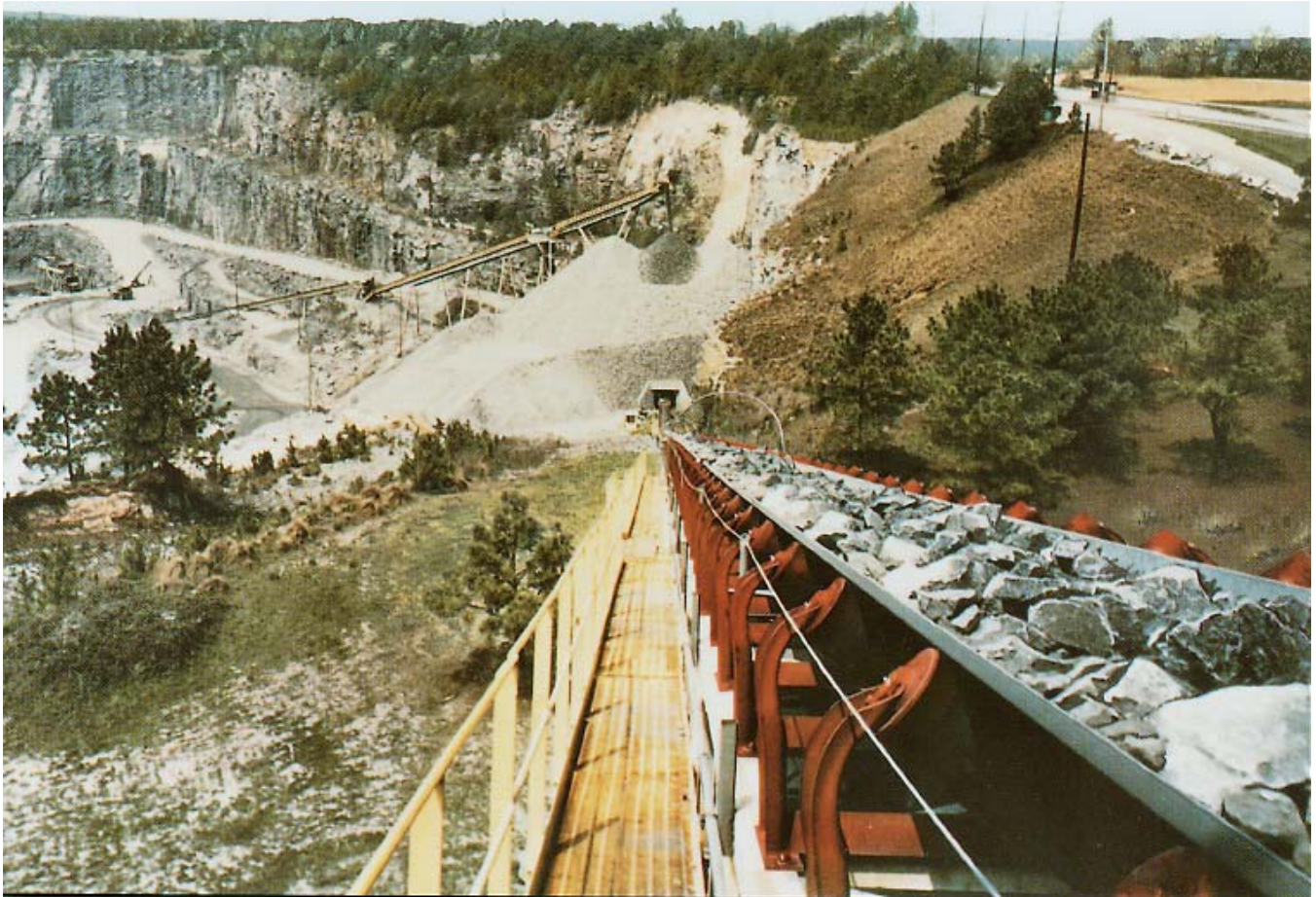




*Prepared by the Engineering Conference of the
Conveyor Equipment Manufacturers Association*



Belt Conveyors for Bulk Materials

**FIFTH EDITION
PDF Version
July, 2002**

Published by the Conveyor Equipment Manufacturers Association

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Printed in the United States of America

Page layout and editing by The Scribe and Word Warriors.

Printing (last digit): 9 8 7 6 5 4 3 2

Library of Congress Cataloging in Publication Data

Conveyor Equipment Manufacturers Association.

Engineering Conference.

Belt conveyors for bulk materials.

Includes index.

1. Belt conveyors. 2. Bulk solids handling.

1. Title.

TJ1390.C65 1979 621.8'675 78-31987

ISBN 1-891171-49-6

PDF Version of

ISBN 1-891171-18-6

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Preface

After over three decades of widespread use throughout the world, *Belt Conveyors for Bulk Materials* has established its place as the premier manual in its field.

Based on the popularity of the first four editions, the Board of Directors of the Conveyor Equipment Manufacturers Association (CEMA) authorized preparation of this Fifth Edition.

Leading belt conveyor executives and engineers of CEMA member companies assisted in the development of new and updated information for the new edition through the Association's Bulk Handling Systems and Equipment Section, Bulk Handling Components Section, and Engineering Conference.

Throughout the history of *Belt Conveyors for Bulk Materials*, the primary objective has remained unchanged: To provide basic engineering data for the design of belt conveyors in a book written for technically qualified people who may not be totally familiar with belt conveyor design.

The Fifth Edition retains all of the most desirable features of book design, information presentation, and illustration incorporated in the previous editions.

The Table of Contents enables the reader to review quickly the entire book and readily find the location of every subject covered.

Special efforts have been made to consolidate into one location all of the pertinent information on each subject. For instance, all information on belt tension, power, and drive engineering has been gathered together in Chapter 6.

Since reader experience with the first four editions favored numbering all illustrative material—photographs, diagrams, tables—this system has been retained. Each figure and table is identified first by the chapter in which it appears, then by its sequence within that chapter. In the case of figures, chapter and sequence numbers are set off by a decimal point; with tables, by a hyphen. Thus, Figure 3.2 is the second figure in Chapter 3, and Table 6-2 is the second table in Chapter 6.

The use of belt conveyors as an important and broadly accepted means of long distance transport of bulk materials prompted the development of Chapter 1. Chapters 7 and 13 are new, and Chapter 7 incorporates the current Rubber Manufacturers Association material and recommendations. Chapter 13 has been completely rewritten and provides conveyor drive evaluation criteria, an overview of conveyor drive systems, and a section on drive controllers, control algorithms, protection controls, interlocking, and control apparatus.

Acknowledgments

The Conveyor Equipment Manufacturers Association is indebted to the members of its Bulk Handling Components and Systems Section, Engineering Conference, Bulk Conveyor Engineering Committee, and the many other individuals who contributed their time and effort to the study and compilation of the data in this book; to the individual member companies who made available the time and talents of their engineers, draftsmen, and publication specialists; to the Rubber Manufacturers Association, Inc.; the Mechanical Power Transmission Association; and other technical associations and professional societies for their assistance in making this book a reality.

Introduction

Over the years since the first edition of this book was published, belt conveying as a means of handling bulk materials has been greatly advanced and has assumed dominance in industry. The increased dependence of industry on belt conveyors is due primarily to the fact that they have had “everything going for them.” Demands of the mining industries as well as the other high-volume basic materials industries have been calling for ever-higher capacities and for more extensive belt conveyor systems having far greater complexity and employing higher degrees of automated control.

Concurrent with the continuing improvements in application and performance standards, there have been more stringent industry requirements for economic justifications of capital investment and relative costs of operation and maintenance. Radical changes in the relative costs of equipment, labor, and energy have further enhanced the favoring of belt conveyors for many relatively long-distance, high-volume haulage applications. Moreover, emphasis on our national problems of energy saving, environmental control, and safety has drawn a sharp focus on the many other advantageous features of belt conveyors.

Technological advances have paralleled and supported these increased application, performance, and economic demands. There has been a quiet revolution in the development of the available widths, normal operating speeds, and tension capabilities of conveyor belts. The mechanical components, too, such as idlers, drive machinery, pulleys, and many accessories have undergone developments to enhance performance and durability. Most recently, developments have been in the applications of automated, electrical controls employing solid state components, multiplex wiring, and many other modern techniques of electrical engineering and equipment. In the first edition of this book, the Conveyor Equipment Manufacturers Association (CEMA) stated its aims as being to make available in a book the experience and technical knowledge of its members as a contribution toward the design and construction of conveyors of superior performance and to provide the basic data and fundamentals of design for application to the ordinary belt conveyor problems toward the achievement

of successful performance. Adhering to these same aims and policy, CEMA is pleased to present this Fifth Edition of *Belt Conveyors for Bulk Materials*. It offers information on numerous examples of modern high-performance and complex systems and includes suggestions and guidelines for economic evaluations and conclusions. In this Fifth Edition, the presentation of belt conveyor design technology and engineering data has been updated both in its technology and in its presentation.

The technical information contained in this book is generally conservative in nature. Variations in specific application requirements or extreme service requirements should be addressed by member company engineering personnel whose depth of experience exceeds that covered in this text.

Belt Conveyor General Applications and Economics

Conveying of a variety of materials
Wide range of capacities
Adaptability to path of travel
Steep angle conveying
Loading, discharging, and stockpiling capabilities
Process functions
Reliability and availability
Environmental advantages
Safety
Low labor costs
Low power costs
Low maintenance costs
Long-distance transportation
Owning and operating costs
Present worth: required rate of return

Introduction

The subject of belt conveyors is of primary interest to all engineers, managers, and others who are responsible for selecting equipment for handling bulk materials. This book is primarily a design manual, but Chapter 1 is included to acquaint the reader with the many uses of belt conveyors and their advantages under widely varying conditions of operation.

Belt conveyors have attained a dominant position in transporting bulk materials due to such inherent advantages as their economy and safety of operation, reliability, versatility, and practically unlimited range of capacities. In addition, they are suitable for performing numerous processing functions in connection with their normal purpose of providing a continuous flow of material between operations. Recently, their conformity to environmental requirements has provided a further incentive for selection of belt conveyors over other means of transportation.

Low labor and low energy requirements are fundamental with belt conveyors as compared with other means of transportation. The dramatic increase in these operating costs since the oil crisis of the seventies has placed conveyors in an extremely favorable position for applications that were not considered previously.

Belt conveyor manufacturers have consistently anticipated the needs of industry with improvements in designs and with components that have exceeded all known requirements. Reliability and safety are outstanding now that stronger and more durable belts are available, as well as greatly improved mechanical parts and highly sophisticated electrical controls and safety devices.

Illustrated and described in this chapter are some of the advantages of belt conveyors, which are performing a wide variety of intraplant functions better and/or in a more innovative manner than is possible with other means of transporting bulk materials. Also included are examples of relatively long-distance belt conveyor systems which are being used extensively because they combine such important benefits as reliability, safety, and low cost per ton of material transported.

Conveying of a Variety of Materials

The size of materials that can be conveyed is limited by the width of the belt. Materials can range from very fine, dusty chemicals to large, lumpy ore, stone, coal or pulpwood logs. See Figure 1.1. Closely sized or friable materials are carried with minimum degradation. Because rubber belts are highly resistant to corrosion and abrasion, maintenance costs are comparatively low when handling highly corrosive materials or those that are extremely abrasive, such as alumina and sinter.

Materials that might cause sticking or packing if transported by other means are often handled successfully on belt conveyors. Even such hot materials as foundry shakeout sand, coke, sinter, and iron ore pellets are conveyed successfully.

Wide Range of Capacities

Current available belt conveyors are capable of handling hourly capacities in excess of any practical requirement. See Figure 1.2. Yet they are also used economically in plants for transporting materials between process units at a wide range of rates — sometimes as little as a mere dribble.

Figure 1.1 60-inch conveyor carries large lumps of abrasive ore on incline.



Belt conveyors operate continuously — around the clock and around the calendar when required — without loss of time for loading and unloading or empty return trips. Scheduling and dispatching are unnecessary as the material is loaded to and unloaded from the belt conveyor automatically. Operating labor costs differ little, regardless of capacity ratings. Overall costs per ton decrease dramatically, however, as annual tonnage handled increases. Such economic considerations are illustrated later in this chapter.

For these reasons, belt conveyors are capable of handling tonnages of bulk materials that would be more costly and often impractical to transport by other means.

Figure 1.2 96-inch conveyor at high-capacity coal-loading facility.



Figure 1.3 Regenerative conveyor lowers coal across existing terrain in direct path from mine.



Adaptability to Path of Travel

Belt conveyor systems provide the means of transporting materials via the shortest distance between the required loading and unloading points. They can follow existing terrain on grades of 30 to 35 percent, compared with the 6 to 8 percent effective limits for truck haulage. See Figure 1.3. They can be provided with structures which prevent the escape of dust to the surrounding atmosphere and are weather-protected. Such structures are economical and are adaptable to special requirements. See Figures 1.4 and 1.5. Belt conveyors provide a continuous flow of material while avoiding the confusion, delays, and safety hazards of rail and motor traffic in plants and other congested areas.

Paths of travel can be quite flexible, and the length of the routes can be extended repeatedly, as required. In some open-pit mining operations, conveyors thousands of feet long are shifted laterally on the bench to follow the progress of excavation at the face.

Technology has advanced substantially in the design and application of horizontally curved belt conveyors. As a conveyor's total length increases, so does the probability that transfer stations will be required to avoid some obstacle in its straight line path. Horizontal curves eliminate the constraints of the straight line conveyor and reduce the installed and operating cost of the conveyor (Figure 1.5).

Horizontally curved conveyors use conventionally troughed conveyor belts and standard components. The loaded and empty belt passes through the carry and return runs of the horizontal curve in unconstrained equilibrium by super-elevating the idlers. Horizontal curves may also be combined with convex and/or concave vertical curves to fit the conveyor to the most economical profile alignment.

Curved conveyors further enhance the reliability, availability and environmental advantages of the standard belt conveyor by eliminating the infrastructure and dust control requirements at transfer stations.

Figure 1.4 Corrugated metal cover over the belt provides weather and environmental protection.



Figure 1.5 Horizontal curve conveyor, conveying following natural terrain.



Steep Angle Conveying

Characteristics of bulk materials such as density, effective angle of internal friction, lump size and shape, are all factors which dictate the maximum incline angle up which material can be conveyed by a standard belt conveyor without having it roll or slip backwards on the belt. In Chapter 3, Table 3-3 lists maximum incline limits that conventional trough conveyors can safely convey various bulk materials. These maximum angles generally range from 10-30 degrees depending on the bulk material. Recent developments increase this incline angle thereby enhancing the flexible path that standard belt conveyors can now negotiate.

Simply making the belt cover irregular will increase its ability to carry materials up slightly greater inclines with most belt manufacturers capable of molding patterns $\frac{1}{4}$ inch to $1\frac{3}{8}$ inches high into the top cover. The belt manufacturers will however, for some materials, only increase the maximum incline recommendations by five degrees over a smooth belt cover. Larger cleats can either be hot or cold bonded, or mechanically fastened onto the belts' top cover after manufacture. These large cleats allow the conveying angle to increase to approximately 45 degrees. The conveying



Figure 1.6 Multiple loading stations feed ore to slope conveyor in open-pit mine.

capacity drops off rapidly as the angle increases. Cleats are available in various sizes, shapes and configurations with most allowing the belt to run in either a troughed or flat position. Cleated belts are usually restricted to short conveyors where few or no return idlers are needed and either the material does not stick to the surface or where the carryback is acceptable.

To improve on the cleat approach, corrugated sidewalls can be added to the transverse cleated belt to form complete rectangular partitions. These sidewalls and a transverse rigid belt carcass allow it to run flat without troughing idlers. The sidewalls also increase the load carrying capacity over the “cleat only” design and allows conveying up to 90 degree inclines.

One family of conveyors known as “pipe” or “tube” conveyors, “fold” belts, and “suspended” belts totally encloses the material with the belt increasing the allowable angle of incline. In each case, the internal area of the enclosed belt is fixed, and therefore is capable of transporting material up steeper inclines only if the internal area is completely loaded with material. At a steep angle these conveyors are unable to completely unload the trailing end of a load when no additional material is coming along to “push” the load up.

Another family using the sandwich belt principle totally encloses the material and applies pressure to secure it. This insures that neither sliding or roll-back will occur, even when conveying vertically. Systems are available that use only standard conveyor belts and components and therefore have high availability and low maintenance. These systems permit high belt speeds resulting in high capacity capabilities.

