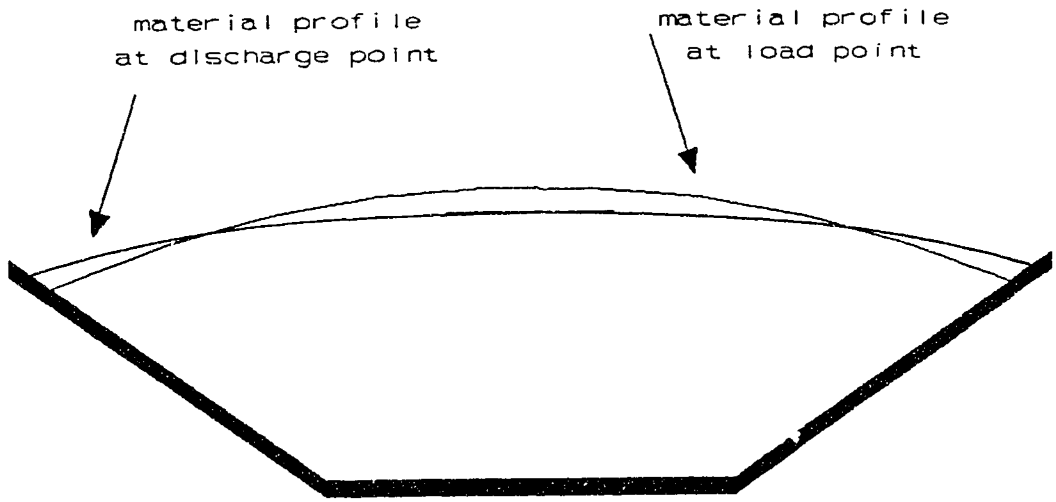

Figure 6.4 Oil sand profile at load point and discharge point

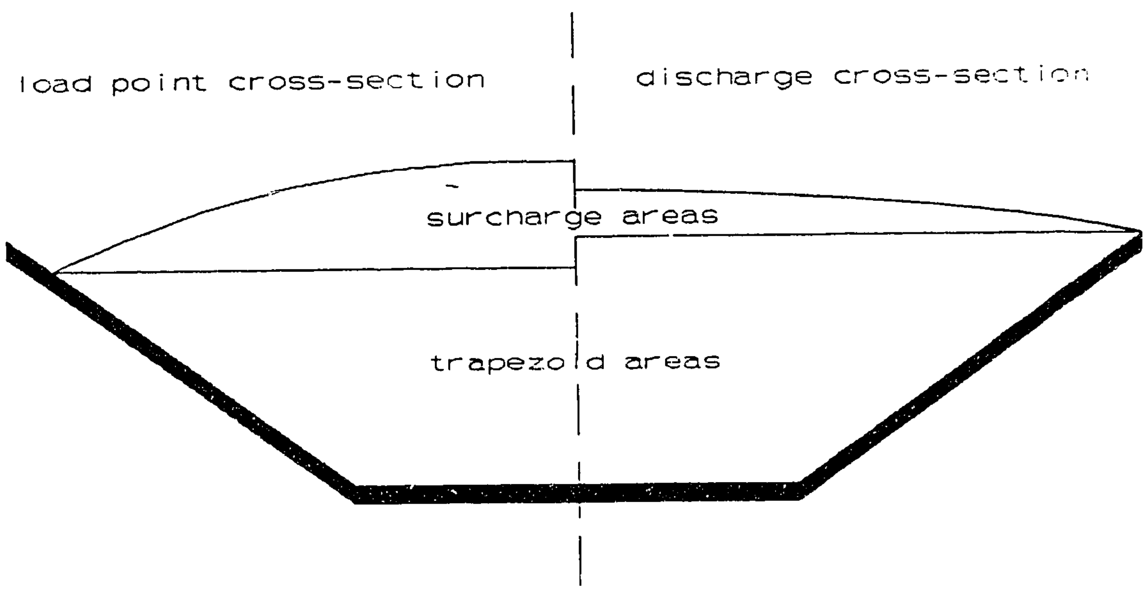

Figure 6.5 Cross-section comparisons

all design standards' recommended edge distances are generic, that is, they apply to all materials. Obviously, materials with higher surcharge angles than that of oil sand exist, and they likely undergo a larger change in their surcharge angles between the load and discharge points. This would explain the discrepancy between the calculated edge distances for oil sand and the recommended value by CEMA. As a result, if the edge distances recommended by the various design standards are followed, oil sand spillage will be minimized. Table 6.7 shows the calculated and CEMA recommended edge distances for three common belt widths for the average surcharge angles, summer extremes and winter extremes.

Table 6.7 Calculated and recommended edge distances

Surcharge angle	Belt width and corresponding edge distance (mm)					
	1524 mm		1829 mm		2134 mm	
	Actual	CEMA	Actual	CEMA	Actual	CEMA
Summer mean 9.4° to 7.5°	12	106	14	123	17	140
Summer extreme 5.0° to 13.5°	43	106	56	123	61	140
Winter extreme 10° to 16°	31	106	36	123	41	140

7. CONCLUSIONS

1. The most significant conclusion that arises from this thesis is the value of the oil sand surcharge angle. The field investigations performed conclude that the surcharge angle is 9.4° in the summer and 13.8° in the winter. However, as illustrated in Section 6.4, the mass conveying rate is virtually identical in the winter and summer because of the lower winter density. As a result, a conservative value of 10° may be used year round to calculate a conveyor's mass conveying rate assuming that one uses the unfrozen density.

2. Statistical studies were carried out to determine the effect that the bitumen content has on the surcharge angle. It was proven that the bitumen content has no significant effect on the surcharge angle. The factor that was found to have significant influence on the surcharge angle, however, was the distance that the oil sand travels on the belt. Field studies showed that the surcharge angle decreases once the oil sand is loaded onto the belt. It starts at 10° and stabilizes at 7.5°.

3. The K_t values (temperature correction factors) recommended by CEMA (with respect to the effect of ambient temperature on belt bending resistance) do not fully apply to the belting used in oil sand conveying. Modified K_t values that apply specifically to the belting used in oil sand conveying were measured and found to differ significantly from those recommended by CEMA. Between 1°C and 5°C the measured values are a maximum of 20% higher than those recommended by CEMA. This discrepancy decreases until both are assumed to be equal between -25°C and -30°C. Similar results regarding the rolling resistance of large idlers were obtained by Golosinski and Cholewa.

4. These results are corroborated by comparing drive power requirements of an existing belt conveyor (unloaded) at different ambient

temperatures with the calculated theoretical power requirements of that same belt obtained using both the modified K_t values presented in this thesis and the recommended CEMA values. Figure 5.3 illustrates this and shows that the modified K_t values more closely estimate the real life values.

5. Another conclusion that can be made is that the oil sand stickiness effects the power required to drive a belt conveyor. This is illustrated in Figure 5.3, where it is shown that the actual power required to drive an unloaded belt conveyor is substantially higher than the theoretical power. The most logical explanation of this larger power requirement is that the power demand increases as the oil sand sticks to the belt, idlers and pulleys (see Section 5.2). This effect appears to cease at approximately -23°C. Actual measurements conducted on empty belts show that the power requirements increase as the temperature decreases to approximately -18°C, and then take a sudden plunge in the temperature range of -18°C to -23°C. Once temperatures are below -23°C, the power demand starts to increase again. From this, the conclusion can be made that the oil sand loses its stickiness between -18°C and -23°C. This is when the majority of it falls off the belt and idlers, which, inturn, reduces the power required to drive the belt at that temperature. It more than compensates for the increased belt flexure forces required with the decreasing temperature. Once the temperature falls below -23°C, the belt flexing forces are all that change, and the drive power starts to increase again.

6. Use of the CEMA belt conveyor design method is preferred when designing a conveyor to be used in a new environment with widely-varying climatic and operating conditions because the resistance to movement is broken into unit components. This allows individual analysis of each

resistance, and ultimately provides greater accuracy.

7. The final conclusion presented in this thesis is that the mass conveying rates are virtually identical on both the summer and the winter. As a result, the summer data (i.e., a surcharge angle of 10°, and an oil sand density of 1284 kg/m^3) should be used when calculating the mass conveying rate of a belt conveyor.

8.0 RECOMMENDATIONS

Recommendations are categorized as those pertaining to mining operations, and those regarding further research into the effect of temperature on belt bending and oil sand stickiness.

1. The surcharge angle of 10°, and the oil sand density of 1284 kg/m^3, are practical numbers that can be of great use in calculating an accurate value for the mass conveying rate of oil sand conveyors. The theoretical conveying rates used by operators in their conveyor design tend to overestimate the amount of oil sand that their conveyors are capable of transporting. Using the values presented in this thesis would provide more realistic and reliable results, and would provide the basis for more accurate conveyor designs in oil sand mines.

2. Regarding further research into the effect of temperature on belt bending, tests should be carried out to determine the actual force required to flex belting around pulleys. The tests would need to be performed over a wide range of temperatures, and with varying belt tensions, to establish a relationship between temperature and flexing forces, as well as belt tension forces. Results obtained from these tests would enhance belt tension calculations, and those which define the overall theoretical power requirements for oil sand conveyors. These results could then be compared to recommended values from CEMA, ISO and DIN to determine how oil sand belting varies from more conventional belting, with respect to belt bending forces, over varying temperatures and tensions.

3. The most significant factor that makes oil sand conveyors require more power than other belt conveyors is the stickiness of the oil sand. As mentioned in this thesis, we know that this unknown force exists, but have no significant data with regards to the degree to which

it affects the power required to drive belt conveyors. Quantifying the "stickiness" of the oil sand at different temperatures, and relating this to how it influences drive power requirements, is essential in determining accurate and reliable results.

9.0 References

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Appendix A - Illustration of Program Use

Appendix A illustrates the possible applications of the belt conveyor tension programs for two typical conveyor layouts. The first example deals with a single-drive conveyor (see Figure A.1), and demonstrates the flexibility of adding accessory tensions such as belt cleaners and skirt boards. The second example deals with a dual-drive conveyor (see Figure A.3), and demonstrates the calculation of vertical curve radii during start-up. The input information provided for the each example conveyor is described in the following paragraphs.

Information describing a belt conveyor is entered via two data files. The first contains the belt specifications. The second contains the coordinates of each station, the mass of material on the belt (as a fraction of the full theoretical load) on a particular section, and if the station is an accessory, it defines the type of accessory. To better understand the numbers within each data file, the input values are described below. The first data file contains:

Line 1	metric or standard US units (M, I)
Line 2	length (m, ft), width (mm, in.), belt mass (kg/m, lb/ft), velocity (m/s, fpm), bottom cover thickness (m, in.), elevation change between head and tail stations (m, ft), percent belt sag, belt breaking strength (N, lb)
Line 3	belt type (fabric, steel, etc.)
Line 4	mass of one carry idler (kg, lb), mass of one return idler (kg, lb), carry idler spacing (m, ft), return idler spacing (m, ft), troughing angle (degrees), carry idler diameter (mm, in.), return idler diameter (mm, in.)
Line 5	carry idler class (CEMA A - E)
Line 6	return idler class (CEMA A - E)

Line 7 material density (kg/m^3, lb/ft^3), surcharge angle (degrees), fraction of full theoretical load (0 -1.0)

Line 8 feed velocity (m/s, fpm), skirt board length (m, ft) height of material on skirts (m, in.), material friction factor, ambient temperature (°C, °F)

Line 9 number of drive pulleys (1 or 2), motor efficiency (0 - 1.0), motor rpm

Line 10 primary wrap angle (degrees), friction coefficient, secondary wrap angle if necessary (degrees), friction coefficient, if necessary

The second data file contains:

Line 1 tail station number, head station number, first drive pulley station number, second drive pulley station number (if necessary)

Line 2 first station, X and Y coordinates of the first station

Line 3 second station, X and Y coordinates of the second station fraction of full load between first and second stations, and identification of station type

Line 4 identical to line 3 except describing station three

Lines 5 and higher describe each successive station of the belt

The description of Line 3 of the second data file requires clarification, with respect to the identification of station type. 'NO' is used if the station is not an accessory. However, if the station is an accessory, then 'BC' is used to identify a belt cleaner, 'SB' for skirtboard, 'PL' for plow, 'PU' for pulley, and 'MA' for material acceleration.

Once the input files have been created the program is run interactively. When running the program, the user is prompted to input specific information pertaining to accessories as the calculation progresses. As mentioned in Chapter 4, the output consists of:

1. a plot of the belt tensions
2. tabulated data related to the belt profile of the carry side of the belt that includes station number, coordinates of each station, rise and run between stations, slope between stations, length of each section, and the percent of the full theoretical load on each section
3. tabulted data describing the entire belt including the station number and type, the K_y, K_x, and K_t factors, tensions accumulated on each section of the belt, and finally the total belt tension at each station
4. tight side tension, slack side tension, effective tension, power required, and minimum allowable tail tension
5. input echoing
6. accleration/vertical curve analysis (optional)
7. deceleration calculations (optional)

A.1 Example #1

The sample conveyor used for Example #1 has a single-drive pulley, is 1524-mm wide, and has a fully loaded capacity of about 5200 tonnes/hour. It is assumed to operate in an ambient temperature of -40°C. Figure A.1 shows the conveyor layout.

The calculated effective tension in the belt of this conveyor is 221 kN, translating into a power requirement of 1210 kW, assuming use of a motor with 95% efficiency. The maximum belt tension is 357 kN, meaning that one needs to select an St 2500 steel belt (breaking stength of 3810

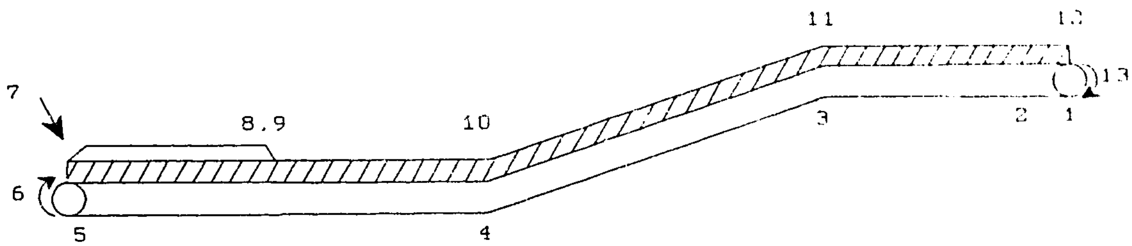

Figure A.1 Conveyor layout for Example #1

kN resulting in a safety factor of about 10). The minimum allowable tail tension, calculated with the 1.22 m idler spacing, is 51 kN. The minimum tension encountered along the carry side of the belt is 169 kN, suggesting that the carry side idler spacing may be increased, without exceeding the sag limits. Increasing the idler spacing increases the minimum allowable tail tension, and decreases the number of idlers required, resulting in cost savings.