Consult with the respective CEMA member manufacturers of steep angle belt conveyors for more information and specific applications.



Figure 1.7 Rail-mounted hopper with feeder provides loading along full length of conveyor.

Loading, Discharging, and Stockpiling Capabilities

Belt conveyors are very flexible in their capabilities for receiving material from one or more locations and for delivering it to points or areas, as required by plant flow sheets. They can provide the main transportation artery while being loaded at several points (Figure 1.6) or anywhere along their length by equipment which provides a uniform feed to the belt (Figure 1.7). They are

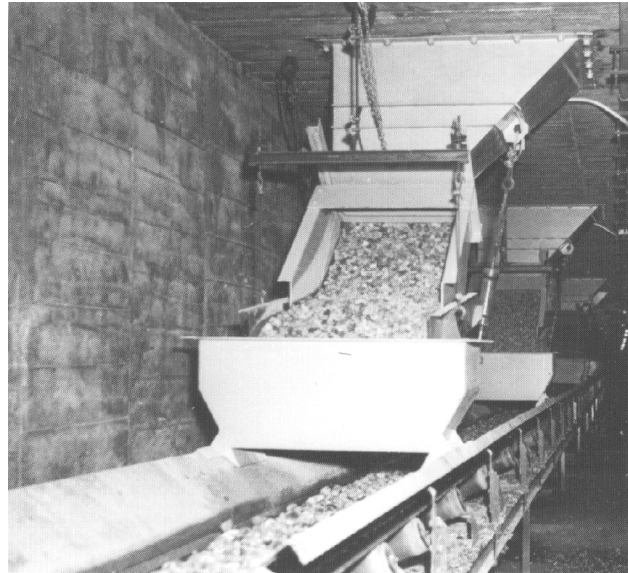


Figure 1.8 Multiple feeders in tunnel beneath stockpile provide efficient reclaiming and blending.



Figure 1.9 Material discharging over conveyor head pulley.

particularly useful in tunnels beneath stockpiles, from which they can reclaim and, where required, blend materials from various piles (Figure 1.8). Material can simply be discharged over the head end of each conveyor (Figure 1.9) or anywhere along its length by means of plows or traveling trippers (Figure 1.10).



Figure 1.10 Power-driven twin trippers distributing coal in power plant.

Figure 1.11 Double-winged stacker discharging into high-capacity stockpiles on either side of the feeding conveyor.



Figure 1.12 Circular stacker reclaimer simultaneously stacks and reclaims over 3,000,000 cubic feet of wood chips at a major paper mill.



Figure 1.13 Shuttle belt conveyors load and trim taconite pellets onto ore vessel on Great Lakes.



Belt conveyors, with their stackers and reclaimers, have become the only practical means for large-scale stockpiling and reclaiming of such bulk materials as coal, ore, wood-chips, and taconite pellets. See Figures 1.11 and 1.12. The combination circular stacker-reclaimer in Figure 1.12 illustrates this trend in rail to ship terminals. Even the shiploaders are equipped with conveyors for filling and trimming the holds of vessels at controlled rates. See Figure 1.13.



Figure 1.14 Self-unloading ship with 78-inch discharge conveyor unloads iron ore pellets at 10,000 tph.



Figure 1.15 Rail-mounted ship unloaders feed 60-inch conveyor system at steel company.

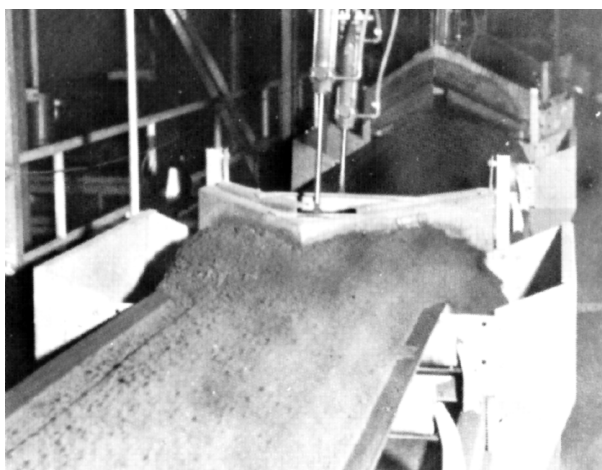


Figure 1.16 V-type plow diverts foundry sand from flat belt conveyor.

Self-unloading ships and lake vessels (Figure 1.14) equipped with belt conveyors can be unloaded in all ports, including those which do not have dockside unloading equipment (Figure 1.15). Unloading capacities of such systems are usually greater than those of several grab bucket unloaders, requiring less turnaround time and lower labor and other operating costs.

In contrast with the above-mentioned high-capacity unloading systems, certain materials, such as foundry sand, can be plowed from the belts (Figure 1.16) at specific locations in quantities controlled by the requirements of the application.

Process Functions

Although belt conveyors are generally used to transport and distribute materials, they are also used with auxiliary equipment for performing numerous functions during various stages of processing. A high degree of blending is accomplished as materials are bedded into and reclaimed from stockpiles.

Accurate samples of the material conveyed can be obtained by devices which cut through the stream of material as it flows from one conveyor to the next. Magnetic objects can be removed from the material (Figure 1.17). While being



Figure 1.17 Self cleaning cross belt magnetic separator.



Figure 1.18 Belt scales monitor rate with varying degrees of accuracy to provide both inventory and process control.

transported on the conveyor, materials can also be weighed accurately and continuously or they can be sorted, picked, or sprayed (Figures 1.18 and 1.19). In many cases, such operations are not only performed more effectively in connection with belt conveyors but are the only practical means.

Reliability and Availability

The reliability of belt conveyors has been proven over decades and in practically every industry. They are operating with the utmost reliability, many serving vital process units whose very success depends on continuous operation, such as handling coal in power plants, and transporting raw bulk materials in

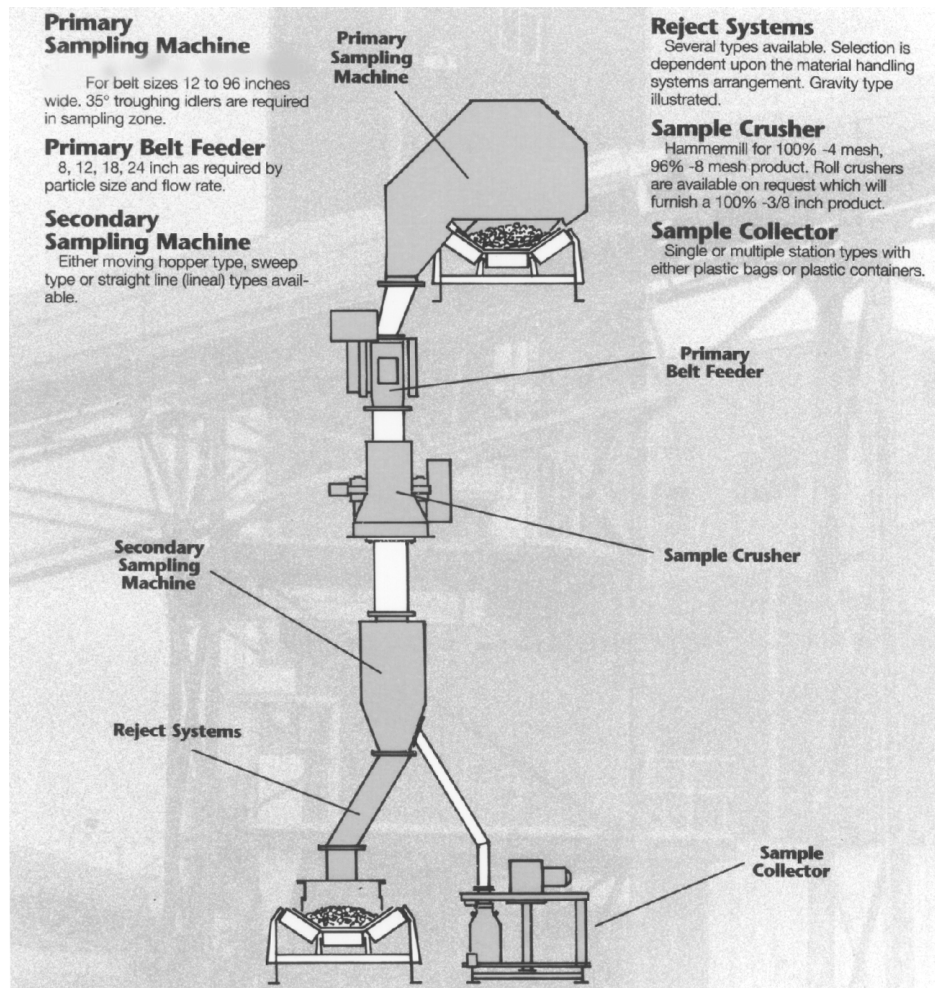


Figure 1.19 Multi-stage sampling systems are used in either the in-line or cross belt configurations to give desired process information on material specifications.

Figure 1.20 Operator controls entire conveyor system from control center with graphic display panels and push-button console.

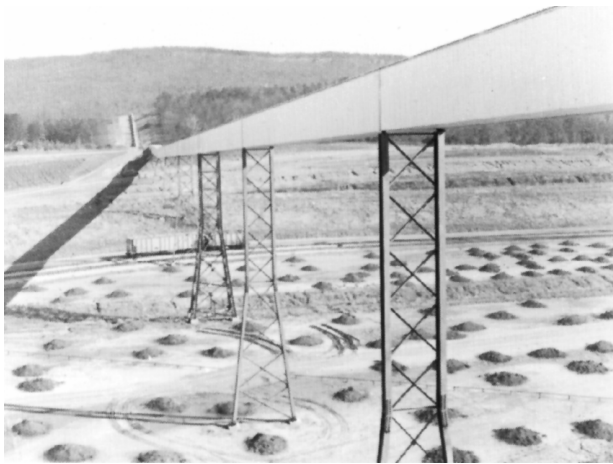
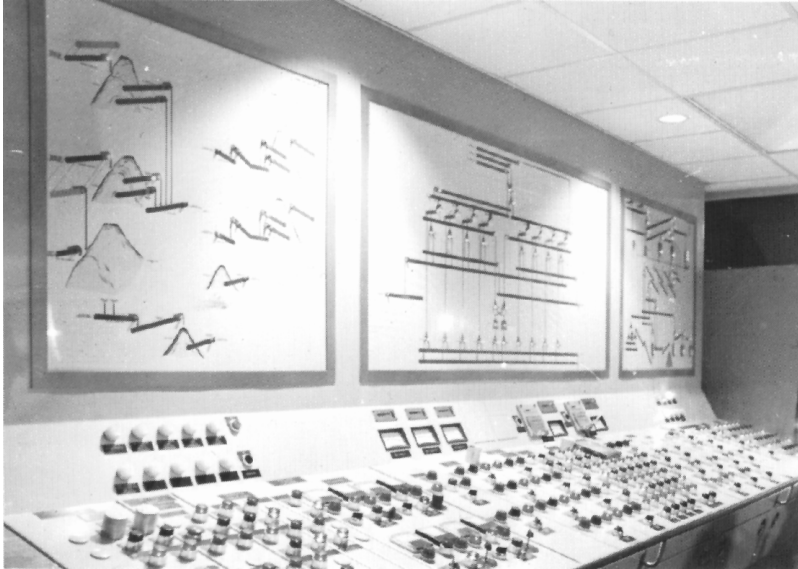


Figure 1.21 Conveyor in completely enclosed gallery carries its load safely overhead, avoiding any interference from highway or rail traffic.



Figure 1.22 Overland conveyor system utilizing a concrete support structure provides a pleasing appearance blending with the landscape.

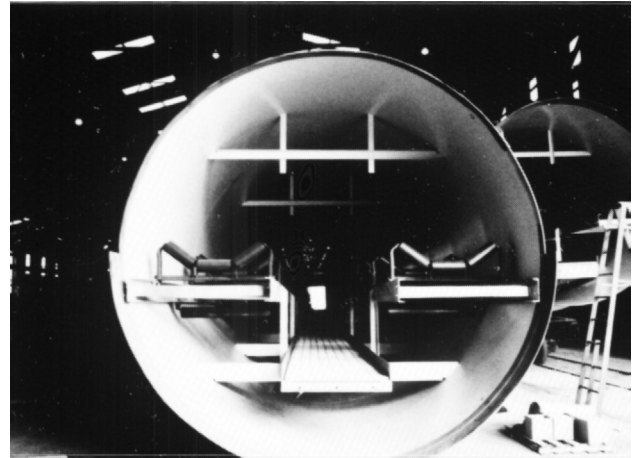
steel plants, cement plants, paper mills, and to and from ships in ports, where downtime is very costly.

Belt conveyors are operated at the touch of a button (Figure 1.20), at any time of the day or week. When required, they can and often do operate continuously, shift after shift. They can be housed so that both they and the material being transported are protected from elements that would otherwise impede the movement of trucks and certain other means of transportation.

Environmental Advantages

Belt conveyors are environmentally more acceptable than other means of transporting bulk materials; they neither pollute the air nor deafen the ears. They operate quietly, often in their own enclosures which, when desirable, can be located above the confusion and safety hazards of surface traffic or in small tunnels—out of sight and hearing. See Figure 1.21. Furthermore, they do not contaminate the air with dust or hydrocarbons. At transfers, dust can be contained within transfer chutes or collected with suitable equipment, if necessary. Finally, overland belt conveyor systems can be designed to blend into the landscape, resulting in an unscarred, quiet, and pollution-free operation. See Figure 1.22.

Figure 1.23 Tubular galleries provide opportunity to optimize shop assembly of components.



Tubular galleries (Figure 1.23) are becoming increasingly popular due to their long span capabilities, environmental features, aesthetics and the ability to shop assemble components to reduce field erection costs.

Safety

Belt conveyors operate with an extremely high degree of safety. Few personnel are required for operation and they are exposed to fewer hazards than with other means of transportation. Material is contained on the belt and personnel are not endangered by falling lumps or the malfunction of huge, unwieldy transport vehicles. Such vehicles also involve public liability, whether they operate over highways or in other areas accessible to the public. Also, conveyors offer less hazards to careless personnel than is inherent in other means of transporting bulk materials. The conveyor equipment itself can be protected from overload and malfunction by built-in mechanical and electrical safety devices.

Low Labor Costs

The labor hours per ton required to operate belt conveyor systems are usually the fewest of any method of transporting bulk materials. Like other low labor-intensive, highly automated operations, belt conveyors have low operating costs and provide a higher return on investment than competitive methods. Most functions of the system can be monitored from a central control panel or controlled by computer, allowing a minimum number of operating personnel to inspect the equipment and report conditions that may require attention by the maintenance department.

The time required for maintenance personnel is also minimal. As noted below in regard to maintenance costs, repairs and replacements of the relatively small parts can be made quickly at the site. Most belts can even be replaced in one shift — and some belts have conveyed well over 100 million tons before wearing out. Later in this chapter several examples of long-distance belt conveyor systems illustrate the effect of low labor costs.

Low Power Costs

The increasing cost of energy emphasizes the importance of power and its relation to the cost per ton for transporting bulk materials. Because belt conveyors are operated by electric power, they are less affected by the prices, shortages, and other limitations of liquid fuel. They consume power only when they are being used. There is no need for empty return trips or idling in line for the next load. On long systems the declined portion often assists in propelling an inclined or horizontal portion. Some conveyor systems are completely regenerative. See Figure 1.3. The cost of power for belt conveyor systems has always contributed to their extremely low operating costs, and this advantage has increased substantially with the rise in the cost of liquid fuels.

Low Maintenance Costs

Maintenance costs for belt conveyors are extremely low compared with most other means of transporting bulk materials. Extensive support systems, such as those commonly associated with truck haulage, are not required. Component parts are usually housed and have very long life compared with that of motor vehicles. Usually, they need only scheduled inspection and lubrication. Any repairs or replacements can be anticipated and unscheduled downtime avoided. Parts are small and accessible so replacements can be made on the site quickly and with minimal service equipment. Also, adequate inventories of spare parts can be maintained at a low cost and require relatively little storage space.

Long-Distance Transportation

The economic benefits of low operating costs for labor and energy, as well as some of the other advantages outlined above, have led to a widespread adoption of belt conveyor systems as a means of transporting bulk materials over increasingly long distances. Not only were these systems the best investments at the time they were installed, but the recent dramatic increases in the costs of both labor and liquid fuel have greatly enhanced their present value.

High Horsepower, High Tension Conveyors

Many design methods and technical issues are outside the scope of this manual. The CEMA design criteria in this manual are somewhat conservative and are based on many field trials over a wide range of operating conditions for average size conveyors.

Advanced technical design methods are often required for high energy, high tension conveyors having some of the following conditions:

- Length over 3,000 feet
- Horizontally curved
- Head and tail driven
- High lift
- Large decline with braking requirements
- Undulating geometrics

Dynamic analysis and computer modeling is often used in assessing the dynamic action and control of long overland and high lift, high horsepower conveyor systems.

It is recommended that you contact members of CEMA or other consultants when considering conveyors of this nature.

An excellent example of the use of sophisticated accounting methods for an investment decision was presented by F. W. Schweitzer and L. G. Dykers. It is based on appropriate cost estimates for a specific set of operating conditions and develops comparisons on present worth, discounted cash flow on investment, and wealth growth rate over a period of 10 years.

For the reader who wishes to compare a particular haulage problem with this example, we include tabular material and a brief discussion of a popular accounting method for this purpose. Dollar amounts are included in the tables simply to clarify the procedure and to illustrate the effects of inflation and depreciation. They do not apply to specific conditions, nor are they intended as an accurate comparison of the two haulage methods.

In order to make a true cost comparison between two or more haulage systems, all costs chargeable to each system must be considered, including ancillary facilities required for each system. For example, crushing facilities might be required to reduce lump size for handling on the belt conveyor but would not be necessary for truck haulage. Conversely, a very large shop with extensive special equipment would be required for a fleet of trucks whereas only relatively inexpensive equipment is necessary for servicing the small components of a belt conveyor. Also, the effect of inflation and added investment for additional or replacement equipment and facilities must be considered.

Owning and Operating Costs

Tables 1-1 and 1-2 present suggested methods for determining the capital investment and annual owning and operating costs for truck and conveyor hauls. These tables show the major factors contributing to the costs of each of the two systems. However, for an actual comparison, factors will be added or deducted to suit the particular project. The purpose of this discussion is to point out items other than actual truck or conveyor costs that must be considered in making a valid comparison.

In order to account for inflation and additional investments, the format allows for annual entries over the life of the project and therefore, gives annual costs. These annual figures are important in analyzing present worth, a method described later in this chapter.

As mentioned above, one of the considerations in the investment decision is the owning and operating cost per ton. The annual owning and operating cost is determined for each year over the life of the project. The cost per ton is then determined by dividing the annual owning and operating cost by the tonnage expected to be handled during that year. It is important to note that the annual operating hours have a significant effect on the results of the cost analysis, since the frequency of replacement and annual depreciation are directly affected.

Even though the owning and operating costs analysis indicates an advantage of one system over another, the more economical plan may require an initial investment which is higher than the alternate plan. The worth of this investment may be the determining factor in the investment decision. The following example illustrates one common method of measuring the worth.

Table 1-1. Owning and operating costs — conveyor haul. Annual expenditures in current dollars.

	Year									
	1	2	3	4	5	6	7	8	9	N
Capital investment										
Loading arrangement										
Conveyor equipment and structures										
Belting										
Erection of equipment and structures										
Foundations										
Electrical equipment and installation										
Site preparation										
Access road construction										
Lighting										
Repair shop and equipment										
Unloading arrangement										
Maintenance equipment for access road, conveyor movement, cleanup, etc.										
Owning costs per year										
Depreciation										
Interest, taxes, and insurance (% of book value)										
Operating costs per year										
Power										
Maintenance and repair of conveyor installation (labor and material)										
Maintenance of haul road										
Maintenance and repair of auxiliary equipment										
Labor										
Total cost										

Table 1-2. Owning and operating costs — truck haul. Annual expenditures in current dollars.

	Year									
	1	2	3	4	5	6	7	8	9	N
Capital investment										
Trucks less tires										
Haul road construction										
Haul road maintenance equipment										
Lighting										
Repair shop and equipment										
Repair parts and storage building										
Loading arrangement										
Unloading arrangement										
Engineering										
Owning costs per year										
Depreciation										
Interest, taxes, and insurance (% of book value)										
Operating costs per year										
Haul road maintenance										
Maintenance and repair of trucks										
Tires										
Fuel/oil/grease										
Labor (usually included in truck repair)										
operators										
mechanics										
Power for lighting										
Total cost										

Present Worth: Required Rate of Return

Tables 1-3, 1-4, and 1-5 demonstrate the procedure for evaluating the present worth of an investment. The cost figures used in Tables 1-3 and 1-4 are arbitrary and do not represent comparative figures for a specific truck and conveyor haul. To simplify the example, investment tax credit and other tax considerations (mainly depreciation), which would be advisable in an actual analysis, have been ignored. The example assumes 5 percent annual inflation.

Tables 1-3 and 1-4 show how the annual expenditures are determined for each system. The present worth of a system is the value of an investment (made today at the rate of return required by the investor) which will produce the annual owning and operating costs for that system over the life of the project. Since each system performs the same function, the system representing the lower investment is more attractive to the investor.

Table 1-3. Cash flow — conveyor system. *In 000's of dollars.*

	Year									
	1	2	3	4	5	6	7	8	9	10
Capital investment										
Conveyor system	1,900									
Loading system	260					330*				
Belting	700					890*				
(Repair shops and parts storage included in operating costs)										
Access road	50									
Operating costs										
Taxes and insurance	116	101	85	70	54	88	70	53	35	18
Power										
Maintenance and repair	287	301	316	332	349	366	385	404	424	445
Labor										
Access road maintenance	3	3	3	3	4	4	4	4	4	5
Total cash flow	3,316	405	404	405	407	1,678	459	461	463	468

* For the purpose of this example, these items are replaced at the end of year 5. Anticipated hours of operation and life expectancy of equipment determine the proper entry.

Table 1-4. Cash flow — truck system. *In 000's of dollars.*

	Year									
	1	2	3	4	5	6	7	8	9	10
Capital investment										
Trucks	1,400					1,787*				
Repair shop and parts storage	50									
Haul road	100									
Loading system	460					588*				
Operating costs										
Taxes and insurance	80	65	49	34	18	98	78	59	39	20
Maintenance of trucks										
Maintenance of loaders										
Labor	704	739	776	815	856	899	944	991	1,040	1,092
Fuel										
Tires										15
Haul road maintenance	10	10	11	12	12	13	13	14	15	
Total cash flow	2,804	814	836	861	886	3,385	1,035	1,064	1,094	1,127

* For the purpose of this example, these items are replaced at the end of year 5. Anticipated hours of operation and life expectancy of equipment determine the proper entry.

Table 1-5. Present value annual costs discounted at 20%. In 000's of dollars.

Year	Belt System			Truck System		
	Cash flow	Factor	Present value	Cash flow	Factor	Present value
1	\$3,316	.833	\$2,762	\$2,804	.833	\$2,336
2	405	.694	281	814	.694	565
3	404	.579	234	836	.579	484
4	405	.482	195	861	.482	415
5	407	.402	164	886	.402	356
6	1,678	.335	562	3,385	.335	1,134
7	459	.279	128	1,035	.279	289
8	461	.233	107	1,064	.233	248
9	463	.194	90	1,094	.194	212
10	468	.162	76	1,127	.162	183
Total			\$4,599			\$6,222

Difference in present value = \$6,222,000 - 4,599,000 = \$1,623,000

Table 1-5 shows the result using 20 percent as the required rate of return. In this example, the conveyor system requires a \$4,599,000 investment and the truck system, a \$6,222,000 investment. Since the conveyor system's present worth is lower than that of the truck system, the conveyor is considered to be a more attractive investment. In fact, it could justify an additional capital expenditure of \$1,623,000 (\$6,222,000 less \$4,599,000).

Belt conveyors have long been the primary choice for providing intraplant transportation of bulk materials in every industry in which such materials must be handled. Often, belts are the only practical means for this purpose. By reviewing their uses and advantages under the various conditions described above, the reader will gain a better understanding of their possibilities for application in his operations.

In recent years belt conveyors have gained wider acceptance as a means of transporting bulk materials over relatively long distances because of their lower overall costs per ton and other important benefits. Such long-distance transportation has traditionally been accomplished by rail or truck operations.

Summary

This book is published by CEMA to provide the reader with the information necessary to make a technical and economic assessment of belt conveyors as a possible solution to bulk materials handling problems. CEMA members will be glad to aid the reader further by furnishing additional data, estimates, and proposals.

Conveyor arrangements
Summaries of Chapters 3-14

The information presented in this book is intended to cover the basic principles of belt conveyor design and to include such formulas, tables and charts as are required to design most belt conveyors. The data are arranged in the order most convenient for the use of an experienced conveyor engineer, but the information is intended to be complete enough for use by engineers less familiar with belt conveyors.

The data are based on industry practice and are reliable when all design considerations are incorporated into the final design. They are applicable to the design of a high percentage of conveyors which are required to operate under reasonably normal conditions. However, some conveyors operate under conditions for which complete data are beyond the scope of this book. These design problems usually require broad experience for a satisfactory solution. A CEMA member should be consulted in such cases, as well as in the design of very large conveyors or complex conveyor systems. Assistance from CEMA members in meeting special design requirements will insure optimum economy and performance.

Belt conveyor design considerations and the location of data pertaining to them are outlined below.

Conveyor Arrangements

Belt conveyors can be arranged to follow an infinite number of profiles or paths of travel. Among these are conveyors which are horizontal, inclined, or declined, or with the inclusion of concave and convex curves, any combination of these. Also, numerous arrangements are possible for loading to and discharging from the conveyor.

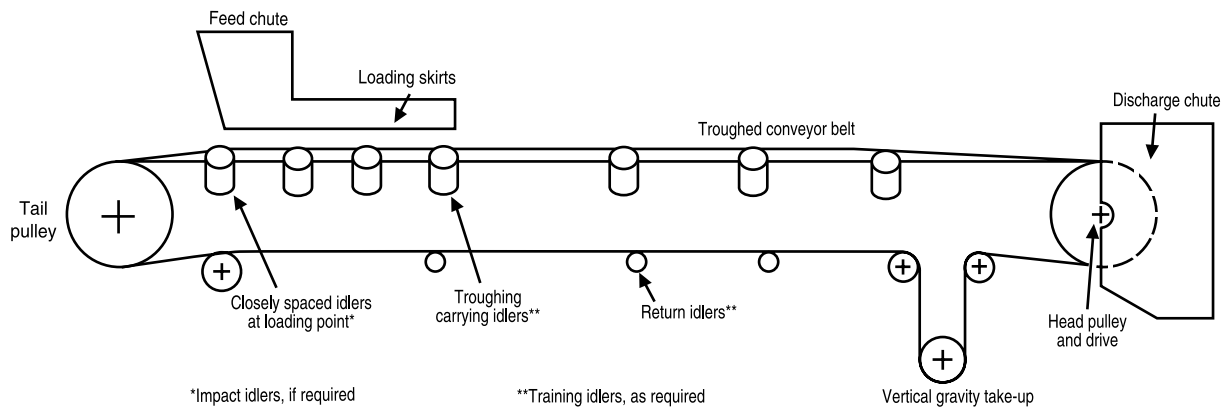


Figure 2.1 Nomenclature of components of a typical belt conveyor.

The nomenclature of typical belt conveyor components is illustrated in Figure 2.1. Various arrangements are illustrated in Figures 2.2 through 2.9; loading and discharging arrangements are shown in Figures 2.10 through 2.18. In addition, belt conveyor stackers can be connected to a mainline conveyor to stockpile material several hundred feet on either or both sides into piles of any practical length, as shown in Figure 1.11 in Chapter 1. Or a combination stacker-reclaimer (Figure 1.12) can be used to stockpile material or to reclaim and return it to the main belt conveyor.

Belt conveyors can be designed for practically any desired path of travel, limited only by the strength of belt, angle of incline or decline, or available space. Some of the profiles shown below are more desirable than others. For example, transfers between conveyors should be avoided where possible due to additional wear on the belts at the loading points, dust raised, and possible plugging in the transfer chutes. For these reasons, the arrangement in Figure 2.3 is preferable to those shown in Figures 2.5 and 2.7. Similarly, the profile shown in Figure 2.4 is usually better than that in Figure 2.6.

TYPICAL BELT CONVEYOR TRAVEL PATHS

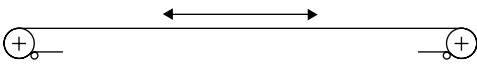


Figure 2.2 Horizontal belt.

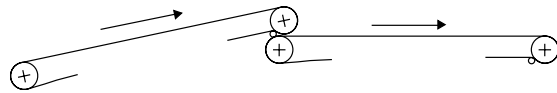


Figure 2.6 Ascending and horizontal path, when advisable to use two conveyor belts.



Figure 2.3 Horizontal and ascending path, when space will permit vertical curve and belt strength will permit one belt.

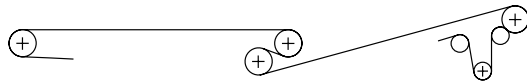


Figure 2.7 Possible horizontal and ascending path, when space will not permit vertical curve but belt strength will permit only one belt.



Figure 2.4 Ascending and horizontal path, when belt tensions will permit one belt and space will permit vertical curve.



Figure 2.8 Compound path with declines, horizontal portions, vertical curves, and incline.

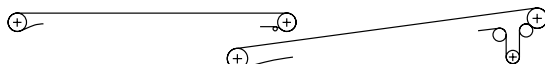


Figure 2.5 Possible horizontal and ascending path, when space will not permit a vertical curve or when the conveyor belt strength requires two belts.

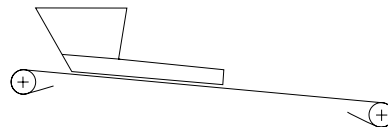


Figure 2.9 Loading can be accomplished, as shown, on minor inclines or declines.

TYPICAL BELT CONVEYOR LOADING AND DISCHARGING ARRANGEMENTS

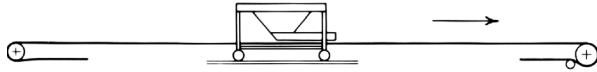


Figure 2.10 Traveling loading chute to receive materials at a number of points along conveyor.

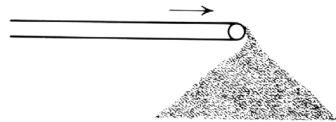


Figure 2.11 Discharge over end pulley to form conical pile.

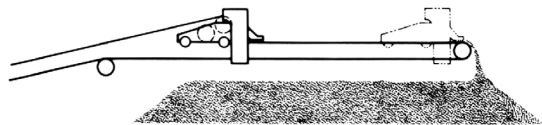


Figure 2.12 Discharge by traveling tripper or through the tripper to the storage pile. See Figure 2.14.

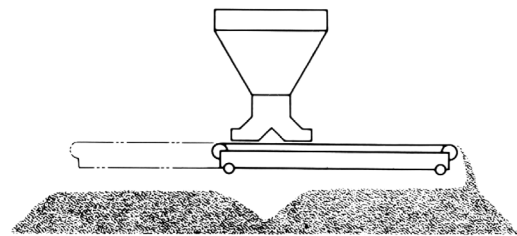


Figure 2.13 Discharge over either end-pulley of a reversible shuttle belt conveyor.

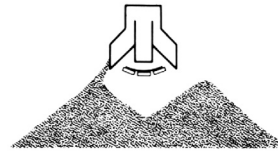


Figure 2.14 Discharge from tripper to one side only, to both sides, or forward again on conveyor belt.

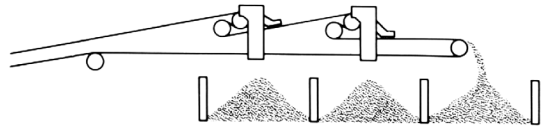


Figure 2.15 Discharge by fixed trippers with or without cross conveyors to fixed piles or bin openings.

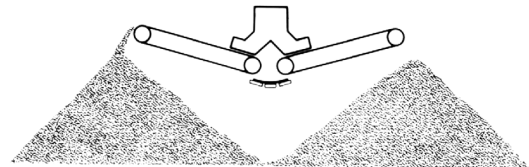


Figure 2.16 Discharge by traveling or stationary trippers to ascending cross conveyors carried by tripper.

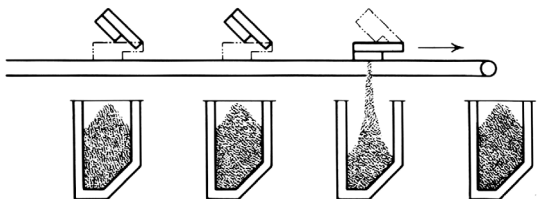


Figure 2.17 Discharge by hinged plows to one or more fixed locations on one or both sides of conveyor plows. Device can be adjusted for proportioned discharge to several places.