Belt specification file for example #1 (Metric units)

```
M
823 1524 66.224 5.207 0.0076 0.0318 17.8 1 3810000
STEEL
54.2 49.6 1.219 3.048 35 152.4 152.4
E
E
1226.1 10 1.0
0 10.7 0.3048 0.275 -40
1 0.95 1750
180 0.3
```

Belt section data file for example #1 (Metric units)

```
6  12  12
1  823  18.3
2  823  18.3  0  'BC'
3  579.1  18.3  0  'NO'
```

```
4  304.8  0  0  'NO'
5  0  0  0  'NO'
6  0  0.9  0  'PU'
7  0  0.9  0  'MA'
8  10.7  0.9  100  'NO'
9  10.7  0.9  0  'SB'
10  304.8  0.9  100  'NO'
11  579.1  19.2  100  'NO'
12  823  19.2  100  'NO'
13  823  18.3  0  'PU'
```

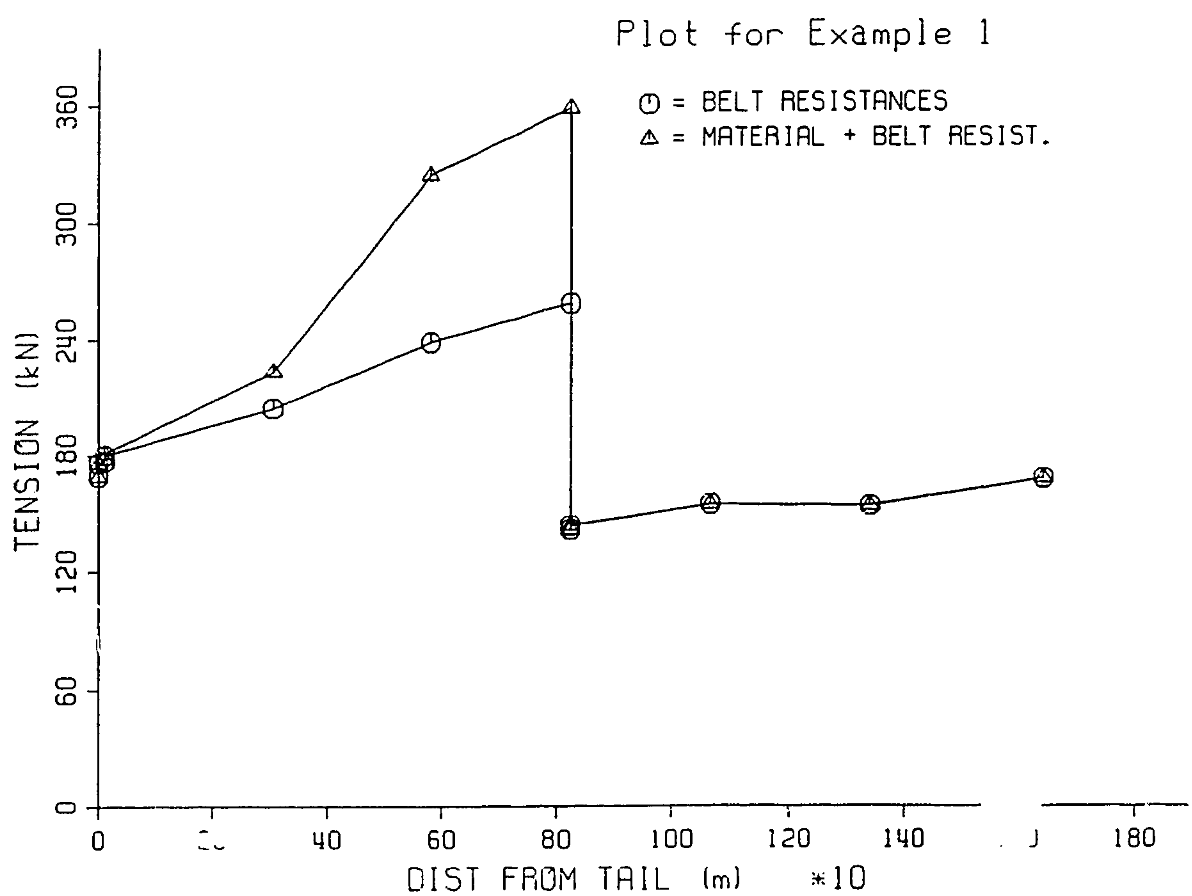

Figure A.2 Graphical form of belt tensions for Example #1

EXAMPLE #1 Metric units input Fully loaded

CARRY SIDE PROFILE

Tail is station 6

All lengths are in meters and slope is in degrees

STN	X-coord	Y-coord	RISE	RUN	SLOPE	LENGTH	%LOAD
6	.0	0.9					
8	10.7	0.9	.0	10.7	.0	10.7	100.0
10	304.8	0.9	.0	294.1	.0	294.1	100.0
11	579.1	19.2	18.3	274.3	3.8	274.9	100.0
12	823.0	19.2	.0	243.9	.0	243.9	100.0

TENSIONS BY SECTION

	STN	KY	KX	TX	TY	TYM	TM	TB	TACC	TOTAL
BC	2								2.003	143.061
RET	3	.219	4.526	3.311	7.135			.000		153.508
RET	4	.219	4.526	3.732	8.043			-11.898		153.386
RET	5	.219	4.526	4.138	8.917			.000		166.441
PU	6								2.471	168.911
MA	7								7.474	176.385
CARRY	8	.346	12.498	.401	.494	.687	.000	.000		177.968
SB	9								2.475	180.443
CARRY	10	.339	12.498	11.024	13.320	18.526	.000	.000		223.314
CARRY	11	.326	12.498	10.305	11.951	16.621	49.641	11.898		323.730
CARRY	12	.317	12.498	9.143	10.324	14.358	.000	.000		357.555
PU	13								4.444	141.050

T1 = 357.555 kN
T2 = 141.050 kN
EFFECTIVE TENSION = 220.949 kN
REQUIRED POWER = 1210. kW
MINIMUM ALLOWABLE TO = 51.305 kN

BELT PARAMETERS

BELT TYPE:	STEEL	CONVEYOR CAPACITY (TPH):	5178.51
BELT LENGTH (m):	823.00	MAT. DENSITY (kg/m3):	1226.16
BELT WIDTH (mm):	1524.00	MAT. WEIGHT (kg/m):	276.59
BELT WEIGHT (kg/m):	66.29	MAT. SURCHARGE (DEG):	10.00
BELT VELOCITY (m/s):	5.22	AMBIENT TEMPERATURE (C):	-40.00
SAG PERCENT:	1.00	KT:	3.00

IDLER PARAMETERS

WEIGHT OF CARRY IDLER (kg):	54.20	WEIGHT OF RETURN IDLER (kg):	49.60
CARRY IDLER SPACING (m):	1.22	RETURN IDLER SPACING (m):	3.05
CARRY TROUGH ANGLE (DEG):	35.00		

DRIVE PARAMETERS

NUMBER OF DRIVES:	1	WRAP ANGLE (DEG):	180.00
DRIVE STATION:	12	WRAP FACTOR:	.64
REQUIRED POWER:	1209.94	EFFICIENCY:	.95

A.2 Example #2

The second example demonstrates the design of conveyors with concave vertical curves. The program calculates the minimum curve radius for a given load configuration and motor starting torque. Because the highest tensions are encountered during start-up, the force induced upon the belt during start-up must be added to the normal operating tensions to define the highest belt tensions and, subsequently, the minimum concave curve radius.

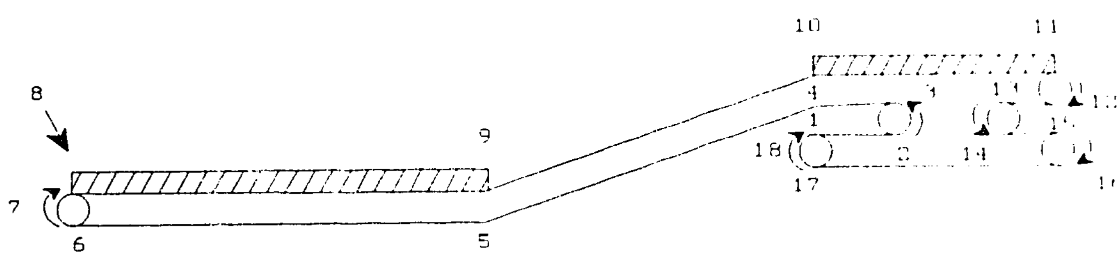

Figure A.3 Conveyor layout for Example #2

The point under analysis is station nine in Figure A.3. The worst possible load configuration is assumed along with an ambient temperature of -40°C. This produces the highest possible belt tensions during start-up and, therefore, the largest curve radius. Use of four 373-kW motors was assumed for calculating the extra start-up tension. The WK^2 values for each motor, reducer, coupling and pulley were 0.413, 0.083, 0.022, and 0.029 N-m^2 (144, 28.8, 7.5 and 10 lb-in^2), respectively (refer to Section 4.2.9). The motor start-up torque is set at 175%, while the maximum allowable belt tension during start-up is 15% of the belt's breaking strength, or 572 kN.

After running the program, the calculated belt acceleration time, limited by belt strength was 5.42 seconds. However, the acceleration time limited to motor torque was 5.92 seconds. Therefore, the minimum allowable accleration time of 5.92 sec results in a 157.6 m concave vertical radius with the belt loaded at station nine, and 814.4 m for an empty belt.

In addition to the acceleration calculations, one can perform deceleration calculations. An assumption that the conveyor coasts to a stop is made, and the volume of material calculated that would be discharged in that time (refer to Section 4.2.9). The deceleration time calculated for the load configuration under consideration was 11.52 sec. In this time, a volume of 6.77 m^3 would be discharged from the conveyor.

Belt specification data file for example #2

```
M
823 1524 66.224 5.207 0.0076 0.0318 17.8 1 3810000
STEEL
54.2 49.6 1.219 3.048 35 152.4 152.4
E
E
1226.1 10 1.0
0 10.7 0.3048 0.275 -40
2 0.95 1750
180 0.3 180 0.3
```

Belt section data file for example #2

```
7  12  15  17
1  762  17.4
2  784.9  17.4  0  'NO'
3  784.9  18.3  0  'PU'
4  579.1  18.3  0  'NO'
5  304.8  0  0  'NO'
6  0  0  0  'NO'
7  0  0.9  0  'PU'
8  0  0.9  0  'MA'
9  304.8  0.9  100  'NO'
10  579.1  19.2  0  'NO'
11  823  19.2  100  'NO'
12  823  18.3  0  'PU'
13  800.1  18.3  0  'NO'
```

14	800.1	17.4	0	'PU'
15	815.3	17.4	0	'NO'
16	815.3	16.5	0	'PU'
17	762	16.5	0	'NO'
18	762	17.4	0	'PU'

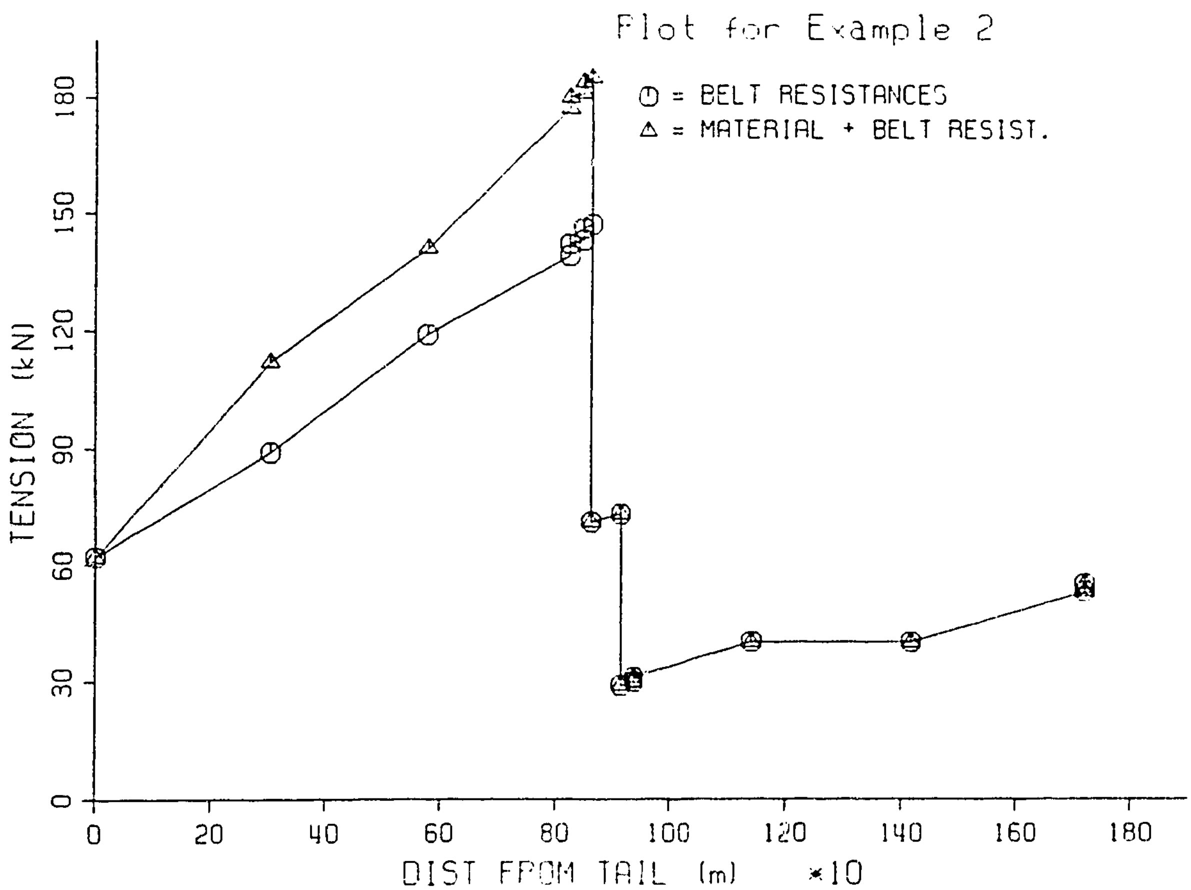

Figure A.4 Graphical form of belt tensions for Example #2

EXAMPLE #2 Dual drive Vertical curve analysis

CARRY SIDE PROFILE

Tail is station 7

All lengths are in meters and slope is in degrees

STN	X-coord	Y-coord	RISE	RUN	SLOPE	LENGTH	%LOAD
7	.0	0.9					
9	304.8	0.9	.0	304.8	.0	304.8	100.0
10	579.1	19.2	18.3	274.3	3.8	274.9	.0
11	823.0	19.2	.0	243.9	.0	243.9	100.0