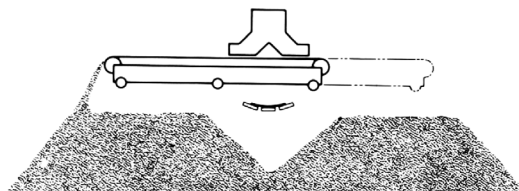


Figure 2.18 Discharge by traveling or stationary tripper carrying reversible shuttle belt.

Figure 2.8 illustrates a profile that can be used on a single, long, overland conveyor, often with several convex and concave curves, as shown in Figure 2.19. On such conveyors, it is essential that belt tensions be determined along the entire length in order to calculate the radii of the curves and locate the maximum belt tension which can occur at an intermediate point.

A few typical arrangements for loading and discharging material are illustrated on page 22.

Figure 1.11 illustrates a traveling stacker stockpiling along a feed conveyor. This type of stacker comprises a traveling tripper with two boom conveyors discharging alternately to each side. The tripper lifts the material from the feed conveyor onto either of the booms for discharge into the piles. The traveling stacker can also be supplied with a single boom fixed for discharge to one side only or with a single, slewing boom which can discharge to both sides of the feed conveyor. See Figure 2.20.



Figure 2.19 This overland conveyor following the natural terrain has an undulating profile with several concave and convex curves.

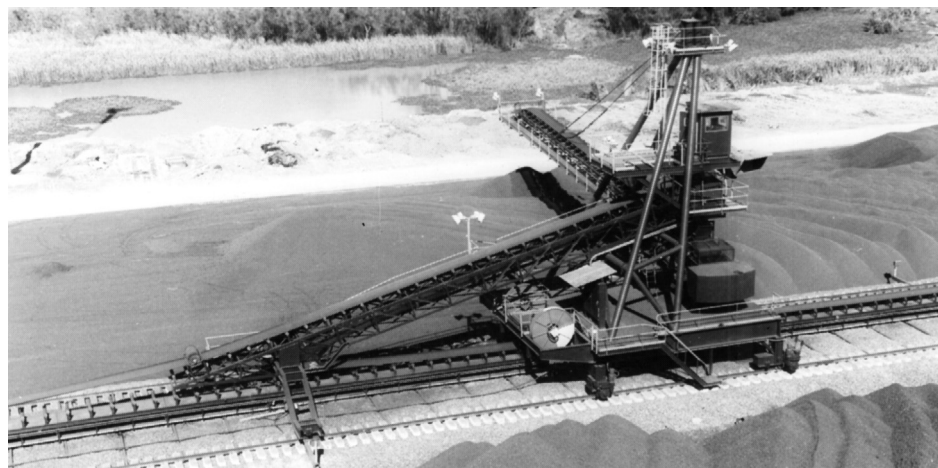


Figure 2.20 Traveling winged stacker with slewing boom stockpiles on both sides of the feed conveyor.

Reclaiming from the elongated piles formed by these stackers is efficiently accomplished by either a combination stacker-reclaimer (Figure 2.21), which utilizes the reversing boom conveyor for both stacking and reclaiming, or by a separate bucket wheel reclaimer, different types of which are shown in Figures 2.22 and 2.23.

Figure 2.21 Bucket wheel of this combination stacker-reclaimer discharges onto the stacker boom conveyor, which is reversed for the reclaim operation.

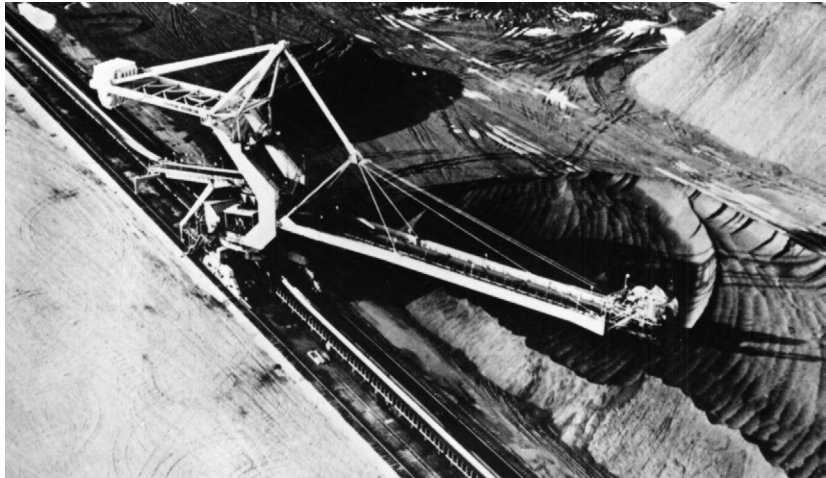


Figure 2.22 Bucket wheel reclaimer with wheel at end of boom.



Figure 2.23 Crawler-mounted bucket wheel reclaimer with wheel mounted on chassis discharges onto a rail-mounted reclaim conveyor with slewing boom.



Summaries of Chapters 3-14

The following chapters cover various engineering design considerations, including pertinent data, formulas, and applications. Each chapter topic is briefly summarized below.

Characteristics and Conveyability of Bulk Materials (Chapter 3)

Belt conveyors are capable of handling an almost unlimited variety of materials. However, their successful performance depends upon a design that is based on a thorough understanding of the characteristics of the material to be handled and careful consideration of its anticipated behavior while being carried on the belt, as well as during delivery to and from it. Some of the characteristics which affect basic design include size of lumps, bulk density, angle of repose, abrasiveness, moisture content, dustiness, stickiness, temperature, and chemical action. The influence of these and other material characteristics on belt conveyor design is discussed in Chapter 3. Most materials which have been handled successfully on belt conveyors are listed and classified in Table 3-3.

Capacities, Belt Widths, and Speeds (Chapter 4)

In its simplest terms the rated hourly capacity of a belt conveyor is the total weight of material conveyed in one hour with the belt continuously carrying a uniform cross section of material and traveling at a uniform speed. However, to obtain a desired capacity with successful operation it is important to consider the many factors which influence both belt width and speed.

Width can be determined by the size of lumps, the percentage of lumps to fines, and the angle of repose of the material. Maximum speeds which are acceptable for conveying some materials under certain conditions may not be appropriate for conveying dusty or fragile materials or heavy, sharp-edged ore. Even the cross-sectional area of the load on the belt may vary according to the type of idler used. Design data

pertaining to belt widths, speeds, and capacities, including tables and formulas, are covered in Chapter 4.

Belt Conveyor Idlers (Chapter 5)

It is important to select the best idler design for a specific condition, since idlers greatly influence belt tensions, power requirements, belt life, capacity, and operational success of a conveyor. Use of the right carrying idler and proper spacing can minimize frictional resistance. Some idlers carry larger cross-sectional loads than others for appropriate materials. Others are needed for such purposes as training the belt, cushioning the belt against heavy lumps, and handling extremely abrasive or sticky materials. The design, selection, and application data for belt conveyor idlers are presented in Chapter 5.

Belt Tension, Power, and Drive Engineering (Chapter 6)

Belt tensions not only determine the required strength of the belt, but also influence the design of many other mechanical parts. Furthermore, the power required to drive the conveyor is calculated from the belt tensions.

Factors which contribute to belt tensions and their relation to power requirements, along with appropriate formulas and tables, are covered in Chapter 6. Among the principal factors discussed are the power required to overcome frictional resistance and lift the load, acceleration and deceleration, and drive arrangements. Good design can assure lower belt tension by the proper arrangement of drive pulleys, their lagging, and degrees of belt wrap. Tensions resulting from acceleration can be limited by appropriate electrical controls, as described in Chapter 13.

In order to assist the reader in the use of this data, two different methods for determining tension and power requirements are illustrated. The graphical method, shown on page 141, is used to calculate quickly and conveniently the horsepower required to drive a conveyor. Belt tensions can be determined based on these calculations of horsepower. This method is suitable for use in designing simple, straight-line, horizontal, or inclined conveyors.

The analytical method, described on page 145, provides calculations of all belt tensions as they occur along the conveyor, thus determining magnitudes at intermediate points. This information is necessary to design convex and concave vertical curves, trippers, and remotely located takeups. It is also used to determine the location of maximum belt tension when there is a multiple change of slope in the belt line. The power required to drive or restrain the conveyor is derived from the tension, which represents the amount of pull at the drive pulley necessary to propel or retard the loaded belt. The analytical method is more accurate than the graphical method, and should be used for designing conveyors which exceed the limits set for the former method.

Belt Selection (Chapter 7)

Selection of the conveyor belt is a most important design consideration, since the belt constitutes a large portion of the initial cost of a belt conveyor. Also, it is subject to the most wear and contributes substantially to the operating costs of a conveyor.

Selection of the most suitable belt for the required service involves careful consideration of the construction of the belt in conjunction with the idlers and other mechanical components. There are so many belt constructions that it is impractical to include more than a few of them in Chapter 7. Also, the rubber industry is making such rapid strides in the use of synthetic fibers and steel-cord carcasses that improved constructions are frequently available.

Chapter 7 contains information pertaining to the conservative selection of suitable belts for most applications. However, for large, heavily loaded belts or complex conveyor systems, it is advisable to consult a CEMA member to assist in selecting the most economical belt for the required service.

**Pulleys and
Shafts
(Chapter 8)**

Because conveyor pulleys and shafts form a composite structure in operation, it is accepted engineering practice to consider their design together. Chapter 8 contains tables and formulas to assist the designer in selecting the most effective pulley and shaft combination. Also included is information on special-purpose pulleys, such as those with rubber lagging for better traction, herringbone-grooved lagging for a relatively “non-skid” grip on a wet belt, and wing or slat pulleys for use in handling sticky materials.

**Vertical Curves
(Chapter 9)**

Two types of vertical curves may be required in the profile of a belt conveyor, namely, concave and convex. In the concave curve the carrying belt turns upward on a radius whose center is above the belt. If improperly designed, the loaded belt may lift off the idlers and result in excessive spillage. The convex curve turns downward on a radius whose center is beneath the belt. In this case, improper design can mean excessive tensions at the edges of the belt, causing damage to it. Chapter 9 includes tables and formulas for use in the proper design of vertical curves.

**A Guide to Steep
Angle Conveying
(Chapter 10)**

The Steep Angle Conveying chapter provides a comprehensive survey of current successful approaches for conveying bulk materials up and down steep inclines. This capability, combined with the path adaptability of bulk conveyor systems, gives belt conveyors a high degree of versatility in moving bulk materials between loading and unloading points.

This chapter draws extensively upon materials developed elsewhere in the book and they are generally relevant to the steep angle conveying problem. However, allowable material cross sections and lump size restrictions differ from standard belt conveyor practice. In some cases, the method of tension calculations must also be modified.

**Belt Takeups,
Cleaners, and
Accessories
(Chapter 11)**

In addition to the data required to design a basic belt conveyor, there are numerous accessory items of equipment which must be considered in light of their contribution to the economical and successful operation of the conveyor. For example, belt takeups are necessary for maintaining the proper belt tension for drive pulley traction. Cleaning devices can reduce material cleanup problems and eliminate a source of misalignment which might otherwise result in damage to the belt.

Other accessory devices are available for protecting the belt and/or performing certain process functions. Such devices include belt covers, magnetic pulleys for removing tramp iron from the material being conveyed, tramp iron detectors, and various methods and devices for weighing and/or sampling the conveyed material accurately and continuously.

Conveyor Loading and Discharge (Chapter 12)

An extremely important consideration in belt conveyor design is the proper loading of the belt. Improper loading may result in accelerated belt wear, spillage, and/or reduced capacity. Discharge from the belt must be considered in relation to the materials handled, where they are to be delivered, and the trajectory of discharge over the head pulley. Trajectories and chute design are particularly important at transfers from one conveyor to another. Discharge at points other than the head pulley can be accomplished by fixed or movable trippers, plows, etc.

In addition to trippers, Chapter 12 considers auxiliary devices for feeding material to the belt and for receiving the flow from the belt for purposes of stockpiling or otherwise disposing of it.

Conveyor Motor Drives and Controls (Chapter 13)

The electrical equipment must be capable not only of propelling the fully loaded conveyor at its designed speed, but also of controlling starting acceleration within design limits to prevent excessive tension in the belt. Chapter 13 contains tables, charts, and other information on motor types and control devices useful for this purpose. Also included are data on necessary interlocks for two or more conveyors of a system.

A number of important electronic control devices are discussed in Chapter 13, as well as the advantages in using such modern electronic equipment as programmable controllers, computers, and multiplexing.

Operation, Maintenance, and Safety (Chapter 14)

Chapter 14 provides guidance on the considerations of operation, maintenance, and safety that are so important to assure dependability, efficient utilization, and reasonable life expectancy of the equipment.

Characteristics and Conveyability of Bulk Materials

Material characteristics
Behavior of materials on a moving belt
Effect of inclines and declines
Material class description

Material Characteristics

The successful design of a belt conveyor must begin with an accurate appraisal of the characteristics of the material to be transported. A few important characteristics require definition.

The angle of repose of a material is the acute angle which the surface of a normal, freely formed pile makes to the horizontal.

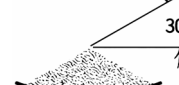
The angle of surcharge of a material is the angle to the horizontal which the surface of the material assumes while the material is at rest on a moving conveyor belt. This angle usually is 5 degrees to 15 degrees less than the angle of repose, though in some materials it may be as much as 20 degrees less.

The flowability of a material, as measured by its angle of repose and angle of surcharge, determines the cross-section of the material load which safely can be carried on a belt. It also is an index of the safe angle of incline of the belt conveyor.

The flowability is determined by such material characteristics as: size and shape of the fine particles and lumps, roughness or smoothness of the surface of the material particles, proportion of fines and lumps present, and moisture content of the material.

Table 3-1 illustrates and defines the normal relationship of the foregoing properties and the general characteristics of materials.

Table 3-1. Flowability—angle of surcharge—angle of repose.

Flow				
Very free flowing 1*	Free flowing 2*	Average flowing 3*		Sluggish 4*
Angle of Surcharge				
5°	10°	20°	25°	30°
				
Angle of Repose				
0-19°	20-25°	30-34°	35-39°	40° - up
Material Characteristics				
Uniform size, very small rounded particles, either very wet or very dry, such as dry silica sand, cement, wet concrete, etc.	Rounded, dry polished particles, of medium weight, such as whole grain and beans.	Irregular, granular or lumpy materials of medium weight, such as anthracite coal, cottonseed meal, clay, etc.	Typical common materials such as bituminous coal, stone, most ores, etc.	Irregular, stringy, fibrous, interlocking material, such as wood chips, bagasse, tempered foundry sand, etc.

*Code designations conform to bulk material characteristics chart, Table 3-2.

Consideration should also be given to the weight per cubic foot of the material; its dustiness, wetness, stickiness, abrasiveness; its chemically corrosive action; and its temperature. Some general information concerning these properties of many materi-

als is given in Table 3-2. Table 3-3 gives a list of materials with their physical characteristics and classification code designations. It must be understood that the data given in this table are for average conditions and average materials. Each characteristic may vary in specific instances, especially angles of repose and maximum conveyor inclinations.

Proper consideration must be given to those materials, the characteristics of which vary under various conditions of handling, atmospheric humidity, age, or long storage. In some cases, accuracy may require that carefully conducted tests be run to establish the material characteristics under the required conditions.

Materials or characteristics omitted from Table 3-3 may be roughly appraised by comparison with listed materials of the same general type.

Behavior of Materials on a Moving Belt

Attention must be drawn to the fact that the normal characteristics of materials are considerably influenced by the movement, slope, and speed of the conveyor belt that carries them.

As the conveyor belt passes successively over each carrying idler, the material on it is correspondingly agitated. This agitation tends to work the larger pieces to the surface of the load and the smaller particles or fines to the bottom. It also tends to flatten the material surface slope (i.e., the angle of surcharge) and explains why this angle is less than the angle of repose.

Any difference between the forward velocity of the material as it is being loaded and the conveyor belt that is receiving it must be equalized by the acceleration of the material. This acceleration causes turbulence in the material.

Any vertical velocity of the material as it is being loaded must be absorbed in the resilient construction of the conveyor belt and the impact idlers used under the loading point. In this process, a further increase in material turbulence is produced.

These three influences are emphasized when the conveyor belt is on an incline or decline, and also when the conveyor belt is operated at high speeds. These influences are emphasized even more when the material handled is loose and contains large rounded lumps, such as coarse washed gravel, the tendency of which is to bounce and roll on the conveyor belt.