TENSIONS BY SECTION

	STN	KY	KX	TX	TY	TYM	TM	TB	TACC	TOTAL
RET	2	.219	4.526	.311	.670			.000		29.997
PU	3								1.221	31.198
RET	4	.219	4.526	2.794	6.021			.000		40.013
RET	5	.219	4.526	3.732	8.043			-11.898		39.891
RET	6	.219	4.526	4.138	8.917			.000		52.946
PU	7								1.431	54.377
MA	8								7.474	61.851
CARRY	9	.393	12.498	11.426	15.995	22.246	.0	.000		111.517
CARRY	10	.234	10.658	8.784	8.579	.0	.0	11.898		140.777
CARRY	11	.352	12.498	9.143	11.453	15.9[illegible]	.0	.000		177.301
PU	12								2.792	180.094
RET	13	.219	4.526	.311	.670			.000		181.075
PU	14								2.605	183.680
RET	15	.219	4.526	.206	.445			.000		184.330
PU	16								2.635	70.407
RET	17	.219	4.526	.724	1.559			.000		72.689
PU	18								1.612	28.883

T1 = 184.330 kN
PRIMARY EFFECTIVE TENSION = 116.560 kN
T3 = 70.407 kN
T2 = 28.997 kN
SECONDARY EFFECTIVE TENSION = 45.419 kN
TOTAL EFFECTIVE TENSION = 161.978 kN
REQUIRED POWER = 887. kW
MINIMUM ALLOWABLE TO = 51.305 kN

BELT PARAMETERS

BELT TYPE:	STEEL	CONVEYOR CAPACITY (TPH):	5178.51
BELT LENGTH (m):	823.00	MAT. DENSITY (kg/m3):	1226.16
BELT WIDTH (mm):	1524.00	MAT. WEIGHT (kg/m):	276.59
BELT WEIGHT (kg/m):	66.29	MAT. SURCHARGE (DEG):	10.00
BELT VELOCITY (m/s):	5.22	AMBIENT TEMPERATURE (C):	-40.00
SAG PERCENT:	1.00	KT:	3.00

IDLER PARAMETERS

WEIGHT OF CARRY IDLER (kg):	54.20	WEIGHT OF RETURN IDLER (kg):	49.60
CARRY IDLER SPACING (m):	1.22	RETURN IDLER SPACING (m):	3.05
CARRY TROUGH ANGLE (DEG):	35.00		

DRIVE PARAMETERS

NUMBER OF DRIVES:	2	PRIMARY WRAP ANGLE (DEG):	180.00
DRIVE STATION 1:	15	PRIMARY WRAP FACTOR:	.64
DRIVE STATION 2:	17	SECONDARY WRAP ANGLE (DEG):	180.00
POWER RATIO:	2.57	SECONDARY WRAP FACTOR:	.64
REQUIRED POWER:	887.01	TOTAL WRAP FACTOR:	.18
EFFICIENCY:	.95		

ACCELERATION CALCULATIONS

Motor Power	373. kW	
Number of motors	4	
Belt breaking strength	3810 kN	
Max operating tension	381 kN	
Max starting tension	572 kN	
Acceleration time (belt strength limitations)		5.42 sec
Acceleration time (torque limitations)		5.92 sec

Minimum allowable acceleration time is 5.92 sec due to TORQUE limitations

The total tension at station 9 is 477. kN
This requires a curve RADIUS of 157.6 m with material on the belt
This requires a curve RADIUS of 814.4 m with an empty belt

DECELERATION CALCULATIONS

Deceleration time	11.52 sec	
Weight in hopper	8288. kg	Assumes belt is loaded
Volume in hopper	6.77 cu m	

Appendix B - Program Source Code

This Appendix contains a listing of the documented source code for the computer described in Chapter 4 and Appendix A of this thesis. The main program is used as a driver to call each of the main subroutines as required. Subroutines are presented in the order in which they are called.

The codes "$Include:'COM'" and "$Include:'CHAR'" are "Include" statements that allow all common variables to be passed between subroutines. The 'COM' file contains integer and real variables, while the 'CHAR' file contains character variables.

The following is a list of global variables contained within the "Include" files "COM" and "CHAR". They are separated into real, integer, and character groups, and displayed in alphabetical order. Local variables are defined within the subroutine in which they are used.

<u>Real global variables</u>

AI	idler friction factor
ALPHA	material surcharge angle
BCT	belt cover thickness
BETA	troughing angle
BREAK	breaking strength of belt
C4,C6,C7,A1,AA1	factors (from Behrends et al [1]) used in calculation of friction factors
CS	skirtboard friction factor
DERAT	overall derating (0.0 - 1.0) of material cross-section
EFF	motor efficiency
FRCO1,FRCO2	primary and secondary drive pulley friction coefficients
H	total lift between tail and head stations
HS	height of material on skirtboards
ID,IDR	idler diameters (carry, return)
KT	temperature correction factor
KXR	return idler friction resistance factor
KYR	return belt flexure resistance friction factor
L	center-to-center belt length
LB	length of skirtboards
LOAD(100)	array specifying the percentage of a full load on a particular section

PWA,SWA	primary and secondary wrap angles
QM	conveyor capacity (t/h)
RAD(100)	array containing slope of each belt section (in radians)
RHO	material density
RISE(100)	array containing rise of each belt section
RUN(100)	array containing run of each belt section
SLOPE(100)	array containing slope of each belt section (in degrees)
LENGTH(100)	array containing length of each belt section
RPM	motor revolutions per minute
SAGG	allowable belt sag (percent)
SCW	scraper width
SIC,SIR	idler spacing (carry and return)
T1	maximum belt tension
T2	slack side belt tension
T2B	slack side belt tension (minimum required tail tension less all resistances to the last drive pulley)
T3	belt tension between primary and secondary drive pulleys
TACC(100)	array containing resistances from accessories
TB(100)	array containing tensions from belt lifting and lowering
TCT	top cover thickness of belt
TECE	first estimate of effective tension performed in Subroutine CEMA
TEF(100)	array containing the total effective tension accumulated at each station
TEFF	total effective tension calculated in Subroutine SECTN2
TEMP	ambient temperature
TEP	primary effective tension
TES	secondary effective tension
THICK	belt thickness
TM(100)	array containing tensions from material lift
TO	minimum allowable tail tension
TOTC(100)	array containing total belt tension at each station
TXC(100)	arrays containing idler resistances over each section (carry and return)
TXR(100)	
TYC(100)	arrays containing belt flexure resistance over each section (carry and return)
TYR(100)	
TYM(100)	array containing material flexure resistance over each section
V	belt velocity
VO	material feed velocity
W	belt width
WB	linear belt weight
WIC,WIR	idler weight (carry and return)
WM(100)	array containing material weight along each section
WMAT	material weight used in Subroutine CEMA
X(100)	coordinates of each station
Y(100)	

<u>Integer global variables</u>

DRIVE1	primary drive pulley station number
DRIVE2	secondary drive pulley station number
HEAD	head pulley station number

NDU	number of drive units
NUM	total number of sections
SECTION(100)	array containing section numbers
STAT(100)	array containing station numbers
TAIL	tail pulley station number

<u>Character global variables</u>

ACCESS(100)	array defining accessory type for each section
BTYP	belt type
IC,IR	idler class for carry and return (A-E)
JOB	job title
PRETEN	read from screen to determine if pretension is to be manually set (Y or N)
UNITS	metric or imperial input units (M or I)

List of Subroutines

```
      PROGRAM BELT
*
*     This is the main driver program that is used to call each of the
*     main  subroutines.  In addition, the user-defined output filename
*     is established, as well as the first line of the output file which
*     will contain the desired job title.
*
*     Routines called:  CEMA
*                       PROFIL
*                       SECTN1
*                       SECTN2
*                       PLOTT
*
*     Output file:      Filename to be user-defined
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
*     User-defined output filename assigned to Unit 7
*
      PRINT*,'DEFINE THE DESIRED OUTPUT FILENAME'
      READ(*,50)OUTPUT
      OPEN(UNIT=7,FILE=OUTPUT)
*
*     First line of the output file containing the desired job title.
*
      PRINT*,'WHAT IS THE JOB TITLE?'
      READ(*,50)JOB
      WRITE(7,50)JOB
      WRITE(7,*)' '
*
*     Subroutine called to establish the first estimate of the Effective
*     Tension of the belt conveyor.
*
      CALL CEMA
*
*     Subroutine called to calculate the carry side profile of the belt
*     conveyor.
*
      CALL PROFIL
*
*     Subroutine called to calculate the first estimate of the belt
*     tensions along the entire length of the belt.
*
      CALL SECTN1
*
*     Subroutine called to calculate the second and third estimates of the
*     belt tensions along the entire length of the belt.
*
      CALL SECTN2
*
 50   FORMAT(A)
      STOP
      END
```

```
      SUBROUTINE CEMA
*
*     This subroutine calls subroutine READ1 and reads the first data
*     file, a belt specification file containing general information
*     regarding the belt conveyor.  With this information it calculates
*     an initial estimate of the effective tension of the belt conveyor.
*
*     Routines called:  READ1
*                       AREA
*
*
*     Local variables:  KXC - carry idler friction factor
*                       KYC - carry belt and material friction factor
*                       ATXC - carry and return idler resistance
*                       ATXR
*                       ATYC - carry and return belt resistance
*                       ATYR
*                       ATYM - material flexure resistance
*                       ATM - material lift resistance
*                       TAM - material acceleration resistance
*                       BCTMET - belt cover thickness (mm)
*                       SICMET - carry idler spacing (m)
*                       IDMET - idler diameter (mm)
*                       WBMET - belt weight (kg/m)
*                       C3,LSP - factors used in friction coefficient
*                       KY1 - friction factor due to imprint
*                       KY23 - friction factor due to belt and material
*                              flexure
*                       IDD,IDDR - column of array for idler friction *
                           factor
*                       ICC,ICR - row of array for idler friction factor
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
      REAL KXC,KYC,ATXC,ATXR,ATYC,ATYR,ATYM,ATM,TAM
      REAL VMET,BCTMET,SICMET,IDMET,WBMET,C3,LSP
      REAL KY1,KY23
      INTEGER IDD,ICC,KK,ICR,IDDR
*
*     2-Dimensional array containing the values for the resistance due to
*     idler rotation (or idler friction factor).
*
      DIMENSION AIDLER(5,4)
      DATA AIDLER /3*2.3,2*0.,4*1.8,0.,0.,0.,2*1.5,2.8,4*0.,2.4/
*
*     Subroutine called to read first data file
*
      CALL READ1
*
*     If the input units are designated as metric, they are converted to
*     imperial units for the purpose of calculations.
*
      IF(UNITS.EQ.'M'.OR.UNITS.EQ.'m')CALL METIMP
```

```
*     Subroutine called to calculate the cross-sectional area of the
*     material being transported on the belt.
*
      CALL AREA
*
*     Defines the row in which the idler friction factor is found in Array
*     Aidler(5,4) based on the idler class of A,B,C,D, or E  (ICC is the
*     idler class for carry side, ICR is the idler class for the return
*     side).
*
      IF(IC.EQ.'A'.OR.IC.EQ.'a')THEN
        ICC=1
       ELSEIF(IC.EQ.'B'.OR.IC.EQ.'b')THEN
        ICC=2
       ELSEIF(IC.EQ.'C'.OR.IC.EQ.'c')THEN
        ICC=3
       ELSEIF(IC.EQ.'D'.OR.IC.EQ.'d')THEN
        ICC=4
       ELSEIF(IC.EQ.'E'.OR.IC.EQ.'e')THEN
        ICC=5
       ENDIF
*
       IF(IR.EQ.'A'.OR.IR.EQ.'a')THEN
        ICR=1
       ELSEIF(IR.EQ.'B'.OR.IR.EQ.'b')THEN
        ICR=2
       ELSEIF(IR.EQ.'C'.OR.IR.EQ.'c')THEN
        ICR=3
       ELSEIF(IR.EQ.'D'.OR.IR.EQ.'d')THEN
        ICR=4
       ELSEIF(IR.EQ.'E'.OR.IR.EQ.'e')THEN
        ICR=5
       ENDIF
*
*     A warning is printed to the screen if the specified idler diameters
*     are standard.  If an idler diameter is not standard, then it is
*     automatically set at a minimum of 3.5 inches, or a maximum of 7
*     inches.
*
      IF(ID.LT.3.5.OR.ID.GE.7.5)THEN
        PRINT*,' '
        PRINT*,'******Idler diameter does not coincide with******'
        PRINT*,'*************standard diameters****************'
        PRINT*,'*******Must be between 4 and 7 inches***********'
        PRINT*,' '
       ENDIF
*
       IF(ID.LT.3.5)ID=3.5
       IF(ID.GE.7.5)ID=7.
*
       IF(IDR.LT.3.5)IDR=3.5
       IF(IDR.GE.7.5)IDR=7.
*
*     Defines the column in which the idler friction factor is contained
*     within Array AIDLER(5,4), based on idler diameter (IDD for the carry
```

```
*     side, IDDR for the return side).
*
      IDD=ID+0.5-3.0
      AI=AIDLER(ICC,IDD)
      IDDR=IDR+0.5-3.0
      AIR=AIDLER(ICR,IDDR)
      WMAT=A*RHO*DERAT
      QM=WMAT*V*60./2000.
      KXC=((0.00068*(WB+WMAT)+(AI/SIC)))
      KXR=((0.00068*WB+(AIR/SIR)))
*
*     For this first estimate, the program automatically sets carry and
*     return belt and material flexure resistance friction factors.
*
      KYC=0.016
      KYR=0.015
*
*     Counter is set to 1
*
      KK=1
*
*     Temperature correction factors based on the ambient temperature are
*     established.
*
     IF(TEMP.GE.30.)THEN
        KT=1.0
        GOTO 25
      ELSEIF(TEMP.LE.-40.)THEN
        KT=3.0
        GOTO 25
      ELSEIF(TEMP.GE.20.)THEN
         KT=1.30
         GOTO 25
      ELSEIF(TEMP.GE.10.)THEN
        KT=1.38
        GOTO 25
      ELSEIF(TEMP.GE.0.)THEN
        KT=1.51
        GOTO 25
      ELSEIF(TEMP.GE.-10.)THEN
        KT=1.63
        GOTO 25
      ELSEIF(TEMP.GE.-15.)THEN
        KT=1.7
        GOTO 25
      ELSEIF(TEMP.GE.-25.)THEN
        KT=1.9
        GOTO 25
      ELSEIF(TEMP.GE.-30.)THEN
        KT=2.1
        GOTO 25
      ELSEIF(TEMP.GE.-40.)THEN
        KT=2.8
        GOTO 25
      ENDIF
```

```
*     Belt tension is calculated due to:
*          carry side idler resistance
*          return side idler resistance
*          carry side belt flexure resistance
*          return side belt flexure resistance
*          material flexure resistance
*          gravitational resistance
*          material acceleration resistance
*
 25   ATXC=L*KXC*KT
      ATXR=L*KXR*KT
      ATYC=L*KYC*WB*KT
      ATYR=L*KYR*WB*KT
      ATYM=L*KYC*WMAT
      ATM=H*WMAT
      TAM=2.8755E-04*QM*(V-VO)
*
*     Resistances are summed and 7% is added on to account for accessories
*     such as belt cleaners, skirtboards, plows, and belt flexure around
*     pulleys.
*
      TECE=ATXC+ATXR+ATYC+ATYR+ATYM+ATM+TAM
      TECE=TECE+0.07*TECE
*
      IF(KK.EQ.2)GOTO 999
*
*     Based on the paper by Behrends et al [1], an initial estimate for
*     the Ky factor to be used in Subroutine SECTN1 is calculated.
*
        VMET=V*5.08E-03
        BCTMET=BCT*25.4
        SICMET=SIC*3.048E-01
        IDMET=ID*25.4
        WBMET=WB*0.4536/.3048/100.
        WMATMET=WMAT*0.4536/.3048/100.
        IF(BETA.LT.40.)THEN
          C3=0.9
          LSP=1.08
          C4=0.75
          C6=2.26
          C7=0.83
        ELSE
          C3=1.1
          LSP=1.05
          C4=1.288
          C6=2.06
          C7=0.76
        ENDIF
        A1=100.*0.85*1.*C3*(VMET**.34)*(BCTMET**.235)*(SICMET**LSP)
     +/((IDMET**1.02)*(10**.055))
        AA1=A1*(10**0.055)
        KY1=A1*((WMATMET+WBMET)**1.3)
        KY23=C4*1.0*((WMATMET+WBMET)**C6)/((TECE/2205.)**C7)
        KYC=(KY1+KY23)/(WMATMET*100.+WBMET*100.)
        KK=2
```

```
        GOTO 25
*
 999  CONTINUE
      RETURN
      END
```

```
      SUBROUTINE METIMP
*
*     This subroutine converts metric input to imperial units for the
*     purpose of the calculations to be performed within the program.
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
      L=L/3.048E-01
      W=W/25.4
      WB=WB/0.4536*0.3048
      V=V/5.08E-03
      H=H/3.048E-01
      RHO=RHO/16.0279
      VO=VO/5.08E-03
      WIC=WIC/4.5359E-01
      WIR=WIR/4.5359E-01
      SIC=SIC/3.048E-01
      SIR=SIR/3.048E-01
      HS=HS/2.54E-02
      LB=LB/3.048E-01
      TEMP=TEMP*1.8+32
      BCT=BCT/2.54E-02
      ID=ID/25.4
      IDR=IDR/25.4
      SCW=SCW/2.54E-02
      BREAK=BREAK/4.448
*
      RETURN
      END
```

```
      SUBROUTINE READ1
*
*     This subroutine reads the first data file that contains information
*     pertaining to the belt's specifications.
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
      CHARACTER*12 SPEC
*
*      Define and open unit 3 to read
*
      PRINT*,'Define the belt specification filename'
      READ(*,10)SPEC
 10   FORMAT(A)
      OPEN(UNIT=3,FILE=SPEC,STATUS='OLD')
*
*      Read in type of units (metric or imperial)
*
      READ(3,11)UNITS
 11   FORMAT(A1)
*
*      Read in belt paramater inputs
*
      READ(3,*)L,W,WB,V,BCT,THICK,H,SAGG,BREAK
      READ(3,12)BTYP
 12   FORMAT(A6)
*
*      Read in idler parameter inputs
*
      READ(3,*)WIC,WIR,SIC,SIR,BETA,ID,IDR
      READ(3,13)IC
      READ(3,13)IR
 13   FORMAT(A1)
*
*      Read in material parameter inputs
*
      READ(3,*)RHO,ALPHA,DERAT
*
*      Read in ancilliary inputs
*
      READ(3,*)VO,LB,HS,CS,TEMP
*
*      Read in drive unit inputs (either 1 or 2 drive units)
*
      READ(3,*)NDU,EFF,RPM
      IF(NDU.EQ.1)THEN
        READ(3,*)PWA,FRCO1
      ELSEIF(NDU.EQ.2)THEN
        READ(3,*)PWA.FRCO1,SWA,FRCO2
      ENDIF
*
      RETURN
      END
```

```
      SUBROUTINE AREA
*
*     This subroutine calculates the cross-sectional area of the material
*     being transported on the belt.
*
*     Variables:  TROF is the troughing angle in radians
*                 SUR is the surcharge angle in radians
*                 AT is the trapezoid area of the material
*                 AS is the surcharge area of the material
*                 WID is the belt width
*
$INCLUDE:'COM'
      REAL TROF,SUR,AT,AS,WID
*
      WID=W
*
*      Convert the troughing and surcharge angles from degrees to
*      radians.
*
      TROF=BETA*3.14159/180.
      SUR=ALPHA*3.14159/180.
*
*      Calculate the trapezoidal and surcharge areas.
*
      AT=(0.371*WID+0.25+(0.2595*WID-1.025)*COS(TROF))*((0.2595*WID-
     +1.025)*SIN(TROF))
      AS=(((0.1855*WID+0.125+(0.2595*WID-1.025)*COS(TROF))/(SIN(SUR)))
     +**2.)*((SUR-(SIN(2.*SUR)/2.)))
*
*      Convert total area to square feet.
*
      A=(AT+AS)/144.
*
      RETURN
      END
```

```
      SUBROUTINE PROFIL
*
*     This subroutine calls Subroutine READ2 and then calculates the
*     length of each section of the carry side of the belt and writes the
*     data pertaining to the carry side profile to the output file.
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
*     Calls Subroutine READ2 that contains data on each of the conveyor
*     sections.
*
      CALL READ2
*
*     Convert X and Y coordinates from metric to imperial units (if
*     necessary).  Calculate the rise, run, slope, and length of each
*     section of belt.
*
      DO 10 J=2,NUM
        IF(UNITS.EQ.'M'.AND.J.EQ.2.OR.UNITS.EQ.'m'.AND.J.EQ.2)THEN
          X(J-1)=X(J-1)/.3048
          Y(J-1)=Y(J-1)/.3048
        ENDIF
        IF(UNITS.EQ.'M'.OR.UNITS.EQ.'m')THEN
          X(J)=X(J)/.3048
          Y(J)=Y(J)/.3048
        ENDIF
        SECTION(J)=J-1
        RISE(J)=Y(J)-Y(J-1)
        RUN(J)=ABS(X(J)-X(J-1))
        IF(RUN(J).EQ.0.0)RUN(J)=1.
        SLOPE(J)=ATAN((RISE(J)/RUN(J)))*(180./3.14159)
        RAD(J)=SLOPE(J)*(3.14159/180.)
        LENGTH(J)=RUN(J)/COS(RAD(J))
 10   CONTINUE
*