Effect of Inclines and Declines

The nominal cross section of the material on a horizontal conveyor belt is measured in a plane normal to the belt. On an inclined or declined conveyor belt, gravity necessitates that the actual cross section of the load be considered in a vertical plane. To maintain the total width of the material load on the belt and to maintain unchanged surcharge angles, the cross section of the load possible on an inclined or declined belt must be less than that on a horizontal belt. Referring to the diagram used in the derivation of belt conveyor capacity (Figure 4.2), the area A_b does not change although the area A_s does decrease as the cosine of the conveyor slope.

The total effect is influenced by the surcharge angle at which the material will ride on the conveyor belt. However, in most cases, the actual loss of capacity is less than 3 percent.

The following three precautions may well be observed: lumps are more likely to roll off the edges of inclined conveyor belts than horizontal ones; for belts of constant slope, the spillage of material is more likely to occur immediately beyond the loading point; materials which aerate excessively, such as some very fine ground cements, or materials in which the proportion of water is so high that a slurry is created, must be carried on inclines and at such a conveyor belt speed that the tendency of the material to slide back is fully offset.

Table 3-2. Material class description.

	Material Characteristics	Code
Size	Very fine—100 mesh and under	A
	Fine— $\frac{1}{8}$ inch and under	B
	Granular—Under $\frac{1}{2}$ inch	C
	Lumpy—containing lumps over $\frac{1}{2}$ inch	D
	Irregular—stringy, interlocking, mats together	E
Flowability	Very free flowing—angle of repose less than 19°	1
Angle of	Free-flowing—angle of repose 20° to 29°	2
Repose	Average flowing—angle of repose 30° to 39°	3
	Sluggish—angle of repose 40° and over	4
Abrasiveness	Nonabrasive	5
	Abrasive	6
	Very abrasive	7
	Very sharp—cuts or gouges belt covers	8
Miscellaneous Characteristics (Sometimes more than one of these characteristics may apply)	Very dusty	L
	Aerates and develops fluid characteristics	M
	Contains explosive dust	N
	Contaminable, affecting use or saleability	P
	Degradable, affecting use or saleability	Q
	Gives off harmful fumes or dust	R
	Highly corrosive	S
	Mildly corrosive	T
	Hygroscopic	U
	Interlocks or mats	V
	Oils or chemical present—may affect rubber products	W
	Packs under pressure	X
	Very light and fluffy—may be wind-swept	Y
	Elevated temperature	Z

Example: A very fine material that is free-flowing, abrasive, and contains explosive dust would be designated: Class A26N.

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Alfalfa meal	17	45		B46Y
Alfalfa pellets	41-43	20-29		C25
Alfalfa seed	10-15	29		B26N
Almonds, broken or whole	28-30	30-44		C36Q
Alum, fine	45-50	30-44		B35
Alum, lumpy	50-60	30-44		D35
Alumina	50-65	22	10-12	B27M
*Aluminum chips	7-15	45		E46Y
Aluminum hydrate	18	34	20-24	C35
Aluminum ore (see bauxite)	—			—
Aluminum oxide	70-120	29		A27M
Aluminum silicate	49	30-44		B35S
Aluminum sulphate	54	32	17	D35
Ammonium chloride, crystalline	45-52	30-44		B36S
Ammonium nitrate	45	30-44		*C36NUS
Ammonium sulphate (granular)	45-58	44		*C35TU
Antimony powder	60	30-44		A36
Aplite	70-80	30-44		A35
Arsenic, pulverized	30	20-29		*A26
Arsenic oxide	100-120	30-44		A35R
Asbestos, ore or rock	81	30-44		D37R
Asbestos, shred	20-25	45		E46XY
*Ash, black, ground	105	32	17	*B35
Ashes, coal, dry, 3 inches and under	35-40	45		D46T
Ashes, coal, wet, 3 inches and under	45-50	45		D46T
Ashes, fly	40-45	42	20-25	A37
Ashes, gas-producer, wet	78			D47T
Asphalt, binder for paving	80-85			C45
Asphalt, crushed, ½ inch and under	45	30-44		C35
Bagasse	7-10	45		E45Y
Bakelite & similar plastics (powdered)	35-45	45		B45
Barite	180	30-44		B36
Barium carbonate	72	45		A45
*Barium carbonate filter cake	72	32		A36

* May vary considerably — consult a CEMA member

Characteristics and Conveyability of Bulk Materials

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Barium hydrate	62-65	43		A36
Barium oxide	150-200			A46
*Bark, wood, refuse	10-20	45	27	E45VY
Barley	37-48	23	10-15	B25N
Basalt	80-103	20-28		B26
Bauxite, ground, dry	68	20-29	20	B26
Bauxite, mine run	80-90	31	17	E37
Bauxite, crushed, 3 inches and under	75-85	30-44	20	D37
Beans, castor, whole	36	20-29	8-10	C25W
Beans, castor, meal	35-40			B15W
Beans, navy, dry	48	29		C25
Beans, navy, steeped	60	35-40		C35
Beet pulp, dry	12-15			E45
Beet pulp, wet	25-45			E46
Beets, whole	48	50		D45
*Bentonite, crude	35-40	42-44		D36X
Bentonite, 100 mesh and under	50-60	42	20	A36XY
Bones	34-40	45		*C46
Boneblack, 100 mesh and under	20-25	20-29		A25Y
Bonechar	27-40	30-44		B36
Bonemeal	50-60	30-44		B36
Borate of lime	60	30-44		A35
Borax, ½-inch screenings	55-60	30-44		C36
Borax, 3 inches and under	60-70	30-44		D35
Boric acid, fine	55	20-29		B26T
Bran	10-20	30-44		B35NY
Brewer's grain, spent, dry	25-30	45		C45
Brewer's grain, spent, wet	55-60	45		C45T
Bronze chips, dry	30-50	44-57		B47
Buckwheat	37-42	25	11-13	B25N
Calcium carbide (crushed)	70-80	30-44		D36N
Carbon, activated, dry, fine	8-20	20-29		B26Y
Carbon black, pelletized	20-25	25		B25Q
Carbon black, powder	4-7	30-44		*A35Y
Carborundum, 3 inches and under	100	20-29		D27
Casein	36	30-44		B35

* May vary considerably — consult a CEMA member

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Cast iron chips	90-120	45		C46
Caustic soda	88	29-43		A36
Cement, Portland	72-99	30-44	20-23	A36M
Cement, Portland, aerated	60-75			A16M
Cement, rock (see limestone)	100-110			D36
Cement clinker	75-95	30-40	18-20	D37
Chalk, lumpy	75-85	45		D46
*Charcoal	18-25	35	20-25	D36Q
Chrome ore (chromite)	125-140	30-44		D37
Cinders, blast furnace	57	35	18-20	*D37T
Cinders, coal	40	35	20	*D37T
Clay (see also bentonite, diatomaceous earth, fullers earth, kaolin, and Marl)	—			—
Clay calcined	80-100			B37
Clay, ceramic, dry, fines	60-80	30-44		A35
Clay, dry, fines	100-120	35	20-22	C37
Clay, dry, lumpy	60-75	35	18-20	D36
Clinker, cement (see cement clinker)	—			—
Clover seed	48	28	15	B25N
Coal, anthracite, river, or culm, 1/8 inch and under	60	35	18	B35TY
Coal, anthracite, sized	55-60	27	16	C26
Coal, bituminous, mined 50 mesh and under	50-54	45	24	B45T
Coal, bituminous, mined and sized	45-55	35	16	D35T
Coal, bituminous, mined, run of mine	45-55	38	18	D35T
*Coal, bituminous, mined, slack, 1/2 inch and under	43-50	40	22	C35T
Coal, bituminous, stripping, not cleaned	50-60			D36T
Coal, lignite	40-45	38	22	D36T
Cocoa beans	30-45	30-44		C35Q
Cocoa nibs	35	30-44		C35
Coffee, chaff	20	20-29		B25MY
Coffee, green bean	32-45	30-44	10-15	C35Q
Coffee, ground	25	23	10	B25
*Coffee, roasted bean	22-26			C25PQU
Coffee, soluble	19			B45PQ
Coke, loose	23-35	30-44	18	B37QVT
Coke, petroleum calcined	35-45	30-44	20	D36Y
Coke breeze, 1/4 inch and under	25-35	30-44	20-22	C37Y

* May vary considerably — consult a CEMA member

Characteristics and Conveyability of Bulk Materials

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Compost	30-50			E45ST
Concrete, cinder	90-100		12-30	D46
Copper ore	120-150	30-44	20	*D37
Copper sulfate	75-85	31	17	D36
Cork, granulated	12-15			C45
Corn, cracked	45-50			C35W
Corn, ear	56			
Corn, shelled	45	21	10	C25NW
Corn sugar	31	30-44		B35
Corn germs	21			B35W
Corn grits	40-45	30-44		B35W
Cornmeal	32-40	35	22	B35W
Cottonseed, dry, de-linted	22-40	29	16	C35W
Cottonseed, dry, not de-linted	18-25	35	19	C35W
Cottonseed cake, crushed	40-45	30-44		B35
Cottonseed cake, lumpy	40-45	30-44		D35W
Cottonseed hulls	12	45		*B45Y
Cottonseed meal	35-40	35	22	B35W
Cottonseed meats	40	30-44		B35W
Cracklings, crushed, 3 inches and under	40-50	45		D45
Cryolite, dust	75-90	30-44		A36
Cryolite, lumpy	90-100	30-44		D36
Cullet	80-120	30-44	20	D37Z
Diatomaceous earth	11-14	30-44		A36MY
Dicalcium phosphate	40-50	45		A45
Disodium phosphate	25-31	30-44		B36QT
Dolomite, lumpy	80-100	30-44	22	D36
Dolomite, pulverized	46	41		B36
Earth, as excavated—dry	70-80	35	20	B36
Earth, wet, containing clay	100-110	45	23	B46
Ebonite, crushed ½ inch and under	65-70	30-44		C35
Emery	230	20-29		A27
Epsom salts	40-50	30-44		B35
Feldspar, ½-inch screenings	70-85	38	18	B36

* May vary considerably — consult a CEMA member

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Feldspar, 1½- to 3-inch lumps	90-110	34	17	D36
Feldspar, 200 mesh	100	30-44		A37
Ferrous carbonate	85-90	30-44		B36
Ferrous sulfate	50-75			C36
Ferrous sulfide	120-135	20-29		C36
Filter press mud, sugar factory	70			A15
Fish meal	35-40			B45W
Fish scrap	40-50			E45W
Flaxseed	45	21	12	B25NW
Flaxseed meal	25	30-44		B35W
Flour, wheat	35-40	45	21	A45PN
Flue dust, boiler house, dry	35-40	20		A17MTY
Fluorspar, ½-inch screenings	85-105	45		C46
Fluorspar, 1½- to 3-inch lumps	110-120	45		D46
Foundry refuse, old sand cores, etc.	70-100	30-44		D37Z
Fuller's earth, dry	30-35	23		B26
Fuller's earth, oily	60-65	20-29		B26
Fuller's earth, oil filter, burned	40	20-29		B26
Fuller's earth, oil filter, raw	35-40	35	20	*B26
Galena (lead sulfide)	240-260	30-44		A36
Gelatin, granulated	32	20-29		C25Q
Glass batch (textile fiber glass)	45-55	0-10		A16LM
Glass batch (wool and container)	80-100	30-44	20-22	D38Z
Glue, ground ⅛-inch and under	40	30-44		B36
Glue, pearl	40	25	11	C25
Glue, vegetable, powdered	40	30-44		
Gluten meal	40	30-44		B35P
Grain, distillery, spent, dry	30	30-44		E35WY
Grain, distillery, spent, wet	40-60	45		C45V
Granite, ½-inch screenings	80-90	20-29		C27
Granite, 1½- to 3-inch lumps	85-90	20-29		D27
Granite, broken	95-100	30-44		D37
Graphite, flake	40	30-44		C35
Graphic flour	28	20-29		A25
Graphite ore	65-75	30-44		D37
Grass seed	10-12	30-44		B35NY

* May vary considerably — consult a CEMA member

Characteristics and Conveyability of Bulk Materials

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Gravel, bank run	90-100	38	20	
Gravel, dry, sharp	90-100	30-44	15-17	D37
Gravel, pebbles	90-100	30	12	D36
Gypsum, ½-inch screenings	70-80	40	21	C36
Gypsum, 1½- to 3-inch lumps	70-80	30	15	D36
Guano, dry	70	20-29		B26
Hominy	37-50	30-44		C35
Hops, spent, dry	35	45		E45
Hops, spent, wet	50-55	45		E45T
Ice, crushed	35-45	19		D16
Ilmenite ore	140-160	30-44		B37
Iron ore	100-200	35	18-20	*D36
Iron ore pellets	116-130	30-44	13-15	D37Q
Iron sponge	100-135	30-44		
Iron sulphate	50-75	30-44		C35
Iron sulfide	120-135	30-44		D36
Kaolin clay, 3 inches and under	63	35	19	D36
Lactose	32	30-44		A35PX
Lead arsenate	72	45		B45R
Lead carbonate	240-260	30-44		A36MR
Lead ores	200-270	30	15	*B36RT
Lead oxides	60-150	45		B45
Lead oxides, pulverized	200-250	30-44		A36
Lead silicate, granulated	230	40		B36
Lead sulfate, pulverized	184	45		B46
Lead sulfide	240-260	30-44		A36
Lignite, air-dried	45-55	30-44		*D35
Lime, ground, ⅛ inch and under	60-65	43	23	B35X
*Lime, hydrated, ⅛ inch and under	40	40	21	B35MX
Lime, hydrated, pulverized	32-40	42	22	A35MXY
Lime, pebble	53-56	30	17	D35
Limestone, agricultural, ⅛ inch and under	68	30-44	20	B36
Limestone, crushed	85-90	38	18	C36X

* May vary considerably — consult a CEMA member

Effect of Inclines and Declines

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Linseed cake, pea size	50	30-44		C35W
Linseed meal	27	34	20	B35
Magnesium chloride	33	40		C45
Magnesium sulfate	40-50	30-44		
*Malt, dry, ground, ½ inch and under	22	30-44		B35NR
Malt, dry, whole	27-30	20-29		C25N
Malt, wet or green	60-65	45		C45
Malt, meal	36-40	30-44		B35
Manganese dioxide	80			*
Manganese ore	125-140	39	20	*D37
Manganese oxide	120	30-44		A36
Manganese sulfate	70	30-44		C37
Marble, crushed, ½ inch and under	80-95	30-44		D37
Marl	80	30-44		C37
Meat scraps	50-55	30-44		E35VW
Mica, flakes	17-22	19		B16MY
Mica, ground	13-15	34	23	*B36
Milk, dried, flaked	5-6	30-44		B35MPY
Milk, dry powder	36	45		B45P
Milk, malted	30-35	45		A45PX
Milk, whole, powdered	20	30-44		B35PUXY
Milk scale	100-125	45		E46T
Milo maize	56	30-44		C35N
*Molybdenite, powdered	107	40	25	B35
Molybdenum ore	107	40		B36
Monosodium phosphate	50	30-44		B36
Mustard seed	45-48	20-29		B25N
Nephelene syenite	90-105	30-44		B36
Niacin	35	30-44		B36
Nickel—cobalt, sulfate ore	80-150	30-44		*D37T
Oats	26-35	21	10	C25M
Oats, rolled	19-24	30-44		C35NY
Oil cake	48-50	45		D45W
Oxalic acid crystals	60	30-44		B35SU