*     Write the tail station number to the output file along with the X
*     and Y coordinates of the tail station.
*
      WRITE(7,202)
      WRITE(7,203)TAIL
      WRITE(7,204)
      WRITE(7,205)
      WRITE(7,200)STAT(TAIL),X(TAIL),Y(TAIL)
*
*     Write the carry side profile to the output file.
*
      DO 20 II=TAIL+1,HEAD
        IF(ACCESS(II).EQ.'NO')THEN
        WRITE(7,201)STAT(II),X(II),Y(II),RISE(II),RUN(II),SLOPE(II),
     +LENGTH(II),LOAD(II)
        ENDIF
 20   CONTINUE
*
      WRITE(7,*)' '
```

```
*     WRITE(7,206)FF
*
 200  FORMAT(I3,3X,F8.1,2X,F8.1)
 201 FORMAT(I3,3X,F8.1,2X,F8.1,2X,F6.1,2X,F8.1,2X,F5.1,2X,F8.1,2X,F5.1)
 202  FORMAT(15X,' CARRY SIDE PROFILE ',/,16X,'------------------',/)
 203  FORMAT('Tail is station ',I3,/)
 204  FORMAT('All lengths are in feet and slope is in degrees',/)
 205  FORMAT(/,'STN',4X,'X-coord',4X,'Y-coord',3X,'RISE',4X,'RUN',4X,
     +'SLOPE',3X,'LENGTH',3X,'%LOAD')
 206  FORMAT(A1)
*
      RETURN
      END
```

```
      SUBROUTINE READ2
*
*     This subroutine reads the second data file containing information
*     on each of the sections of belt.
*
$INCLUDE:'CHAR'
$INCLUDE:'COM'
      CHARACTER*12 SECT
*
*     State the sections filename.  It is read from Unit 4.
*
      PRINT*,'Define sections filename'
      READ(*,19)SECT
 19   FORMAT(A)
      OPEN(UNIT=4,FILE=SECT,STATUS='OLD')
*
*     Read the station numbers for the tail, head, first drive, and/or
*     second drive stations (depending on whether it is a single or dual
*     drive conveyor).
*
      IF(NDU.EQ.1)READ(4,*)TAIL,HEAD,DRIVE1
      IF(NDU.EQ.2)READ(4,*)TAIL,HEAD,DRIVE1,DRIVE2
*
*     Read in station 1 and its coordinates.
*
      I=1
      READ(4,*)STAT(I),X(I),Y(I)
      I=2
*
*     Read in the rest of the stations, their coordinates, load, and
*     accessory type.
*
 10   READ(4,*,END=999)STAT(I),X(I),Y(I),LOAD(I),ACCESS(I)
      I=I+1
      GOTO 10
*
 999  CONTINUE
      NUM=I-1
*
      RETURN
      END
```

```
      SUBROUTINE SECTN1
*
*     This suboutine establishes the first estimates of the belt tensions
*     at each station, and of the effective tension, for single or dual
*     drive conveyors.
*
*     Routines called:  SUBACC
*
*     Local variables:  WMKY - material weight (kg/m)
*                       WKBY - belt weight (kg/m)
*                       KYC(100) - array containing carry belt and
*                                  material friction coefficients for each
*                                  section
*                       KXC(100) - array containing carry idler friction
*                                  coefficients for each section
*                       TXC(100) - arrays containing resistance over each
*                                  section from:
*                       TCR(100)   carry and return idler friction,
*                       TYC(100)   carry and return belt flexure,
*                       TYR(100)   material flexure,
*                       TYM(100)   material lift.
*                       TM(100)
*                       TB(100) - array containing values of belt tensions
*                                 due to belt lift
*                       CW - total wrap factor
*                       CW1,CW2 - primary and secondary wrap factor
*                       T2A - slack side tension defined by TEFF*CW
*                       ALEN - length of belt from tail pulley to last
*                              drive pylley
*                       KY1 - friction coefficient for imprint
*                       KY23 - friction coefficient for belt and material
*                              flexure
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
      REAL WMKY,WBKY,KYC(100),KXC(100)
*     REAL TXC(100),TXR(100),TYC(100),TYR(100),TYM(100),TM(100),TB(100)
      REAL CW,CW1,CW2,T2A,ALEN,KY1,KY23
*
*     Define the minimum allowable tail tension based on the specified
*     belt sag.
*
*
      TO=(SIC*(WB+WMAT))/(8.*SAGG/100.)
      KK=1
      TEF(1)=0.
*
*     Establish whether the user wants to manually set the belt
*     pretension.
*
     PRINT*,' Do you wish to manually set the pretension? (Y or N)'
      READ(*,111)PRETEN
      IF(PRETEN.EQ.'Y'.OR.PRETEN.EQ.'y')THEN
        PRINT*,' What is the pretension in LBS?'
```

```
          READ*,T2
        ENDIF
*
*       Single drive conveyor calculations.
*
        IF(NDU.EQ.1)THEN
*
*       Calculate the wrap factor for the single drive unit.  If the
*       pretension is not set by the user, then the pretension will be the
*       larger of T2A (the effective tension multiplied by the wrap factor),
*       or T2B (the minimum required tail tension less all resistances to
*       the last drive pulley).
*
         CW=1./((2.7183**(FRCO1*PWA/180.*3.14159))-1)
*
          IF(PRETEN.EQ.'Y'.OR.PRETEN.EQ.'y')GOTO 14
          T2A=TECE*CW
          DO 12 M=2,TAIL
            IF(ACCESS(M).EQ.'NO')THEN
              ALEN=LENGTH(M)+ALEN
            ENDIF
 12       CONTINUE
          T2B=TO-(Y(TAIL)-Y(DRIVE1))*WB-ABS(ALEN)*(KXR*KT+KYR*KT*WB)
*
*       Pretension is selected to be the larger of T2A and T2B.
*
         T2=AMAX1(T2A,T2B)
*
*       Initialize the belt tension at slack side of the drive pulley to be
*       T2.
*
 14       TOTC(1)=T2
*
*       Calculate the resistance over each section, establishing the total
*       belt tension and the accumulated effective tension at each station
*
          DO 10 I=2,NUM
          IF(ACCESS(I).EQ.'NO')THEN
            IF(I.GT.TAIL.AND.I.LE.HEAD)THEN
*
*       Between the tail and head stations, i.e., carry side of belt.
*
              WM(I)=A*RHO*DERAT*LOAD(I)/100.
              WMKY=WM(I)*0.4536/0.3048/100.
              WBKY=WB*0.4536/0.3048/100.
              KY1=A1*((WMKY+WBKY)**1.3)
              KY23=C4*1.0*(WMKY+WBKY)**C6/((TECE/2005.)**C7)
              KYC(I)=((KY1+KY23)/(WMKY*100.+WBKY*100.))
              IF(KYC(I).LT.0.016)KYC(I)=0.016
              KXC(I)=((0.00068*(WB+WM(I))+(AI/SIC)))
              TXC(I)=LENGTH(I)*KXC(I)*KT
              TYC(I)=LENGTH(I)*KYC(I)*KT*WB
              TYM(I)=LENGTH(I)*KYC(I)*WM(I)
              TM(I)=RISE(I)*WM(I)
              TB(I)=RISE(I)*WB
```

```
          TEF(I)=TEF(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)
          TOTC(I)=TOTC(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)+TB(I)
*
*     The return side of the belt (not including the point of slack side
*     tension, i.e., T2)
*
        ELSEIF(I.LE.TAIL.OR.(I.GT.HEAD.AND.I.NE.(DRIVE1+1)))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=TOTC(I-1)+TXR(I)+TYR(I)+TB(I)
*
*     Tension immediately after the drive pulley
*
        ELSEIF(I.EQ.(DRIVE1+1))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=T2+TXR(I)+TYR(I)+TB(I)
         ENDIF
       ELSE
*
*     If the station is an accessory, Subroutine SUBACC is called and
*     tensions from accessories are included.
*
        CALL SUBACC(I,KK,ACC)
         TACC(I)=ACC
*
*     A new T2 is established and total belt tension is the tension at the
*     previous station, plus accessory tension, i.e., belt flexure minus
*     the effective tension that is transferred to the belt by the drive
*     pulley.
*
         IF(I.EQ.(DRIVE1+1))THEN
           TOTC(I)=TOTC(I-1)+TACC(I)-TECE
           GOTO 21
         ELSE
           TOTC(I)=TOTC(I-1)+TACC(I)
         ENDIF
 21      CONTINUE
         TEF(I)=TEF(I-1)+TACC(I)
       ENDIF
 10    CONTINUE
*
*     Establish the effectvie tension to be used as a starting point for
*     Subroutine SECTN2.
*
      TEFF=TEF(NUM)
*
     ENDIF
*
*     Dual drive conveyor calculations.
*
```

```
      IF(NDU.EQ.2)THEN
*
*     Calculate the primary and secondary wrap factors, then the total
*     wrap factor.
*
        CW1=1./((2.7183**(FRCO1*PWA/180.*3.14159))-1)
        CW2=1./((2.7183**(FRCO2*SWA/180.*3.14159))-1)
        CW=(CW1*CW2)/(CW1+CW2+1)
*
*     Calculate the pretension in the same manner as described for the
*     single drive conveyor.
*
        IF(PRETEN.EQ.'Y'.OR.PRETEN.EQ.'y')GOTO 13
        ALEN=0.0
        T2A=TECE*CW
        DO 11 M=2,TAIL
          IF(ACCESS(M).EQ.'NO')THEN
            ALEN=LENGTH(M)+ALEN
          ENDIF
 11     CONTINUE
        T2B=TO-(Y(TAIL)-Y(DRIVE2))*WB-ABS(ALEN)*(KXR*KT+KYR*KT*WB)
        T2=AMAX1(T2A,T2B)
*
*     Initialize the belt tension at the slack side of second drive pulley
*     to be T2.  Calculate the primary and secondary effective tensions,
*     as well as T3.
*
13      TOTC(1)=T2
        TEP=(TECE*(1+CW2))/(1+CW1+CW2)
        TES=TECE-TEP
        T3=TES+T2
*
*     Calculate the resistances over each section, establishing the total
*     belt tension and accumulated effective tension at each station.
*
        DO 15 I=2,NUM
        IF(ACCESS(I).EQ.'NO')THEN
*
*     Between the tail and head stations, i.e., carry side of belt.
*
          IF(I.GT.TAIL.AND.I.LE.HEAD)THEN
            WM(I)=A*RHO*DERAT*LOAD(I)/100.
            WMKY=WM(I)*0.4536/0.3048/100.
            WBKY=WB*0.4536/0.3048/100.
            KY1=A1*((WMKY+WBKY)**1.3)
            KY23=C4*1.0*(WMKY+WBKY)**C6/((TECE/2005.)**C7)
            KYC(I)=((KY1+KY23)/(WMKY*100.+WBKY*100.))
            IF(KYC(I).LT.0.016)KYC(I)=0.016
            KXC(I)=((0.00068*(WB+WM(I))+(AI/SIC)))
            TXC(I)=LENGTH(I)*KXC(I)*KT
            TYC(I)=LENGTH(I)*KYC(I)*KT*WB
            TYM(I)=LENGTH(I)*KYC(I)*WM(I)
            TM(I)=RISE(I)*WM(I)
            TB(I)=RISE(I)*WB
            TEF(I)=TEF(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)
```

```
          TOTC(I)=TOTC(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)+TB(I)
*
*     Return side of belt (not including the tensions after the first and
*     second drive units
*
        ELSEIF(I.LE.TAIL.OR.(I.GT.HEAD.AND.I.NE.(DRIVE1+1)).AND.(I.GT.
     +   HEAD.AND.I.NE.(DRIVE2+1)))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=TOTC(I-1)+TXR(I)+TYR(I)+TB(I)
*
*     Tension after primary drive pulley.
*
        ELSEIF(I.EQ.(DRIVE1+1))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=T3+TXR(I)+TYR(I)+TB(I)
*
*     Tension after secondary drive pulley.
*
        ELSEIF(I.EQ.(DRIVE2+1))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=T2+TXR(I)+TYR(I)+TB(I)
         ENDIF
       ELSE
*
*     If station is an accessory, Subroutine SUBACC is called and tensions
*     from accessories are included.
*
        CALL SUBACC(I,KK,ACC)
         TACC(I)=ACC
*
*     A new T3 is established.
*
        IF(I.EQ.(DRIVE1+1))THEN
          TOTC(I)=TOTC(I-1)+TACC(I)-TEP
          GOTO 19
*
*     A new T2 is established.
*
        ELSEIF(I.EQ.(DRIVE2+1))THEN
          TOTC(I)=TOTC(I-1)+TACC(I)-TES
          GOTO 19
         ELSE
          TOTC(I)=TOTC(I-1)+TACC(I)
         ENDIF
  19     CONTINUE
         TEF(I)=TEF(I-1)+TACC(I)
```

```
      ENDIF
 15   CONTINUE
*
*    Effective tension, maximum belt tension, and T3 that are to be used
*    in Subroutine SECTN2.
*
      TEFF=TEF(NUM)
      T1=TOTC(DRIVE1)
      T3=T1+TACC(DRIVE1+1)-TEP
*
     ENDIF
*
 111 FORMAT(A1)
     RETURN
     END
```

```
      SUBROUTINE SECTN2
*
*     This subroutine uses the values calculated in Subroutine SECTN1 as
*     a starting point and re-calculates the effective tensions, T2, T3,
      and belt tensions.  All tensions are based on these values.  As the
*     tensions at each station is calculated, it is printed to the output
*     file along with unit tensions that are accumulated over that
*     particular section.  The friction factors that are used in this
*     subroutine are calculated in the same manner as those that are based
*     on the field studies by Behrends et al [1], with the tensions at the
*     end of each section being defined from the previous iteration.
*
*     Routines called:   SUBACC
*
*     Local variables:   WMKY - material weight (kg/m)
*                        WKBY - belt weight (kg/m)
*                        KYC(100) - array containing carry belt and *
                              material friction coefficients for each *
                              section
*                        KXC(100) - array containing carry idler friction
*                                   coefficients for each section
*                        TXC(100) - arrays containing resistance over each
*                                   section from:
*                        TCR(100)   carry and return idler friction,
*                        TYC(100)   carry and return belt flexure,
*                        TYR(100)   material flexure,
*                        TYM(100)   material lift.
*                        TM(100)
*                        TB(100) - array containing values of belt tensions
*                                  due to belt lift
*                        CW - total wrap factor
*                        CW1,CW2 - primary and secondary wrap factor
*                        T2A - slack side tension defined by TEFF*CW
*                        KY1 - friction coefficient for imprint
*                        KY23 - friction coefficient for belt and material
*                               flexure
*                        SAFE - safety factor in calculation of KY1 value
*                        POWRAT - power ratio between primary and secondary
*                                 drives
*                        HP - required horsepower to drive belt
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
      REAL WMKY,WBKY,KYC(100),KXC(100)
      REAL TXC(100),TXR(100),TYC(100),TYR(100),TYM(100),TM(100),TB(100)
      REAL TMAX1,TMAX2,TCWT,CW,CW1,CW2,T2A,SAFE,POWRAT,HP,KY1,KY23
*
*     Common PRISEC transfers variables between this subroutine,
*     Subroutine PRINT, and Subroutine START.
*
      COMMON /PRISEC/POWRAT,CW,CW1,CW2,HP
*
      KK=2
```

```
      WRITE(7,120)
*
*     Single drive conveyor calculations.
*
      IF(NDU.EQ.1)THEN
*
*     Recalculate the wrap factor for the single drive unit.
*
       CW=1./((2.7183**(FRCO1*PWA/180.*3.14159))-1)
*
*     Set iteration counter to 1.
*
        NN=1
 222    TEFF=TEF(NUM)
*
*     Recalculate T2A based on the effective tension calculated in
*     Subroutine SECTN1.  Select the larger of T2A and T2B.
*
       IF(PRETEN.EQ.'Y'.OR.PRETEN.EQ.'y')GOTO 333
        T2A=TEFF*CW
        T2=AMAX1(T2A,T2B)
*
*     Initialize belt tension at slack side tension, T2, as established
*     in Subroutine SECTN1.
*
 333    TOTC(1)=T2
*
        IF(NN.EQ.3)WRITE(7,90)
*
*     Recalculate the resistances over each section, establishing total
*     belt tension and accumulated effective tension at each station.
*
        DO 10 I=2,NUM
        IF(ACCESS(I).EQ.'NO')THEN
*
*     Between the tail and head stations, i.e., carry side of belt.
*
         IF(I.GT.TAIL.AND.I.LE.HEAD)THEN
            WMKY=WM(I)*0.4536/0.3048/100.
            WBKY=WB*0.4536/0.3048/100.
            SAFE=(100.*(TOTC(I)+TOTC(I-1))/(2.*BREAK))**.055
            KY1=(AA1/SAFE)*((WMKY+WBKY)**1.3)
            KY23=C4*1.0*((WMKY+WBKY)**C6)/
     +      (((TOTC(I)+TOTC(I-1))/2./2005.)**C7)
            KYC(I)=((KY1+KY23)/(WMKY*100.+WBKY*100.))
            IF(KYC(I).LT.0.016)KYC(I)=0.016
            KXC(I)=((0.00068*(WB+WM(I))+(AI/SIC)))
            TXC(I)=LENGTH(I)*KXC(I)*KT
            TYC(I)=LENGTH(I)*KYC(I)*KT*WB
            TYM(I)=LENGTH(I)*KYC(I)*WM(I)
            TM(I)=RISE(I)*WM(I)
            TB(I)=RISE(I)*WB
            TEF(I)=TEF(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)
            TOTC(I)=TOTC(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)+TB(I)
            IF(NN.EQ.3)THEN
```

```
          WRITE(7,30)STAT(I),KYC(I),KXC(I),TXC(I),TYC(I),TYM(I),TM(I),
     +    TB(I),TOTC(I)
          ENDIF
*
*     Return side of the belt (not including point of slack side tension,
*     T2).
*
        ELSEIF(I.LE.TAIL.OR.(I.GT.HEAD.AND.I.NE.(DRIVE1+1)))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=TOTC(I-1)+TXR(I)+TYR(I)+TB(I)
          IF(NN.EQ.3)THEN
          WRITE(7,40)STAT(I),KYR,KXR,TXR(I),TYR(I),TB(I),TOTC(I)
          ENDIF
*
*     Tension immediately after drive pulley.
*
        ELSEIF(I.EQ.(DRIVE1+1))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=T2+TXR(I)+TYR(I)+TB(I)
          IF(NN.EQ.3)THEN
          WRITE(7,40)STAT(I),KYR,KXR,TXR(I),TYR(I),TB(I),TOTC(I)
          ENDIF
         ENDIF
       ELSE
*
*     Re-calculate the belt flexure resistance around pulleys based on the
*     new belt tensions.
*
        IF(I.EQ.(DRIVE1+1))THEN
          IF(ACCESS(I).EQ.'PU')THEN
            CALL SUBACC(I,KK,ACC)
            TACC(I)=ACC
          ENDIF
          TOTC(I)=TOTC(I-1)+TACC(I)-TEFF
          GOTO 21
         ELSE
          IF(ACCESS(I).EQ.'PU')THEN
            CALL SUBACC(I,KK,ACC)
            TACC(I)=ACC
          ENDIF
          TOTC(I)=TOTC(I-1)+TACC(I)
         ENDIF
 21      CONTINUE
         IF(NN.EQ.3)THEN
         WRITE(7,80)ACCESS(I),STAT(I),TACC(I),TOTC(I)
         ENDIF
         TEF(I)=TEF(I-1)+TACC(I)
       ENDIF
 10    CONTINUE
```

```
*     Increase counter by 1, if the third iteration has been completed,
*     the calculations are complete.
*
        NN=NN+1
        IF(NN.LT.4)GOTO 222
*
*     Calculate the required horsepower based on the total effective
*     tension.  Establish the maximum belt tension.
*
        HP=TEFF*V/(33000.*EFF)
        T1=TOTC(DRIVE1)
        WRITE(7,60)T1,T2,TEFF,HP,TO
*
      ENDIF
*
*     Dual drive conveyor calculations
*
      IF(NDU.EQ.2)THEN
*
*     Re-calculate the primary, secondary, and total wrap factors.
*
       CW1=1./((2.7183**(FRCO1*PWA/180.*3.14159))-1)
        CW2=1./((2.7183**(FRCO2*SWA/180.*3.14159))-1)
        CW=(CW1*CW2)/(CW1+CW2+1)
*
*     Set counter to 1
*
        NN=1
 111    TEFF=TEF(NUM)
*
*     Re-calculate T2A based on effective tension calculated in Subroutine
*     SECTN1.  Pretension will be the larger of T2A and T2B.
*
        IF(PRETEN.EQ.'Y'.OR.PRETEN.EQ.'y')GOTO 444
        T2A=TEFF*CW
        T2=AMAX1(T2A,T2B)
*
*     Re-calculate the primary, secondary, and effective tensions, as well
*     as T3 and the power ratio, based on the values calculated in
*     Subroutine SECTN1.
*
444    TEP=(TEFF*(1+CW2))/(1+CW1+CW2)
        TES=TEFF-TEP
        POWRAT=TEP/TES
        T3=T1-TEP
*
*     Initialize the belt tension at slack side tension, T2, as
*     established in Subroutine SECTN1.
*
        TOTC(1)=T2
*
        IF(NN.EQ.3)WRITE(7,90)
*
*     Recalculate resistances over each section establishing total belt
*     tension accumulated effective tension at each station.
```

```
      DO 15 I=2,NUM
      IF(ACCESS(I).EQ.'NO')THEN
*
*     Between the tail and head stations, i.e., carry side of belt.
*
        IF(I.GT.TAIL.AND.I.LE.HEAD)THEN
          WMKY=WM(I)*0.4536/0.3048/100.
          WBKY=WB*0.4536/0.3048/100.
          SAFE=(100.*(TOTC(I)+TOTC(I-1))/(2.*BREAK))**.055
          KY1=(AA1/SAFE)*((WMKY+WBKY)**1.3)
          KY23=C4*1.0*((WMKY+WBKY)**C6)/
     +    (((TOTC(I)+TOTC(I-1))/2./2005.)**C7)
          KYC(I)=((KY1+KY23)/(WMKY*100.+WBKY*100.))
          IF(KYC(I).LT.0.016)KYC(I)=0.016
          KXC(I)=((0.00068*(WB+WM(I))+(AI/SIC)))
          TXC(I)=LENGTH(I)*KXC(I)*KT
          TYC(I)=LENGTH(I)*KYC(I)*KT*WB
          TYM(I)=LENGTH(I)*KYC(I)*WM(I)
          TM(I)=RISE(I)*WM(I)
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)
          TOTC(I)=TOTC(I-1)+TXC(I)+TYC(I)+TYM(I)+TM(I)+TB(I)
          IF(NN.EQ.3)THEN
          WRITE(7,30)STAT(I),KYC(I),KXC(I),TXC(I),TYC(I),TYM(I),TM(I),
     +    TB(I),TOTC(I)
          ENDIF
*
*     Return side of belt (not including tensions after first and second
*     drive units.
*
        ELSEIF(I.LE.TAIL.OR.(I.GT.HEAD.AND.I.NE.(DRIVE1+1)).AND.(I.GT.
     +  HEAD.AND.I.NE.(DRIVE2+1)))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=TOTC(I-1)+TXR(I)+TYR(I)+TB(I)
          IF(NN.EQ.3)THEN
          WRITE(7,40)STAT(I),KYR,KXR,TXR(I),TYR(I),TB(I),TOTC(I)
          ENDIF
*
*     Tension after primary drive pulley.
*
        ELSEIF(I.EQ.(DRIVE1+1))THEN
          TXR(I)=LENGTH(I)*KXR*KT
          TYR(I)=LENGTH(I)*KYR*KT*WB
          TB(I)=RISE(I)*WB
          TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
          TOTC(I)=T3+TXR(I)+TYR(I)+TB(I)
          IF(NN.EQ.3)THEN
          WRITE(7,40)STAT(I),KYR,KXR,TXR(I),TYR(I),TB(I),TOTC(I)
          ENDIF
*
*     Tension after secondary drive pulley.
*
```

```
          ELSEIF(I.EQ.(DRIVE2+1))THEN
             TXR(I)=LENGTH(I)*KXR*KT
             TYR(I)=LENGTH(I)*KYR*KT*WB
             TB(I)=RISE(I)*WB
             TEF(I)=TEF(I-1)+TXR(I)+TYR(I)
             TOTC(I)=T2+TXR(I)+TYR(I)+TB(I)
             IF(NN.EQ.3)THEN
             WRITE(7,40)STAT(I),KYR,KXR,TXR(I).TYR(I),TB(I),TOTC(I)
             ENDIF
           ENDIF
         ELSE
           IF(I.EQ.(DRIVE1+1))THEN
*
*     Re-calculate the belt flexure resistance around pulleys based on new
*     belt tensions.
*
            IF(ACCESS(I).EQ.'PU')THEN
               CALL SUBACC(I,KK,ACC)
               TACC(I)=ACC
             ENDIF
*
*     A new T3 is established
*
            TOTC(I)=TOTC(I-1)+TACC(I)-TEP
             GOTO 19
           ELSEIF(I.EQ.(DRIVE2+1))THEN
             IF(ACCESS(I).EQ.'PU')THEN
               CALL SUBACC(I,KK,ACC)
               TACC(I)=ACC
             ENDIF
*
*     A new T2 is established
*
            TOTC(I)=TOTC(I-1)+TACC(I)-TES
             GOTO 19
           ELSE
             IF(ACCESS(I).EQ.'PU')THEN
               CALL SUBACC(I,KK,ACC)
               TACC(I)=ACC
             ENDIF
             TOTC(I)=TOTC(I-1)+TACC(I)
           ENDIF
 19        CONTINUE
           IF(NN.EQ.3)THEN
           WRITE(7,80)ACCESS(I),STAT(I),TACC(I),TOTC(I)
           ENDIF
           TEF(I)=TEF(I-1)+TACC(I)
         ENDIF
 15      CONTINUE
*
*     Increase counter by 1, calculate horsepower required, establish
*     maximum belt tension and final T3.
*
         NN=NN+1
         IF(NN.LT.4)GOTO 111
```

```
        T1=TOTC(DRIVE1)
        HP=TEFF*V/(33000.*EFF)
        T3=T1-TEP+TACC(DRIVE1+1)
        WRITE(7,70)T1,TEP,T3,T2,TES,TEFF,HP,TO
*
      ENDIF
*
      WRITE(7,110)FF
      CALL PRINT
*
 30   FORMAT('CARRY  ',I3,2X,F6.4,2X,F5.3,2X,F9.1,2X,F9.1,2X,F9.1,2X,
     +F9.1,2X,F9.1,12X,F9.1)
 40   FORMAT('RET    ',I3,2X,F6.4,2X,F5.3,2X,F9.1,2X,F9.1,24X,F9.1,
     +12X,F9.1)
 60   FORMAT(/,'T1 = ',F8.0,' LBS'
     +,/,'T2 = ',F8.0,' LBS',/,'EFFECTIVE TENSION = ',F8.0,' LBS',/,
     +'REQUIRED HORSEPOWER = ',F8.0,' HP',/,'MINIMUM ALLOWABLE TO = '
     +,F8.0,' LBS')
 70   FORMAT(/,'T1 = ',F8.0,' LBS'
     +,/,'PRIMARY EFFECTIVE TENSION = ',F8.0,' LBS',/,'T3 = ',F8.0,'LBS'
     +,/,'T2 = ',F8.0,' LBS',/,'SECONDARY EFFECTIVE TENSION = ',F8.0,
     +' LBS',/,'TOTAL EFFECTIVE TENSION = ',F8.0,' LBS',/,
     +'REQUIRED HORSEPOWER = ',F8.0,' HP',/, 'MINIMUM ALLOWABLE TO = '
     + ,F8.0,' LBS')
 80   FORMAT(A2,5X,I3,70X,F9.1,3X,F9.1)
 90   FORMAT(7X,'STN',5X,'KY',6X,'KX',8X,'TX',9X,'TY',8X,'TYM',7X,'TM',
     +10X,'TB',5X,'TACC',7X,'TOTAL',/)
 110  FORMAT(A1)
 120  FORMAT(T40,'TENSIONS BY SECTION',/,T40,'-------------------',/)
*
      RETURN
      END
```

```
      SUBROUTINE PRINT
*
*     This subroutine echoes all data pertaining to the belt
*     specifications, as well as those variables in the Common block
*     PRISEC (defined in Subroutine SECTN2), to the output file.
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
      COMMON /PRISEC/POWRAT,CW,CW1,CW2,HP
*
      WRITE(7,100)
      WRITE(7,101)BTYP,QM,L,RHO,W,WMAT,WB,ALPHA,V,TEMP,SAGG,KT
      WRITE(7,102)
      WRITE(7,103)WIC,WIR,SIC,SIR,BETA
      WRITE(7,104)
      IF(NDU.EQ.1)THEN
        WRITE(7,105)NDU,PWA,DRIVE1,CW,HP,EFF
      ELSE
        WRITE(7,106)NDU,PWA,DRIVE1,CW1,DRIVE2,SWA,POWRAT,CW2,HP,CW,EFF
      ENDIF
 100  FORMAT(T35,'BELT PARAMFTERS',/,T35,'---------------',/)
 101  FORMAT('BELT TYPE:',T28,A8,T45,'CONVEYOR CAPACITY (TPH):',T75,
     +F9.2,/,'BELT LENGTH (FT):',T28,F9.2,T45,'MAT. DENSITY (PCF):'
     +,T75,F9.2,/, 'BELT WIDTH (IN):',T28,F9.2,T45,'MAT. WEIGHT (PPF):'
     +,T75,F9.2,/,'BELT WEIGHT (PPF):',T28,F9.2,T45,'MAT. SURCHARGE '
     +'(DEG):',T75,F9.2,/,'BELT VELOCITY (FPM);',T28,F9.2,T45,'AMBIENT'
     +' TEMPERATURE (F):',T75,F9.2,/,'SAG PERCENT:',T28,F9.2,T45,
     +'KT:',T75,F9.2)
 102  FORMAT(//,T35,'IDLER PARAMETERS',/,T35,'----------------',/)
 103  FORMAT('WEIGHT OF CARRY IDLER (LBS):',T28,F9.2,T45,'WEIGHT OF'
     +' RETURN IDLER (LBS):',T75,F9.2,/,'CARRY IDLER SPACING (FT):',
     +T28,F9.2,T45,'RETURN IDLER SPACING (FT):',T75,F9.2,/,
     +'CARRY TROUGH ANGLE (DEG):',T28,F9.2)
 104  FORMAT(//,T35,'DRIVE PARAMETERS',/,T35,'----------------'/)