* May vary considerably — consult a CEMA member

Characteristics and Conveyability of Bulk Materials

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Oyster shells, ground, under ½ inch	50-60	30-44		C36T
Oyster shells, whole	80	30-44		D36TV
Paper pulp stock	40-60	19		*E15MV
Peanuts, in shells	15-24	30-44		D35Q
Peanuts, shelled	35-45	30-44		C35Q
Peas, dried	45-50			C15NQ
Petroleum coke (see coke)	—			—
Phosphate, acid, fertilizer	60	26	13	B25T
Phosphate, triple super, ground fertilizer	50-55	45	30	B45T
Phosphate rock, broken, dry	75-85	25-29	12-15	D26
Phosphate rock, pulverized	60	40	25	B36
Polyethylene pellets	35	23		B25PQ
Polystyrene pellets	35	23		B25PQ
Polyvinyl chloride	20-30	45		A45KT
Potash (muriate), dry	70	20-29		B27
Potash (muriate), mine run	75	30-44		D37
Potash salts, sylvite, etc.	80	20-29		B25T
Potassium carbonate	51	20-29		B26
Potassium chloride, pellets	120-130	30-44		C36T
Potassium nitrate	76-80	20-29		C26T
Potassium sulfate	42-48	45		B36X
Pumice, ⅛ inch and under	40-45	45		B47
Pyrites, iron, 2- to 3-inch lumps	135-145	20-29		D26T
Pyrites, pellets	120-130	30-44		C36T
Quartz, ½ inch screenings	80-90	20-29		C27Z
Quartz, 1½- to 3-inch lumps	85-95	20-29		D27Z
Rice, hulled or polished	45-48	19	8	B15
Rice, rough	36	30-44		B35M
Rice grits	42-45	30-44		B35
Rock, crushed	125-145	20-29		D26
Rock, soft, excavated with shovel	100-110	30-44	22	D36
Rubber, pelletized	50-55	35	22	D35
Rubber, reclaim	25-30	32	18	D35
Rye	42-46	23	8	B25N

* May vary considerably — consult a CEMA member

Effect of Inclines and Declines

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Rye meal	35-40	19		B15
Safflower cake	50	30-44		D35
Safflower meal	50	30-44		B35
Safflower seed	45	20-29		B25N
Salicylic acid	29			B25U
Salt, common dry, coarse	40-55		18-22	C36TU
Salt, common dry, fine	70-80	25	11	D26TUW
Salt cake, dry, coarse	85	36	21	B36TW
Salt cake, dry, pulverized	60-85	20-29		B26NT
Saltpeter	80	30-44		A35T
Sand, bank, damp	105-130	45	20-22	B47
Sand, bank, dry	90-110	35	16-18	B37
Sand, core	65	41	26	B35X
Sand, foundry, prepared	80-90	30-44	24	B37
Sand, foundry, shakeout	90-100	39	22	D37
Sand, silica, dry	90-100	20-29	10-15	B27
Sandstone, broken	85-90	30-44		D37
Sawdust	10-13	36	22	*B35
Sesame seed, dry	27-41	20-29		B25N
Sewage (sludge)	40-50	20-29		E25TW
Sewage sludge, dried	45-55	30-44		B36
Sewage sludge, moist	55	30-44		B36
Shale, broken	90-100	20-29		D26QZ
Shale, crushed	85-90	39	22	C36
Shellac	80	45		C45
Shellac, powdered or granulated	31			B35PY
Silica gel (silica acid), dry	45	30-44		C37U
Sinter	100-135	35		*D37
Slag, blast furnace, crushed	80-90	25	10	A27
Slag, furnace, granular, dry	60-65	25	13-16	C27
Slag, furnace, granular, wet	90-100	45	20-22	B47
Slate, crushed, ½ inch and under	80-90	28	15	C36
Slate, 1½- to 3-inch lumps	85-95			D26
Soap beads or granules	15-25	30-44		C35Q
Soda ash, briquettes	50	22	7	C26
Soda ash, heavy	55-65	32	19	B36

* May vary considerably — consult a CEMA member

Characteristics and Conveyability of Bulk Materials

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Soda ash, light	20-35	37	22	A36Y
Sodium aluminate, ground	72	30-44		B36
Sodium aluminum sulfate	75	30-44		A36
Sodium antimonate, crushed	49	31		C36
Sodium nitrate	70-80	24	11	*D25
Sodium phosphate	50-65	37		B36
Sodium sulfite, dry	96	45		B45X
Sorghum seed	32-52	30-44		B36
Soybeans, cracked	30-40	35	15-18	C36NW
Soybeans, whole	45-50	21-28	12-16	C27NW
Soybean cake, over ½ inch	40-43	32	17	D35W
Soybean flakes, raw	20-26	30-44		C35Y
Soybean meal, cold	40	32-37	16-20	B35
Soybean meal, hot	40	30-44		B35T
Starch	25-50	24	12	*B25
Steel chips, crushed	100-150	30-44		D37WZ
Steel trimmings	75-150	35	18	E37V
Sugar, raw, cane	55-65	45		B46TX
Sugar, refined, granulated, dry	50-55	30-44		B35PU
Sugar, refined, granulated, wet	55-65	30-44		C35X
Sugar, beet pulp, dry	12-15	20-29		C26
Sugar, beet pulp, wet	25-45	20-29		C26X
Sugar cane, knifed	15-18	45		E45V
Sulfate, crushed, ½ inch and under	50-60	30-44	20	C35NS
Sulfate, powdered	50-60	30-44	21	B35NW
Sulfate, 3 inch and under	80-85	30-44	18	D35NS
Sunflower seed	19-38	20		C25
Taconite, pellets	116-130	30-44	13-15	D37Q
Talc, ½-inch screenings	80-90	20-29		C25
Talc, 1½- to 3-inch lumps	85-95	20-29		D25
Timothy seed	36	20-29		B25NY
Titanium dioxide	140	30-44		B36
Titanium sponge	60-70	45		E47
Tobacco leaves, dry	12-14	45		E45QV
Tobacco scraps	15-25	45		D45Y
Tobacco stems	15	45		E45Y

* May vary considerably — consult a CEMA member

Table 3-3. Material characteristics and weight per cubic foot.

Material	Average weight (lbs/cu ft)	Angle of repose (degrees)	Recommended maximum inclination (degrees)	Code
Traprock, ½-inch screenings	90-100	30-44		C37
Traprock, 2- to 3-inch lumps	100-110	30-44		D37
Tricalcium phosphate	21-50	45+		A45
Trisodium phosphate	60			D36
Trisodium phosphate, granular	60	30-44	11	B35
Trisodium phosphate, pulverized	50	40	25	B35
Urea pills, dry	43-46	25		B25
Vermiculite, expanded	16	45		C45Y
Vermiculite ore	70-80		20	D36Y
Walnut shells, crushed	35-45	30-44		B37
Wheat	45-48	28	12	C25N
Wheat, cracked	35-45	30-44		B35N
Wheat germ, dry	18-28	20-29		B25
White lead	75-100	30-44		A36MR
Wood chips	10-30	45	27	E45WY
Wood chips, hogged, fuel	15-25	45		D45
Wood shavings	8-15			E45V
Zinc concentrates	75-80			B26
Zinc ore, crushed	160	38	22	*
Zinc ore, roasted	110	38		C36
Zinc oxide, heavy	30-35	45-55		A45X
Zinc oxide, light	10-15	45		A45XY

* May vary considerably — consult a CEMA member

Capacities, Belt Widths, and Speeds

Belt widths

Lump size considerations

Belt speeds

Belt conveyor capacities

Troughed belt load areas — standard edge distance

Flat belt load areas — standard edge distance

Belt conveyor capacity tables and their use

Belt Widths

The width of conveyor belts customarily is expressed in inches. The belt widths treated in this manual — and which are available from conveyor belt manufacturers in the United States — are as follows: 18, 24, 30, 36, 42, 48, 54, 60, 72, 84, and 96 inches.

Generally, for a given speed, the belt width and the belt conveyor capacity increase together. However, the width of the narrower belts may be governed by the size of lumps to be handled. Belts must be wide enough so that any combination of prevailing lumps and finer material does not load the lumps too close to the edge of the conveyor belt. Also, the inside dimensions of loading chutes and the distance between skirtboards must be sufficient to pass various combinations of lumps without jamming.

Lump Size Considerations

The lump size influences the belt specifications and the choice of carrying idlers. There is also an empirical relationship between lump size and belt width.

The recommended maximum lump size for various belt widths is as follows: For a 20-degree surcharge, with 10 percent lumps and 90 percent fines, the recommended maximum lump size is one-third of the belt width ($b/3$). With all lumps and no fines, the recommended maximum lump size is one-fifth of the belt width ($b/5$).

For a 30-degree surcharge, with 10 percent lumps and 90 percent fines, the recommended maximum lump size is one-sixth of the belt width ($b/6$). With all lumps and no fines, the recommended maximum lump size is one-tenth of the belt width ($b/10$).

Another way to determine belt width for a specific lump size is illustrated in Figure 4.1. This simple chart shows the belt width necessary for a given size lump, for various proportions of lumps and fines, and for various surcharge loadings.

Belt Speeds

Suitable belt conveyor speeds depend largely upon the characteristics of the material to be conveyed, the capacity desired, and the belt tensions employed.

Powdery materials should be conveyed at speeds low enough to minimize dusting, particularly at the loading and discharge points. Fragile materials also limit belt speeds. Low speeds are necessary so that degradation of fragile materials will not occur at the loading and discharge points, as the conveyor belt and the material being carried move over the idlers.

Heavy, sharp-edged materials should be carried at moderate speeds. This is good practice because the sharp edges are likely to wear the belt cover unduly, particularly if the loading velocity of the material in the direction of belt travel is appreciably lower than the belt speed.

General recommendations for maximum speeds of belt conveyors are shown in Table 4-1.

Under favorable loading and transfer conditions, for troughed conveyor belts wider than 30 inches, speeds in excess of those specified in Table 4-1 are used for fine

materials, damp sand, coal, earth with no large lumps, and crushed stone. An increase in belt speed permits decreases in belt width and tension. However, these benefits must be weighed against the possible disadvantage of increased belt wear, material degradation, windage losses, lump impact on carrying idlers, and generally reduced life of all conveyor components.

Consult a CEMA member company when considering operation at these higher speeds.

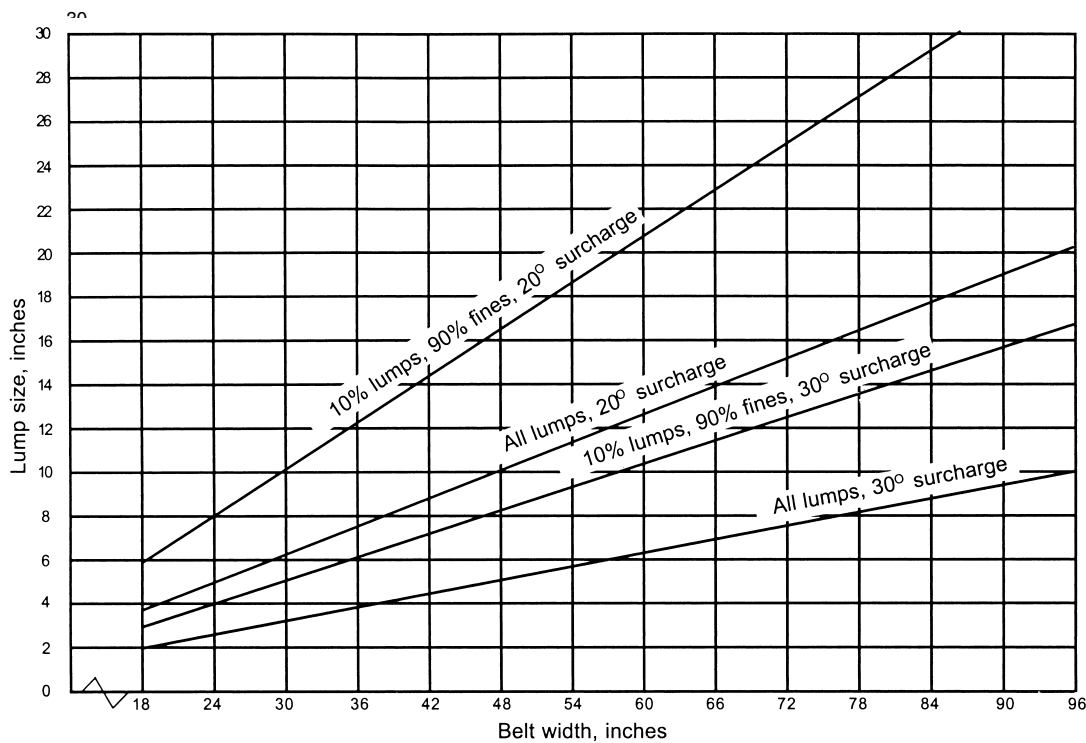


Figure 4.1 Belt width necessary for a given lump size. Fines: no greater than 1/10 maximum lump size.

The design of the loading area and the discharge of the material over the head pulley also must be considered when choosing the belt speed. If the material is dry and fine and the belt velocity is high, dusting of the material may be intolerable. Also, if the material is heavy or contains large lumps, or if the particle edges are angular and sharp, a high velocity of discharge may cause undue wear on the discharge or transfer chutes.

Belt Conveyor Capacities

For a given speed, belt conveyor capacities increase as the belt width increases. Also, the capacity of a belt conveyor depends on the surcharge angle and on the inclination of the side rolls of three-roll troughing idlers.

The nominal cross section of the material on a belt is measured in a plane normal to the belt. On an inclined or declined conveyor, the material tends to conform to its

surcharge angle as measured in a vertical plane. This decreases the area, A_s , as the cosine of the angle of conveyor slope. See Figure 4.2. However, in most cases, the actual loss of capacity is very small.

Assuming a uniform feed to the conveyor, the cross-sectional area of the load on the conveyor belt is the determinant of the belt conveyor capacity. In this manual, the cross-sectional area is based upon the following two conditions. First, the material load on the troughed belt does not extend to the belt edges. The distance from the edges of the material load to the edges of the belt is set at “standard edge distance,” which is defined as $0.055b + 0.9$ inch, where b is the width of the belt in inches. Throughout this manual, standard edge distance is presumed to be in effect unless otherwise specified. Second, the top of the load of the material is the arc of a circle tangent, at the edges of the load, to the surcharge angle of loading.

Table 4-1. Recommended maximum belt speeds.

Material Being Conveyed	Belt Speeds (fpm)	Belt Width (inches)
Grain or other free-flowing, nonabrasive material	500	18
	700	24-30
	800	36-42
	1000	48-96
Coal, damp clay, soft ores, overburden and earth, fine-crushed stone	400	18
	600	24-36
	800	42-60
	1000	72-95
Heavy, hard, sharp-edged ore, coarse-crushed stone	350	18
	500	24-36
	600	Over 36
Foundry sand, prepared or damp; shake-out sand with small cores, with or without small castings (not hot enough to harm belting)	350	Any width
Prepared foundry sand and similar damp (or dry abrasive) materials discharged from belt by rubber-edged plows	200	Any width
Nonabrasive materials discharged from belt by means of plows	200, except for wood pulp, where 300 to 400 is preferable	Any width
Feeder belts, flat or troughed, for feeding fine, nonabrasive, or mildly abrasive materials from hoppers and bins	50 to 100	Any width

Troughed Belt Load Areas—Standard Edge Distance

Referring to Figure 4.2, the area of load cross section is divided into two parts. One is the trapezoidal area, A_b ; the other is the circular segment area, A_s , which is termed the surcharge area. The sum of these two areas ($A_b + A_s$) equals A_t , which is the total cross-sectional area.

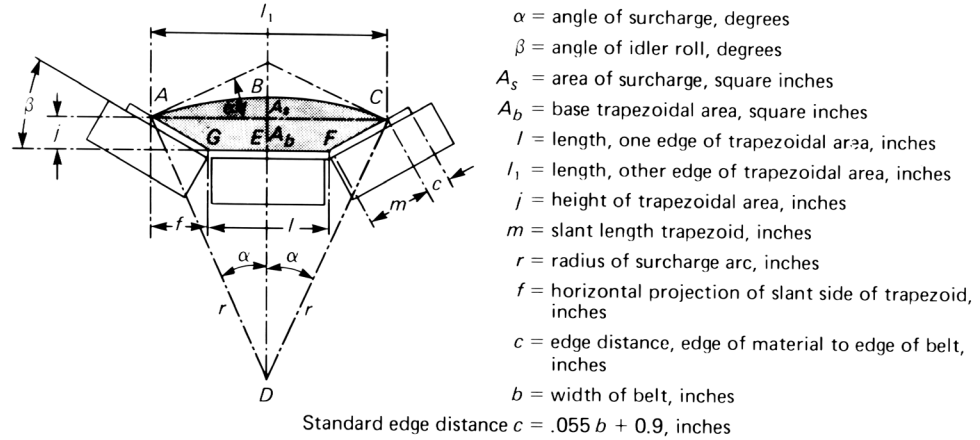


Figure 4.2 Area of load cross section.