 105  FORMAT('NUMBER OF DRIVES:',T28,I7,T45,'WRAP ANGLE (DEG):',T75,F9.2
     +,/,'DRIVE STATION:',T28,I7,T45,'WRAP FACTOR:',T75,F9.2,/,
     +'REQUIRED HORSEPOWER:',T28,F9.2,T45,'EFFICIENCY:',T75,F9.2)
 106  FORMAT('NUMBER OF DRIVES:',T28,I7,T45,'PRIMARY WRAP ANGLE (DEG):',
     +T75,F9.2,/,'DRIVE STATION 1:',T28,I7,T45,'PRIMARY WRAP FACTOR:',
     +T75,F9.2,/,'DRIVE STATION 2:',T28,I7,T45,'SECONDARY WRAP ANGLE'
     +' (DEG):',T75,F9.2,/,'POWER RATIO:',T28,F9.2,T45,'SECONDARY WRAP'
     +' FACTOR:',T75,F9.2,/,'REQUIRED HORSEPOWER:',T28,F9.2,
     +T45,'TOTAL WRAP FACTOR:',T75,F9.2,/,'EFFICIENCY:',T28,F9.2)
      RETURN
      END
```

```
      SUBROUTINE SUBACC(I,KK,ACC)
*
*     This subroutine calculates the resistance of each accessory.  All
*     are initially calculated form subroutine SECTN1.   Because belt
*     tensions are re-calculated in Subroutine SECTN2, the belt flexure
*     resistance around pulleys is re-calculated in Subroutine SECTN2 as
*     this resistance is dependant upon belt tensions.
*
*     Local variables:  ACC - accessory resistance returned to subroutine
*                             SECTN1 or SECTN2
*                       AVTEN - average belt tension over a section
*                       WID - belt width
*                       PULDIM(100) - diameter of a pulley at a particular
*                                     station
*                       SCRWID - scraper width
*                       SBLEN - skirtboard length
*                       SBCS - skirtboard friction factor
*                       SBHS - height of material on skirtboards
*                       PBRES(100) - pulley bearing resistance
*                       PLOW - 7 character variable denoting type of plow
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
      REAL ACC,AVTEN,WID,PULDIM(100),SCRWID,SBLEN,SBCS,SBHS
      REAL PBRES(100)
      CHARACTER*7 PLOW
*
*     If this subroutine is being called from Subroutine SECTN2, (i.e.,
*     calculation of belt flexure resistance around pulleys only) then go
*     to 111.
*
      IF(KK.GE.2.AND.ACCESS(I).EQ.'PU')GOTO 11
*
*     Initial calculations for belt flexure resistance around pulleys.
*     This is called from Subroutine SECTN1.
*
      IF(ACCESS(I).EQ.'PU')THEN
        WRITE(*,100)I
 100    FORMAT(' PULLEY STATION ',I3,/)
        PRINT*,'What is the pulley diameter in meters?'
        READ*,PULDIM(I)
        IF(UNITS.EQ.'I'.OR.UNITS.EQ.'i')THICK=THICK*2.54E-02
        WID=W*2.54E-02
        AVTEN=TOTC(I-1)*4.448
        ACC=(12.*WID*(200.+0.01*AVTEN/WID)*(THICK/PULDIM(I)))*KT
        ACC=ACC/4.448
        IF(UNITS.EQ.'I'.OR.UNITS.EQ.'i')THICK=THICK/2.54E-02
        PRINT*,'What is the pulley bearing resistance in lbs?'
        PRINT*,'Tight side 200 lbs, Slack side 150 lbs, Other 100 lbs'
        READ*,PBRES(I)
        ACC=ACC+PBRES(I)
        RETURN
*
```

```
*     Calculate belt cleaner resistance
*
     ELSEIF(ACCESS(I).EQ.'BC')THEN
        WRITE(*,101)I
 101    FORMAT(' BELT CLEANER STATION ',I3,/)
        PRINT*,'What is the scraper width in inches?'
        READ*,SCRWID
        ACC=SCRWID*5.0
        RETURN
*
*     Calculate material accleration resistance
*
     ELSEIF(ACCESS(I).EQ.'MA')THEN
        WRITE(*,102)I
 102    FORMAT(' MATERIAL ACCELERATION STATION ',I3,/)
        ACC=2.8755E-04*QM*(LOAD(I)/100.)*(V-VO)
        RETURN
*
*     Calculate skirtboard resistance
*
     ELSEIF(ACCESS(I).EQ.'SB')THEN
        WRITE(*,103)I
 103    FORMAT(' SKIRTBOARD STATION ',I3,/)
        PRINT*,'What is the skirtboard length in feet?'
        READ*,SBLEN
        PRINT*,'What is the material height on the skirts in inches?'
        READ*,SBHS
        PRINT*,'What is the skirtboard friction factor? (pg 91  EMA)'
        READ*,SBCS
        ACC=SBLEN*(SBCS*SBHS**2.+6.)
        RETURN
*
*     Calculate resistance from plows
*
     ELSEIF(ACCESS(I).EQ.'PL')THEN
        WRITE(*,104)I
 104    FORMAT(' PLOW STATION ',I3,/)
        PRINT*,'Is the plow a full vee or partial vee? (pg 90 CEMA)'
        READ(*,99)PLOW
 99     FORMAT(A7)
        IF(PLOW.EQ.'FULL'.OR.PLOW.EQ.'full')THEN
          ACC=W*5.0
          RETURN
        ELSEIF(PLOW.EQ.'PARTIAL'.OR.PLOW.EQ.'partial')THEN
          ACC=W*3.0
          RETURN
        ENDIF
      ENDIF
*
*     Re-calculation of belt flexure resistance around pulleys (called
*     from Subroutine SECTN2).
*
 11     AVTEN=TOTC(I-1)*4.448
        IF(UNITS.EQ.'I'.OR.UNITS.EQ.'i')THICK=THICK*2.54E-02
        ACC=(12.*WID*(200.+0.01*AVTEN/WID)*(THICK/PULDIM(I)))*KT
```

```
  IF(UNITS.EQ.'I'.OR.UNITS.EQ.'i')THICK=THICK/2.54E-02
  ACC=ACC/4.448
  ACC=ACC+PBRES(I)
  RETURN
*
      RETURN
      END
```

```
      SUBROUTINE PLOTT
*
*     This subroutine organizes the tension values at each section along
*     the entire length of the belt, allowing Subroutine GRAPH to plot the
*     points.
*
*     Routines called:  START
*                       DECEL
*
*     Local variables:  DIST(100) - distance along belt that each station
*                                   is from the last drive pulley
*                       TEN1(100) - belt resistances at a station
*                       TEN2(100) - total resistance at a station
*                       TEN3(100) - material resistances at a station
*                       PLAWT - character inputs at prompt (Y or N) for
*                       SCREEN  a plot and acceleration calculations
*                       ACCALC
*                       PLODAT - user-defined filename for plot data
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
      REAL DIST(100),TEN1(100),TEN2(100),TEN3(100)
      CHARACTER*1 PLAWT,SCREEN,ACCALC
      CHARACTER*12 PLODAT
*
*     User-defined filename for the plot data, read from Unit 9.
*
      PRINT*,'DEFINE THE DESIRED PLOTTING DATA FILENAME'
      READ(*,50)PLODAT
      OPEN(UNIT=9,FILE=PLODAT)
*
*     Initializing the belt resistances and total resistances starting
*     point at slack side tension and write to plot data file.
*
      TEN1(1)=(TOTC(1))
      TEN2(1)=(TOTC(1))
      WRITE(9,*)DIST(1),TOTC(1)
*
*     Calculate the belt tension at each station along the belt.
*
      DO 10 I=2,NUM
      TEN1(I)=(TEN1(I-1)+TXC(I)+TXR(I)+TYC(I)+TYR(I)+TB(I)+TACC(I))
      TEN3(I)=(TEN3(I-1)+TYM(I)+TM(I))
      IF(I.GT.TAIL.AND.I.LE.HEAD)THEN
        TEN2(I)=(TEN1(I)+TEN3(I))
      ELSE
        TEN2(I)=(TEN1(I))
      ENDIF
      IF(NDU.EQ.2)THEN
        IF(I.EQ.DRIVE1+1)THEN
          TEN1(I)=T3
          TEN2(I)=(TEN1(I))
        ELSEIF(I.EQ.DRIVE2+1)THEN
```

```
        TEN1(I)=(TOTC(DRIVE2+1))
        TEN2(I)=(TEN1(I))
      ENDIF
    ENDIF
    IF(NDU.EQ.1)THEN
      IF(I.EQ.DRIVE1+1)THEN
        TEN1(I)=T2
        TEN2(I)=(TEN1(I))
      ENDIF
    ENDIF
 10 CONTINUE
*
*   Calculate the corresponding distance of belt between the station
*   under investigation and the tail station.
*
    DIST(TAIL)=0.0
    X(TAIL)=X(TAIL)
    DO 20 J=TAIL+1,NUM
    PRINT*,X(J)
    DIST(J)=(DIST(J-1)+ABS(X(J)-X(J-1)))
 20 CONTINUE
    DIST(1)=DIST(NUM)
    X(1)=X(NUM)
    DO 30 K=2,TAIL-1
    PRINT*,X(K)
    DIST(K)=(DIST(K-1)+ABS(X(K)-X(K-1)))
 30 CONTINUE
*
*   Convert from imperial to metric units.
*
   DO 40 J=TAIL,NUM
    DIST(J)=DIST(J)*.3048
    TEN1(J)=TEN1(J)*4.45/1000
    TEN2(J)=TEN2(J)*4.45/1000
    TOTC(J)=TOTC(J)*4.45/1000
    WRITE(9,99)DIST(J),TEN1(J),TEN2(J),TOTC(J)
 40 CONTINUE
    DO 60 K=1,TAIL-1
    DIST(K)=DIST(K)*.3048
    TEN1(K)=TEN1(K)*4.45/1000
    TEN2(K)=TEN2(K)*4.45/1000
    TOTC(K)=TOTC(K)*4.45/1000
    WRITE(9,99)DIST(K),TEN1(K),TEN2(K),TOTC(K)
 60 CONTINUE
*
*   Asking the user if a plot is desired.  If yes, then Subroutine GRAPH
*   is called.
*
   WRITE(*,100)
    READ(*,102)PLAWT
    IF(PLAWT.EQ.'Y'.OR.PLAWT.EQ.'y')THEN
    CALL GRAPH
    ELSE
    GOTO 70
    ENDIF
```

```
*     Asking the user if a plot is desired for a second and final time.
*     If yes, then Subroutine GRAPH is called again.  This may allow the
*     user to 'experiment' with the graph layouts.
*
      WRITE(*,101)
      READ(*,102)SCREEN
      IF(SCREEN.EQ.'Y'.OR.SCREEN.EQ.'y')THEN
      CALL GRAPH
      ELSE
      GOTO 70
      ENDIF
*
*     Asking the user if they wish acceleration and deceleration
*     calculations.  If yes, then Subroutines START and DECEL are called.
*
 70   WRITE(*,103)
      READ(*,102)ACCALC
      IF(ACCALC.EQ.'Y'.OR.ACCALC.EQ.'y')THEN
      CALL START
      CALL DECEL
      GOTO 70
      ELSE
      STOP
      ENDIF
*
 50   FORMAT(A)
 99   FORMAT(F7.0,3F9.0)
 100  FORMAT(' Do you require a plot?  (Y or N)')
 101  FORMAT(' Call plot routine again? (Y or N)')
 102  FORMAT(A1)
 103  FORMAT(' Do you require acceleration and deceleration',
     +' calculations (Y or N)')
      RETURN
      END
```

```
      SUBROUTINE GRAPH
*
*     This subroutine uses the software package PLOT88 to plot the belt
*     tensions that have been tabulated by Subroutine PLOTT, starting at
*     the tail pulley and working around the entire belt.  Two tensions
*     are plotted: total tension and belt resistance tensions.  The
*     difference between the two on the carry side of the belt is the
*     material resistance tension.  The return side has only belt
*     resistances.  All subroutines called within GRAPH are from PLOT88.
*
*     Local variables:  FACT - factor to alter plot size
*                       XPOINT(100) - distance along the belt of a
*                                     particular station from the last
*                                     drive pulley
*                       TEN1(100) - belt resistances at a station
*                       TEN2(100) - total resistance at a station
*                       SORTX(100) - temporary file for bubble sort of X
*                       SORTY(100)   and Y coordinates
*                       INCX - characters inputted at prompt (Y or N) to
*                       INCY   increase plot axes
*                       TITLE - title of plot
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
      REAL FACT,XPOINT(100),TEN1(100),TEN2(100)
      REAL SORTX(100),SORTY(100)
      CHARACTER*1 INCX,INCY
      CHARACTER*30 TITLE
*
*     Rewind Unit 9, the file containing the plot data.  Read all of the
*     data and assign the total tension and distance to temporary array
*     to perform a bubble sort to later define the X and Y limits of the
*     graph.
*
      REWIND 9
      READ(9,*)XPOINT(1),TOTC(1)
      TEN1(1)=TOTC(1)
      TEN2(1)=TOTC(1)
      DO 10 I=1,NUM
        READ(9,*)XPOINT(I),TEN1(I),TEN2(I),TOTC(I)
        SORTY(I)=TOTC(I)
        SORTX(I)=XPOINT(I)
 10   CONTINUE
      SORTX(1)=0.0
      SORTY(1)=0.0
*
*     Perform bubble sort
*
      DO 20 J=1,NUM-1
        IF(SORTX(J).GT.SORTX(J+1))THEN
          TEMPX=SORTX(J)
          SORTX(J)=SORTX(J+1)
          SORTX(J+1)=TEMPX
```

```
        ENDIF
 20   CONTINUE
      DO 30 K=1,NUM-1
        IF(SORTY(K).GT.SORTY(K+1))THEN
          TEMPY=SORTY(K)
          SORTY(K)=SORTY(K+1)
          SORTY(K+1)=TEMPY
        ENDIF
 30   CONTINUE
*
*     Define minimum and maximum X and Y coordinates for the graph
*
      XMIN=SORTX(1)
      XMAX=SORTX(NUM)
      YMIN=SORTY(1)
      YMAX=SORTY(NUM)
*
*     Define where the graph is to be viewed/printed.
*
      PRINT*,'Define the printer IOPORT and MODEL numbers'
      READ*,IOPORT,MODEL
      CALL PLOTS(0,IOPORT,MODEL)
*
*     Define the size of the graph as a percentage of a full graph on the
*     device that it is being viewed/printed.
*
      PRINT*,'Define the FACTOR size'
      READ*,FACT
      CALL FACTOR(FACT)
*
*     Initialize the starting point of the graph.
*
      CALL PLOT(1.0,1.0,-3)
*
*     Defining the limits of the graph.  The X and Y axes may by
*     increased or decreased, depending on what the user requires.
*
      XPOINT(NUM+1)=0.0
      XPOINT(NUM+2)=(XMAX-XMIN)/9.5
      TOTC(NUM+1)=0.0
      TOTC(NUM+2)=(YMAX-YMIN)/6.5
      WRITE(*,111)XPOINT(NUM+2)
      READ(*,110)INCX
      IF(INCX.EQ.'Y'.OR.INCX.EQ.'y')THEN
        PRINT*,'What is the new X increment value?'
        READ*,XPOINT(NUM+2)
      ELSE
        XPOINT(NUM+2)=XPOINT(NUM+2)
      ENDIF
*
      WRITE(*,112)TOTC(NUM+2)
      READ(*,110)INCY
      IF(INCY.EQ.'Y'.OR.INCX.EQ.'y')THEN
        PRINT*,'What is the new Y increment value?'
        READ*,TOTC(NUM+2)
```

```
      ELSE
        TOTC(NUM+2)=TOTC(NUM+2)
      ENDIF
      TEN1(NUM+1)=0.0
      TEN2(NUM+1)=0.0
      TEN1(NUM+2)=TOTC(NUM+2)
      TEN2(NUM+2)=TOTC(NUM+2)
*
      CALL STAXIS(.16,.20,.16,.1,-1)
*
*     Placing labels on each axis.
*
      CALL AXIS(0.,0.,'DIST FROM TAIL (m)',-19,9.5,.0
     +,XPOINT(NUM+1),XPOINT(NUM+2))
      CALL AXIS(0.,0.,'TENSION (kN)',13,6.5,90.,TOTC(NUM+1)
     +,TOTC(NUM+2))
*
*     Plotting the graph.
*
      CALL STLINE(1,.16,0.)
      CALL LINE(XPOINT,TEN1,NUM,1,1,1)
      CALL STLINE(1,.16,0.)
      CALL LINE(XPOINT,TOTC,NUM,1,1,2)
*
*     User-defined plot title, and the title's positioning.
*
      WRITE(*,*)' What is the plot title? (30 characters max)'
      READ(*,113)TITLE
      WRITE(*,*)' How far from the Y-axis do you want the title? (in)'
      READ*,TDIST
*
*     Placing symbols and legeng on the plot
*
      CALL SYMBOL(TDIST,6.5,.22,TITLE,0.,30)
      CALL SYMBOL(TDIST+.3,6.0,.16,CHAR(1),0.,-1)
      CALL SYMBOL(999.0,5.94,0.16,'  = BELT RESISTANCES',0.,24)
      CALL SYMBOL(TDIST+.3,5.7,0.16,CHAR(2),0.,-1)
      CALL SYMBOL(999.0,5.64,0.16,'  = MATERIAL + BELT RESIST.',0.,27)
      CALL PLOT(0.0,0.0,999)
 110  FORMAT(A1)
 113  FORMAT(A30)
 111  FORMAT(' MINIMUM X-AXIS INCREMENT = ',F8.0,/,'     Do you wish to'
     +' increase the minimum increment? (Y or N)')
 112  FORMAT(' MINIMUM Y-AXIS INCREMENT = ',F8.0,/,'     Do you wish to'
     +' increase the minimum increment? (Y or N)')
      RETURN
      END
```

```
      SUBROUTINE START
*
*     This subroutine calculates starting belt tensions based on user
*     inputs, and uses this to determine the minimum allowable
*     acceleration time, and minimum concave curve radii.
*
*     Local variables:  COUWK2 - WK2 values for couplings, drive pulleys,
*                       PULWK2         reducers, motors, and the total WK2.
*                       REDWK2
*                       MOTWK2
*                       TOTWK2
*                       DRIVWT - equivalent weight of drive system
*                       PERCOP - maximum allowable operating and starting
*                       PERCST   tensions as percent of belt breaking
*                                strength
*                       OPTENS - maximum allowable operating and starting
*                       STTENS   tensions
*                       STT2 - maximum starting slack side tension
*                       BELTC - total weight of carry and return side belt
*                       BELTR   strands
*                       MATWT - material weight
*                       TOTPWT - total weight of non-driven pulleys
*                       PULWT - weight of one non-driven pulley
*                       SYSTWT - total system weight
*                       MOTSIZ - motor size
*                       STAV1 - available extra tension induced on the
*                               belt to accelerate it
*                       TIMEA1 - time to accelerate belt based on belt
*                                strength limitations
*                       PERCTQ - percentage of motor torque used during
*                                start-up
*                       MAXTEN - maximum tension induced on the belt *
*                                during start-up
*                       TENAV2 - available accelerating tension from motor
*                                torque (MAXTEN - TEFF)
*                       TIMEA2 - time to accelerate belt based on motor
*                                torque limitations
*                       RAD1 - concave curve radii for loaded and unloaded
*                       RAD2   belt
*                       MOTNUM - number of motors
*                       POINT - station at which curve analysis takes *
                          place
*                       LIFT - user-inputted (Y or N) if a lift check is
*                              required
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
*     Common blocks PRISEC (defined in Subroutine SECTN1), and STDEC
*     which passes variables between this subroutine and Subroutine DECEL.
*
      COMMON /PRISEC/POWRAT,CW,CW1,CW2,HP
      COMMON /STDEC/SYSTWT,DRIVWT
      REAL COUWK2,PULWK2,TOTWK2,DRIVWT,PERCOP,PERCST,OPTENS,STTENS
```

```
      REAL STT2,BELTC,BELTR,MATWT,TOTPWT,PULWT,SYSTWT,MOTSIZ
      REAL STAV1,TIMEA1,PERCTQ,MAXTEN,REQT2,REQT1,TENAV2,TIMEA2,RAD1
      REAL RAD2,MOTWK2,REDWK2
      INTEGER MOTNUM,POINT
      CHARACTER*1 LIFT
*
*     Initialize starting values
*
      BELTC=0.
      BELTR=0.
      MATWT=0.
      TOTPWT=0.
*
      WRITE(7,106)
*
*     Determine total belt weight and material weight to be accelerated.
*
      DO 10 I=2,NUM
        IF(I.GT.TAIL.AND.I.LE.HEAD)THEN
          BELTC=BELTC+LENGTH(I)
          MATWT=MATWT+(WM(I)*LENGTH(I))
        ELSE
          BELTR=BELTR+LENGTH(I)
        ENDIF
*
        IF(I.EQ.DRIVE1+1)GOTO 10
        IF(I.EQ.DRIVE2+1)GOTO 10
*
*     Calculate the total pulley weight to be accelerated.
*
       IF(ACCESS(I).EQ.'PU')THEN
          WRITE(*,101)I
 101      FORMAT(' What is the weight of the pulley at station ',I3)
          READ*,PULWT
          TOTPWT=TOTPWT+PULWT
        ENDIF
*
 10   CONTINUE
*
*     Total system weight to be accelerated, including belt, material,
*     pulleys, and idlers.
*
      SYSTWT=BELTC*WB+BELTR*WB+(BELTC*WIC/SIC)+(BELTR*WIR/SIR)+
     +MATWT+(TOTPWT*2./3.)
*
*     User inputs for the motor size and quantity, as well as, WK2 of the
*     motor and associated parts.
*
      WRITE(*,102)HP
 102  FORMAT(2X,'The required horsepower is ',F5.0,/,2X,
     +'What motor size and how many will be used?',/)
      READ*,MOTSIZ,MOTNUM
      PRINT*,' '
      PRINT*,' What is the WK2 of the motor in lb-ft2 ?'
      READ*,MOTWK2
```

```
      PRINT*,' '
      PRINT*,' What is the WK2 of the reducer in lb-ft2 ?'
      PRINT*,' Usually 20% of the motor WK2)'
      READ*,REDWK2
      PRINT*,' '
      PRINT*,' What is the WK2 of the coupling in lb-ft2 ?'
      READ*,COUWK2
      PRINT*,' '
      PRINT*,' What is the WK2 of the driven pulley in lb-ft2 ?'
      READ*,PULWK2
      PRINT*,' '
*
*     Total WK2 of the motors, as well as the equivalent weight of the
*     drive pulleys.
*
      TOTWK2=(MOTWK2+REDWK2+COUWK2)*MOTNUM+(PULWK2*NDU)
      DRIVWT=TOTWK2*((2*3.14159*RPM)/V)**2.
      PERCWT=SYSTWT/(SYSTWT+DRIVWT)
*
      PRINT*,' What is the maximum OPERATING tension as a percentage'
      PRINT*,' of the belt breaking strength?'
      READ*,PERCOP
      PRINT*,' '
      PRINT*,' What is the maximum STARTING tension as a percentage'
      PRINT*,' of the belt breaking strength?'
      READ*,PERCST
      PRINT*,' '
*
*     Calculating the maximum allowable operating belt tension, maximum
*     allowable starting belt tension, maximum starting slack side
*     tension, allowable extra accelerating tension due to belt strength,
*     and the minimum allowable acceleration time due to belt strength
*     limitations.
*
      OPTENS=PERCOP*BREAK/100.
      STTENS=PERCST*BREAK/100.
      STT2=STTENS*CW
      STAV1=STTENS-TEFF-STT2
      TIMEA1=((SYSTWT+DRIVWT)/(STAV1*32.2))*((V-VO)/60.)
*
*     Determining the maximum tension that can be applied to the belt due
*     to motor torque.
*
      PRINT*,' What percentage of the motor torque will be used'
      PRINT*,' during start-up?'
      READ*,PERCTQ
      PRINT*,' '
      MAXTEN=33000.*MOT[illegible]Z*MOTNUM*PERCTQ*EFF/(V*100.)
*
*     Allowable extra accelerating tension due to motor torque, and
*     minimum allowable acceleration time due to motor torque
*     limitations.
*
      TENAV2=MAXTEN-TEFF
      TIMEA2=((V-VO)/60.)/(TENAV2/((SYSTWT+DRIVWT)/32.2))
```

```
      IF(TIMEA1.LE.0..OR.TIMEA1.GT.999.)WRITE(7,109)
      IF(TIMEA2.LE.0..OR.TIMEA2.GT.999.)WRITE(7,110)
      IF(TIMEA1.LE.0..AND.TIMEA2.LE.0..OR.TIMEA1.LE.0..AND.TIMEA2.GT.
     +999..OR.TIMEA1.GT.999..AND.TIMEA2.GT.999..OR.TIMEA1.GT.999..AND.
     +TIMEA2.LE.0.)THEN
        WRITE(7,103)MOTSIZ,MOTNUM,BREAK,OPTENS,STTENS,TIMEA1,TIMEA2
        WRITE(7,111)
        GOTO 20
      ENDIF
      WRITE(7,103)MOTSIZ,MOTNUM,BREAK,OPTENS,STTENS,TIMEA1,TIMEA2
      IF(TIMEA1.GT.TIMEA2)THEN
        IF(TIMEA2.GT.0.)WRITE(7,104)TIMEA1
      ELSE
        IF(TIMEA1.GT.0.)WRITE(7,105)TIMEA2
      ENDIF
*
*     Checking for belt lifting at concave curves.
*
      PRINT*,'Do you require a check for belt lifting during start-up?'
      PRINT*,'Y or N'
      READ(*,108)LIFT
      NN=1
      IF(LIFT.EQ.'Y'.OR.LIFT.EQ.'y')THEN
        PRINT*,'At which station do you wish to check for lifting?'
        READ*,POINT
 30     IF(Y(POINT+1).LE.Y(POINT))THEN
          WRITE(*,112)POINT
          PRINT*,'At which station do you wish to check for lifting?'
          READ*,POINT
          IF(NN.GT.3)GOTO 20
          NN=NN+1
          GOTO 30
        ENDIF

*
*     Determining the maximum tension at concave point based on minimum
*     allowable accelerating time.
*
        IF(TIMEA1.GT.TIMEA2)THEN
          TOTAL=STTENS+TOTC(POINT)
        ELSE
          TOTAL=MAXTEN+TOTC(POINT)
        ENDIF
*
*     Calculating the curve radius with material on the belt and with no
*     material on the belt.
*
       RAD1=1.11*TOTAL/(WB+WM(POINT))
        RAD2=1.11*TOTAL/WB
        WRITE(7,107)POINT,TOTAL,RAD1,RAD2
      ENDIF
 20   CONTINUE
*
 103  FORMAT(//,'Motor HP ',T25,F5.0,/,'Number of motors ',T25,I2,/,
     +'Belt breaking strength ',T25,F8.0,' lbs',/,
```

```
    +'Max operating tension', T25,F8.0,' lbs',/,
    +'Max starting tension ',T25,F8.0,' lbs',/,
    +'Acceleration time (belt strength limitations) ',T55,F6.2,' sec',
    +/,'Acceleration time (torque limitations) ',T55,F6.2,' sec',/)
104  FORMAT('Minimum allowable acceleration time is ',F6.2,
    +' sec  due to BELT STRENGTH limitations',/)
105  FORMAT('Minimum allowable acceleration time is ',F6.2,
    +' sec  due to TORQUE limitations',/)
106  FORMAT(//,15X,'ACCELERATION CALCULATIONS',/,15X,
    +'-------------------------')
107  FORMAT('The total tension at station ',I2,' is ',F8.0,' lbs',/,
    +'This requires a curve RADIUS of ',F6.0,' ft',' with material on'
    +' the belt',/,'This requires a curve RADIUS of ',F6.0,' ft',
    +' with an empty belt',/)
108  FORMAT(A)
109  FORMAT('The belt is not strong enough to accomodate acceleration'
    +,/)
110  FORMAT('The applied motor torque will not accelerate the belt',/)
111  FORMAT('Both the belt strength and motor torque are not ',/,
    +'sufficient to accelerate the belt',/)
112  FORMAT(' There is no concave curve at station ',I3)
*
     RETURN
     END
```

```
      SUBROUTINE DECEL
*
*     This subroutine calculates the deceleration time when motors are
*     used to stop the conveyor.
*
*     Local variables:  KINENG - kinetic energy of moving belt
*                       RETARD - retarding force of motors
*                       AVEVEL - average belt velocity
*                       DECELT - time to stop belt
*                       HOPWT - weight of material discharged to the *
                          hopper during shut-down
*                       HOPVOL - required hopper volume based on HOPWT
*
*
$INCLUDE:'COM'
$INCLUDE:'CHAR'
*
      COMMON /PRISEC/POWRAT,CW,CW1,CW2,HP
      COMMON /STDEC/SYSTWT,DRIVWT
*
      REAL KINENG,RETARD,AVEVEL,DECELT,HOPWT,HOPVOL
*
*     Establishing the kinetic energy of the moving conveyor, the
*     retarding force of the motors, and the average velocity of the
*     conveyor.
*
      KINENG=((SYSTWT+DRIVWT)/32.2)*(((V/60.)**2.)/2.)
      RETARD=HP*EFF*33000./V
      AVEVEL=V/2.
*
*     Calculating the deceleration time of the conveyor, the weight of
*     material that goes into the hopper during shut-down, and the
*     required hopper volume.
*
      DECELT=(KINENG/(AVEVEL*RETARD))*60.
      HOPWT=AVEVEL*DECELT*WMAT/60.
      HOPVOL=HOPWT/RHO
*
      PRINT*,KINENG,RETARD,AVEVEL
      WRITE(7,101)
      WRITE(7,102)DECELT,HOPWT,HOPVOL
 101  FORMAT(/,15X,'DECELERATION CALCULATIONS',/,15X,
     +'-------------------------',//)
 102  FORMAT('Deceleration time ',T25,F7.2,T34,'sec',/,
     +'Weight in hopper ',T25,F9.0,T34,'lbs  Assumes belt is loaded',/,
     +'Volume in hopper ',T25,F7.0,T34,'cu ft',/)
*
      RETURN
      END
```