Based on an analysis of the three-equal-roll troughing idlers of eight manufacturers, the length of the flat surface of the center roll averages $0.371b$, where b is the belt width in inches. Graphical full-scale analysis of a 5-ply belt with $1/8$ -inch and $1/32$ -inch covers, lying on an average three-equal-roll troughing idler, indicates that the flat distance on the belt-carrying surface over the center idler roll is $1/4$ -inch greater than center roll length. So:

Trapezoidal Area, A_b

$$1. \text{ Area trapezoid } (AECFG) A_b = \left(\frac{l + l_1}{2} \right) j$$

$$2. \text{ Width belt, } b = l + 2m + 2c$$

$$3. l_1 = l + 2f$$

$$f = m \cos \beta$$

$$l = 0.371b + 0.25$$

$$c = 0.055b + 0.9$$

$$b = 0.371b + 0.25 + 2m + 2(0.055b + 0.9)$$

$$2m = b - 0.418b - 2.05$$

$$m = 0.2595b - 1.025$$

$$f = m \cos \beta = (0.2595b - 1.025) \cos \beta$$

$$2f = 2(0.2595b - 1.025) \cos \beta$$

$$l_1 = 0.371b + 0.25 + 2(0.2595b - 1.025) \cos \beta$$

$$4. \frac{l + l_1}{2} = \frac{0.371b + 0.25 + 0.371b + 0.25 + 2(0.2595b - 1.025)\cos\beta}{2}$$

$$= 0.371b + 0.25 + (0.2595b + 1.025) \cos\beta$$

$$5. j = m \sin\beta$$

$$j = (0.2595b - 1.025) \sin\beta$$

$$6. \text{Area of Trapezoid } A_b = \left(\frac{l + l_1}{2}\right)j$$

or

$$[0.371b + 0.25 + (0.2595b - 1.025) \cos\beta] \times$$

$$[0.2595b - 1.025) \sin\beta]$$

Circular segment (surcharge) area, A_s

$$7. \text{Area whole sector } (ABCD) = \frac{\pi r^2 2\alpha}{360}$$

$$8. \text{Area triangle } (AECD) = \frac{r^2 \sin 2\alpha}{2}$$

$$9. \text{Area segment } (ABCE)A_s = \frac{\pi r^2 \alpha}{360} - \frac{r^2 \sin 2\alpha}{2}$$

or

$$\text{Area } A_s = r^2 \left(\frac{\pi\alpha}{180} - \frac{\sin 2\alpha}{2} \right)$$

$$10. r = \frac{0.5l_1}{\sin\alpha} = \frac{l_1}{2\sin\alpha}$$

$$r = \frac{0.371b + 0.25 + 2(0.2595b - 1.025)\cos\beta}{2\sin\alpha}$$

$$= \frac{0.1855b + 0.125 + (0.2595b - 1.025)\cos\beta}{\sin\alpha}$$

$$11. A_s = \left(\frac{0.1855b + 0.125 + (0.2595b - 1.025)\cos\beta}{\sin\alpha} \right)^2 \times \left(\frac{\pi\alpha}{180} - \frac{\sin 2\alpha}{2} \right)$$

$$12. \text{Total Area, } A_t(ft^2) = \frac{A_b + A_s}{144}$$

Flat Belt Load Areas—Standard Edge Distance

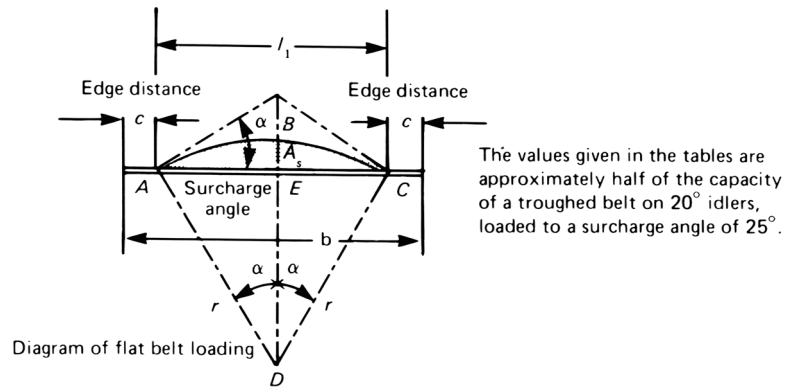


Figure 4.3. Flat belt loading.

$$1. \text{ Area sector } (ABCD) = \frac{2\pi r^2 \alpha}{360} = \frac{\pi r^2 \alpha}{180}$$

$$\text{Area triangle } (CDE) = \frac{(r \cos \alpha)(r \sin \alpha)}{2} = \frac{r^2 \sin 2\alpha}{4}$$

$$\text{Area triangle } (ACD) = \frac{2r^2 \sin 2\alpha}{4} = \frac{r^2 \sin 2\alpha}{2}$$

$$A_s = \text{Area Sector } (ABCE) = \frac{\pi r^2 \alpha}{180} - \frac{r^2 \sin 2\alpha}{2} = r^2 \left(\frac{\pi \alpha}{180} - \frac{\sin 2\alpha}{2} \right)$$

$$2. \quad l_1 = 2EC = 2r \sin \alpha \quad r = \frac{l_1}{2 \sin \alpha}$$

For standard edge distance, $c = 0.055b + 0.9\text{inch}$

$$l_1 = b - 2c = b - 2(0.055b + 0.9) = 0.890b - 1.8$$

$$r = \frac{0.890b - 1.8}{2 \sin \alpha} = \frac{0.445b - 0.9}{\sin \alpha}$$

$$3. \quad A_s = \left(\frac{0.445b - 0.9}{\sin \alpha} \right)^2 \times \left(\frac{\pi \alpha}{180} - \frac{\sin 2\alpha}{2} \right)$$

$$4. \text{ For flat belt, Total Area } A_t(f t^2) = \frac{A_s}{144}$$

Belt Conveyor Capacity Tables and Their Use

Troughed and flat belt conveyor capacities are detailed in Tables 4-2 through 4-5. These tables are set up for 20 degrees, 35 degrees, and 45 degrees troughing idler shapes and for flat belts; for various degrees of surcharge angles which correspond to the slumping characteristics of the materials to be conveyed; and for belt speeds of 100 fpm.

To make the best use of these tables, the following eight steps should be taken:

1. Referring to Tables 3-1 and 3-3, determine the surcharge angle of the material. The surcharge angle, on the average, will be 5 degrees to 15 degrees less than the angle of repose.
2. Refer to Table 3-3 to determine the density of the material in pounds per cubic foot (lb/ft^3).
3. Choose the idler shape suited to the material and to the conveying problem. Refer to Chapter 5.
4. Refer to Table 4-1, "Recommended Maximum Belt Speeds." Select a suitable conveyor belt speed.
5. Convert the desired tonnage per hour (tph) to be conveyed to the equivalent in cubic feet per hour (ft^3/hr).

$$\text{ft}^3/\text{hr} = \frac{\text{tph} \times 2000}{\text{material density}}$$

6. Convert the desired capacity in cubic feet per hour to the equivalent capacity at a belt speed of 100 fpm.

$$\text{Capacity (equivalent)} = (\text{ft}^3/\text{hr}) \times \left(\frac{100}{\text{actual belt speed (fpm)}} \right)$$

7. Using the equivalent capacity so found, refer to Tables 4-2 through 4-5 and find the appropriate belt width.
8. If the material is lumpy, check the selected belt width against the curves in Figure 4.1. The lump size may determine the belt width, in which case the selected belt speed may require revision.

Table 4-2. 20-degree troughed belt—three equal rolls standard edge distance = $0.055b + 0.9$ inch.

Belt Width (Inches)	A_t - Cross Section of Load (ft ²)							Capacity at 100 FPM (ft ³ /hr)						
	Surcharge Angle							Surcharge Angle						
	0°	5°	10°	15°	20°	25°	30°	0°	5°	10°	15°	20°	25°	30°
18	.089	.108	.128	.147	.167	.188	.209	537	653	769	886	1005	1128	1254
24	.173	.209	.246	.283	.320	.359	.399	1041	1258	1477	1698	1924	2155	2394
30	.284	.343	.402	.462	.522	.585	.649	1708	2060	2414	2772	3137	3511	3897
36	.423	.509	.596	.684	.774	.866	.960	2538	3057	3579	4107	4645	5196	5765
42	.588	.708	.828	.950	1.074	1.201	1.332	3533	4250	4972	5703	6447	7210	7997
48	.781	.940	1.099	1.260	1.424	1.592	1.765	4691	5640	6594	7560	8544	9552	10592
54	1.002	1.204	1.407	1.613	1.822	2.037	2.258	6013	7225	8444	9678	10935	12223	13552
60	1.249	1.501	1.753	2.009	2.270	2.537	2.812	7498	9006	10522	12057	13621	15223	16876
72	1.826	2.192	2.560	2.933	3.312	3.701	4.102	10961	13155	15364	17599	19876	22210	24617
84	2.513	3.014	3.519	4.030	4.551	5.085	5.635	15079	18089	21119	24186	27309	30511	33813
96	3.308	3.967	4.631	5.302	5.986	6.687	7.411	19850	23806	27787	31816	35921	40128	44466

Capacities, Belt Widths, and Speeds

Table 4-3. 35-degree troughed belt—three equal rolls standard edge distance = $0.055b + 0.9$ inch.

Belt Width (Inches)	A_L - Cross Section of Load (ft ²)							Capacity at 100 FPM (ft ³ /hr)						
	Surcharge Angle							Surcharge Angle						
	0°	5°	10°	15°	20°	25°	30°	0°	5°	10°	15°	20°	25°	30°
18	.144	.160	.177	.194	.212	.230	.248	864	964	1066	1169	1274	1381	1492
24	.278	.309	.341	.373	.406	.440	.474	1668	1857	2048	2241	2438	2640	2847
30	.455	.506	.557	.609	.662	.716	.772	2733	3039	3346	3658	3975	4300	4636
36	.676	.751	.826	.903	.980	1.060	1.142	4058	4508	4961	5419	5886	6364	6857
42	.940	1.044	1.148	1.254	1.361	1.471	1.585	5644	6266	6891	7524	8169	8830	9511
48	1.248	1.385	1.523	1.662	1.804	1.949	2.099	7491	8312	9138	9974	10825	11698	12598
54	1.599	1.774	1.950	2.128	2.309	2.494	2.686	9598	10646	11700	12768	13855	14969	16118
60	1.994	2.211	2.429	2.651	2.876	3.107	3.345	11966	13269	14580	15906	17257	18642	21058
72	2.913	3.229	3.547	3.869	4.197	4.532	4.879	17484	19378	21285	23215	25182	27196	29275
84	4.007	4.440	4.876	5.317	5.766	6.226	6.701	24043	26641	29256	31902	34597	37360	40210
96	5.274	5.842	6.415	6.994	7.584	8.189	8.812	31645	35058	38490	41966	45506	49134	52876

Table 4-4. 45-degree troughed belt—three equal rolls standard edge distance = $0.055b + 0.9$ inch.

Belt Width (Inches)	A_L - Cross Section of Load (ft ²)							Capacity at 100 FPM (ft ³ /hr)						
	Surcharge Angle							Surcharge Angle						
	0°	5°	10°	15°	20°	25°	30°	0°	5°	10°	15°	20°	25°	30°
18	.170	.184	.199	.214	.230	.245	.262	1021	1109	1198	1289	1380	1475	1572
24	.327	.355	.383	.411	.439	.469	.499	1967	2132	2299	2467	2638	2814	2996
30	.536	.580	.625	.670	.716	.763	.812	3218	3484	3752	4023	4299	4581	4873
36	.795	.860	.926	.992	1.060	1.129	1.200	4775	5165	5558	5955	6360	6775	7204
42	1.106	1.195	1.286	1.377	1.470	1.566	1.664	6636	7175	7717	8265	8824	9397	9987
48	1.467	1.585	1.704	1.825	1.948	2.074	2.204	8803	9514	10229	10953	11690	12445	13224
54	1.879	2.030	2.182	2.336	2.492	2.653	2.819	11276	12182	13094	14017	14957	15921	16915
60	2.342	2.529	2.718	2.909	3.104	3.303	3.509	14053	15179	16312	17458	18626	19823	21059
72	3.420	3.693	3.967	4.245	4.528	4.818	5.117	20524	22160	23807	25473	27171	28910	30705
84	4.702	5.076	5.452	5.832	6.220	6.617	7.027	28216	30458	32713	34997	37322	39706	42165
96	6.188	6.678	7.172	7.671	8.180	8.701	9.239	37128	40071	43032	46029	49081	52210	55437

Belt Conveyor Capacity Tables and Their Use

Table 4-5. Flat belt capacity standard edge distance = $0.055b + 0.9$ inch.

Belt Width (Inches)	A_s - Cross Section of Load (ft ²)							Capacity at 100 FPM (ft ³ /hr)						
	Surcharge Angle							Surcharge Angle						
	0°	5°	10°	15°	20°	25°	30°	0°	5°	10°	15°	20°	25°	30°
18		.020	.041	.062	.083	.105	.127	123	246	372	498	630	762	
24		.039	.077	.117	.157	.198	.241	232	466	702	942	1190	1444	
30		.063	.126	.190	.255	.321	.390	376	756	1137	1527	1928	2340	
36		.092	.185	.280	.376	.474	.575	555	1113	1677	2253	2844	3450	
42		.130	.257	.387	.520	.656	.796	768	1540	2322	3120	3936	4776	
48		.169	.340	.512	.688	.868	1.053	1016	2037	3072	4126	5208	6318	
54		.216	.434	.654	.879	1.109	1.346	1298	2604	3927	5273	6654	8076	
60		.269	.540	.814	1.093	1.380	1.675	1614	3240	4885	6560	8278	10050	
72		.392	.786	1.186	1.593	2.010	2.440	2353	4720	7116	9558	12060	14640	
84		.538	1.080	1.628	2.186	2.758	3.349	3229	6478	9767	13117	16550	20091	
96		.707	1.419	2.139	2.873	3.625	4.400	4243	8514	12835	17238	21750	26404	

Belt Conveyor Idlers

Idler Requirements
Idler Classifications
General Types of Belt Conveyor Idlers
Troughing Carrying Idlers
Impact Idlers
Belt Training Idlers, Carrying
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Idler Selection Procedure
Example Idler Selection
Belt Alignment

Idler Requirements

Important requirements for idlers are proper support and protection for the belt and proper support for the load being conveyed.

Belt conveyor idlers for bulk materials are designed to incorporate rolls with various diameters. The rolls are fitted with antifriction bearings and seals, and are mounted on shafts.

Frictional resistance of the idler roll influences belt tension and, consequently, the horsepower requirement. Roll diameter, bearing design, and seal requirements constitute the major components affecting frictional resistance.

This manual does not discuss the relative merits of the various antifriction bearings used, nor the merits of the seals to protect these bearings from dirt and moisture and to retain the lubricant. Each belt conveyor idler manufacturer chooses a particular bearing and seal arrangement. Much ingenuity has been exercised by these idler manufacturers to provide dependable idlers.

Idler Classifications

Selection of the proper roll diameter and size of bearing and shaft is based on the type of service, operating condition, load carried, and belt speed. For ease and accuracy of idler selection, the various idler designs can be grouped into classifications as shown in Table 5-1.

Table 5-1. Idler classification.

Classification	Former Series Number	Roll Diameter (inches)	Belt Width (inches)	Description
A4	STANDARD WITHDRAWN			
A5	OCTOBER 1, 1996			
B4	II	4"	18" through 48"	Light Duty
B5	II	5"	18" through 48"	"
C4	III	4"	18" through 60"	Medium Duty
C5	III	5"	18" through 60"	"
C6	IV	6"	24" through 60"	"
D5	None	5"	24" through 72"	"
D6	None	6"	24" through 72"	"
E6	V	6"	36" through 96"	Heavy Duty
E7	VI	7"	36" through 96"	"

General Types of Belt Conveyor Idlers

There are two basic types of belt conveyor idlers: carrying idlers, which support the loaded run of the conveyor belt; and return idlers, which support the empty return run of the conveyor belt. See Figures 5.1 through 5.3.

Carrying Idlers

Carrying idlers are of two general configurations. One is used for troughed belts and usually consists of three rolls. The two outer rolls are inclined upward; the center roll is horizontal. The other configuration is used for supporting flat belts. This idler generally consists of a single horizontal roll positioned between brackets which attach directly to the conveyor frame.

Return Idlers

Return idlers usually are horizontal rolls, positioned between brackets which normally are attached to the underside of the support structure on which the carrying idlers are mounted. Two-roll “Vee” return idlers are also used for better training and higher load ratings.

Figure 5.1 35° troughing idler.

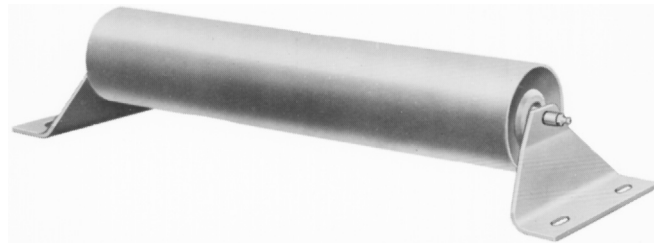
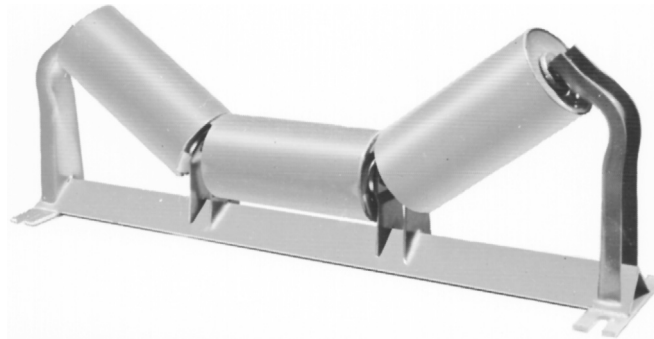


Figure 5.2 Flat belt idler.



Figure 5.3 Return belt idler.

Troughing Carrying Idlers

Due to the increased cross sectional fill depth, troughed belts can carry far greater tonnages than flat belts of the same width and speed. Troughing carrying idlers are sometimes referred to as troughers or carriers, and are the most common type of belt conveyor idler used. Rolls are commonly fabricated from steel tube with end disc (bearing housings) welded to the tube ends. Rolls made from high molecular weight polyethylene are used where abrasion, material buildup, or corrosion create short shell life with steel roll idlers.

Historically, 20-degree troughing idlers have longer application histories than either 35-degree or 45-degree troughing idlers. As conveyor belt design technology has advanced, allowing greater transverse flexibility, 35-degree troughing idlers have become the most widely used type of troughing idler.

Troughing idlers are made as either in-line or offset center roll design. Three roll, in-line, equal length roll troughing idlers are most commonly used and offer the best all-around shape to carry a maximum load cross section.

Figure 5.4 shows a 20-degree in-line troughing idler. An offset center roll (commonly known as a grain idler) idler is shown in Figure 5.5. This idler, utilizing the wing or side rolls located in an adjacent parallel plane to the center roll, is used in either the grain industry where thin belts are used or underground mining where height clearance is minimal. A picking and feeder (or picker) idler is shown in Figure 5.6. This design uses a long (extended) center roll and short side rolls inclined at 20 degrees to allow maximum product dispersal for inspection or sortation. Unequal length roll troughing idlers are also available where the side rolls are inclined at 35 degrees or 45 degrees.



Figure 5.4 20° troughing idler.

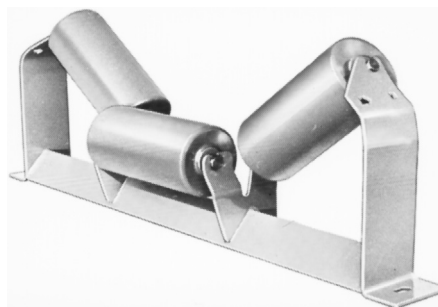


Figure 5.5 35° offset troughing idler.



Figure 5.6 20° picking belt idler.

Impact Idlers

Impact troughing idlers, sometimes referred to as “cushion idlers,” are used at loading points where impact resulting from lump size, material density, and height of material free fall could seriously damage the belt, if the belt were rigidly supported.

Many types of impact troughing idler are available using pneumatic tires, semi-pneumatic tires, heavy rubber covers vulcanized to steel rolls, and individual narrow discs pressed onto a steel tube. This latter type, as shown in Figure 5.7, is the most common type of construction. Each disc is made of a resilient material such as a soft (40 to 50 durometer) natural rubber, grooved and relieved to allow the rubber to move under impact. The continuous (massed) row of disc gives better support to the belt than most pneumatic or semi-pneumatic types. The resilient discs help absorb energy from impact loads, which could save the belt from impact damage. The discs are sacrificed in favor of reducing the risk of belt damage.

Impact troughing idlers with three equal length rolls have the same load rating as standard troughing idlers for a given CEMA class (series).

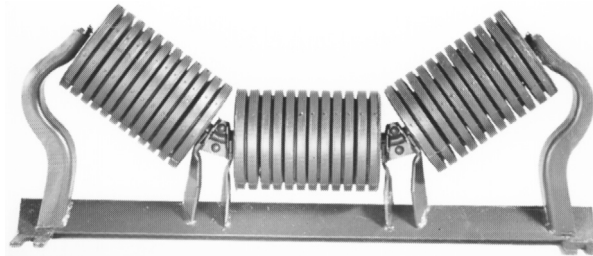


Figure 5.7 35° troughing rubber-cushion impact idler.

Flat belt impact rolls as shown in Figure 5.8 have the same load rating as a single roll return for a given CEMA class. Flat belt type impact rolls are also available in a “live shaft” design supported by pillow block style bearings. These are frequently used on heavy duty belt feeders. They have a much higher load rating and are not covered by a CEMA standard, but are available from CEMA members on an application-by-application basis.



Figure 5.8 Flat-belt rubber-cushion impact idler with fixed shaft.

Even though an impact idler provides for some cushioning under the belt to help soften the force and reduce the possibility of damage, the impacting force has to be dissipated. The magnitude of these impact loads and their dissipation is not covered in CEMA load ratings.

Impact idlers with removable end stands, reinforced frames, and other special features to aid skirtboard and belt sealing systems are available from your CEMA idler manufacturer.

Using an impact style troughing idler as a transition idler is not recommended.

Although not classified as idlers, there are numerous designs and configurations of fixed impact bars, impact saddles, impact cradles, and impact/slider beds available. These can solve some specific impact or sealing system application problems, but are not a “cure all.” These are not covered by CEMA standards, but are available from CEMA members on an application-by-application basis.

Belt Training Idlers, Carrying

The normal carrying idlers are the primary devices which control the belt alignment. No self-alignment idlers are needed under well designed, precisely assembled, and maintained belt conveyors. There are transient conditions, however, that may cause conveyor belts to become misaligned despite all efforts to assure proper installation and maintenance. For this reason, conveyor manufacturers also furnish belt training idlers to help control belt alignment in difficult situations.

The training idlers pivot about an axis vertically perpendicular to the center line of the belt, and when the belt becomes off-center, they swing about so that the axes of the rolls themselves become canted in a corrective direction. This swinging about the center pivot is accomplished in various ways usually associated with the pressure of the off-center belt against a fixed arm attached to the idler frame. See Figure 5.9.



Figure 5.9 35° troughed belt training idler.

If the belt is to be reversed, the self-aligning idlers must be of a type which will swing about their pivot in a corrective direction regardless of belt direction. Those types which depend on friction of the off-center belt to shift the idler will work in both directions of belt movement. Even with properly designed self-aligning idlers, the training of a reversing belt requires very careful alignment of all idlers and pulleys as well as leveling and alignment of the conveyor structure itself.

If belt training idlers are required, they should be spaced from 100 to 150 feet apart, and at least one training idler should be used on conveyors less than 100 feet long. Belt training idlers should not be used in areas of belt transitions.

Fixed guide rolls placed perpendicular to the edge of the conveyor belt are not generally recommended, because continuous contact with the conveyor belt edge accelerates belt edge wear, appreciably reducing belt life.

In general, the greater the belt tensions, the less effective the training idlers.

Suspended or Garland Idlers

Garland idlers are available as two-roll, three-roll, or five-roll units. Normally, two-roll units are utilized as return idlers and serve to aid in belt training due to the trough that is formed. Three-roll and five-roll units are used as carrying or impact units. Rubber discs can be utilized on three-roll units to provide additional cushioning. Five-roll units do not use rubber discs. All garland idlers are suspended from the conveyor framework by means of various devices such as hooks or chains.

The suspended design aids in belt alignment and handles large, irregular lumps because of the flexible connections and available vertical move. Five-roll designs offer a deep trough configuration and greater load capacities than conventional three-roll units. Garlands can be furnished with quick release suspensions that allow the unit to be lowered away from belt contact in the case of roll failure.

Garland design idlers can be utilized on rigid frame or wire rope supported conveyor systems. Figure 5.10 shows the available types and their configuration under off-center loads. Garland design idlers are also sometimes referred to as catenary idlers.

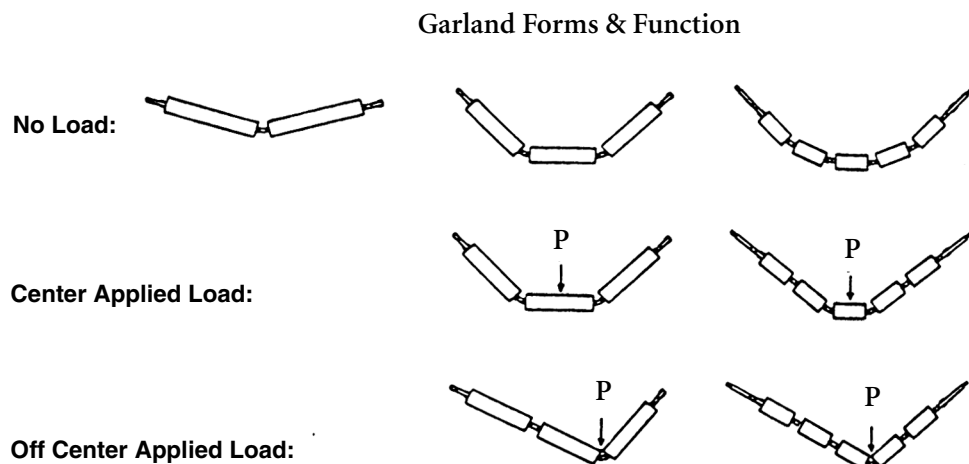


Figure 5.10 Suspended or garland idlers.

Return Idlers

These idlers are used to support the return run of the belt. They usually are suspended below the lower flanges of the stringers which support the carrying idlers. It is preferable that return idlers be mounted so that the return run of the belt is visible below the conveyor frame. Figure 5.11 illustrates a typical return idler.



Figure 5.11 Return belt idler.

Flat Return Idlers

The flat return idler consists of a long single roll, fitted at each end with a mounting bracket. Idler roll length, bracket design, and mounting-hole spacing should allow for adequate transverse belt movement without permitting the belt edges to contact any stationary part of the conveyor or its frame.

Self-Cleaning Return Idlers

An important consideration with return idler applications is the adherence of materials to the carrying surface of the belt. Such material may be abrasive and wear the shell of the return idler rolls. Or, this buildup may be sticky and adhere to the return idler rolls. A large buildup may cause misalignment of the return run of the belt.

Several styles of return idler rolls are available to overcome these difficulties. When sticky materials are a problem, rubber or urethane disc, or rubber coated helically shaped, self-cleaning return idlers can be used. Disc and helical rolls present very narrow surfaces for adhesion and thus reduce the tendency for material build up. This type of return idler sometimes is erroneously called a “belt cleaning idler.” Even though such idlers do “trackoff” material adhering to the belt surface on the return run, they do not constitute belt cleaning devices. See Figures 5.12 and 5.13. If additional abrasion resistance is required and sticky material is not the major concern, steel roll idlers with rubber polyurethane or polyethylene covering will extend the wear life of the roll. The preferred disc type return idler has massed disc at each end of the roll to provide better belt support, if the belt is running off center of structure.

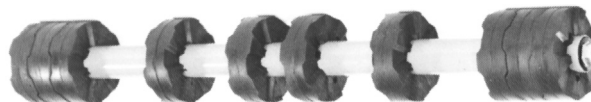


Figure 5.12 Rubber-disc return idler.

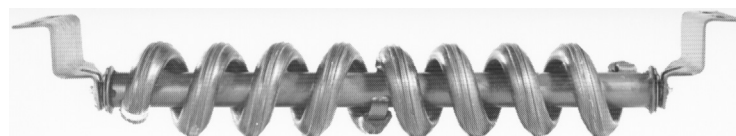


Figure 5.13 Helical or spiral self-cleaning return idler.

Two-Roll “Vee” Return Idlers

On short conveyors, it may be necessary to equip the complete return run with self-cleaning idlers. On long return belt runs, it is necessary to use these idlers only to the point where the material on the belt surface no longer will adhere to and build up on normal return idler rolls. Beyond this point, standard return idlers can be used.

Return Belt Training Idlers

Return belt idlers can be pivotally mounted to train or align the return belt in a manner similar to the training idlers previously described for the carrying run of the belt. See Figure 5.14. Training idlers for both one-way or reversible belts are available. In general, a training idler designed for use on a one-way belt travel will not work on a reversible belt.

Return belt training idlers are usually more effective than trough training idlers, due to lower belt tension.



Figure 5.14 Return training idler.

Two-Roll “Vee” Return Idlers

With the increased use of heavy, high-tension fabric and steel cable belts, the need for better support and belt training has resulted in the development of “Vee” return idlers. A basic “Vee” return idler consists of two rolls, each tilted at a 5-, 10-, or 15-degree angle. These rolls are either of the garland (suspended) or rigid design. See Figures 5.15 and 5.16. The “Vee” return idler has some training effect on the belt, while allowing greater idler spacing because of its increased load rating. The trough shape of the belt also tends to reduce or eliminate vibration along the conveyor.

“Vee” return idlers can be supplied with steel rolls, rolls coated with some type of polymer, or spaced disc of rubber, urethane, or other material.

A decrease in wear life of the roll shell or spaced disc may occur with “Vee” return idlers. This is caused by most of the belt weight contacting the roll about one-fourth of the roll length from the center line of the idler.

Figure 5.15 Two-roll “Vee” return idler, rigid design.

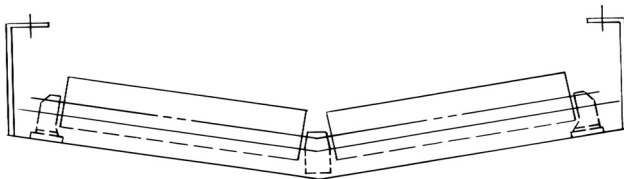


Figure 5.16 Two-roll suspended “Vee” return idler.

Idler Spacing

Factors to consider when selecting idler spacing are belt weight, material weight, idler load rating, belt sag, idler life, belt rating, belt tension, and radius in vertical curves (see Chapter 9).

More complex issues (such as belt flap or vibration stability in wind, and power usage from belt indentation, material tramping, and rolling resistance) will be affected less by idler spacing.

If too much sag of a loaded troughed belt is permitted between the troughing idlers, the material may spill over the edges of the belt. For the best design, especially on long-center troughed belt conveyors, the sag between idlers should be limited as described in Chapter 6.

Table 5-2 lists suggested normal troughing idler spacing for use in general engineering practice, when the amount of belt sag is not specifically limited. These figures on spacing should be used in conjunction with the information on sag selection in Chapter 6. Spacing is normally varied in 6-inch increments.

Some conveyor systems have been designed successfully utilizing extended idler spacing and/or graduated idler spacing. Extended idler spacing is simply greater than normal spacing. This is sometimes applied where belt tension, sag, belting strength, and idler rating permit. Advantages may be lower idler cost (fewer used) and better belt training.

Graduated idler spacing is greater than normal spacing at high tension portions of the belt. As the tension along the belt increases, the idler spacing is increased. Usually this type of spacing occurs toward and near the discharge end.

Extended and graduated spacing are not commonly used but if either is employed, care should be taken not to exceed idler load rating and sag limits during starting and stopping.

Table 5-2. Suggested normal spacing of belt idlers (S_i).*

Belt Width (inches)	Troughing Idler Spacing						Return Idlers
	Weight of Material Handled, lbs/cu ft						
	30	50	75	100	150	200	
18	5.5	5.0	5.0	5.0	4.5	4.5	10.0
24	5.0	4.5	4.5	4.0	4.0	4.0	10.0
30	5.0	4.5	4.5	4.0	4.0	4.0	10.0
36	5.0	4.5	4.0	4.0	3.5	3.5	10.0
42	4.5	4.5	4.0	3.5	3.0	3.0	10.0
48	4.5	4.0	4.0	3.5	3.0	3.0	10.0
54	4.5	4.0	3.5	3.5	3.0	3.0	10.0
60	4.0	4.0	3.5	3.0	3.0	3.0	10.0
72	4.0	3.5	3.5	3.0	2.5	2.5	8.0
84	3.5	3.5	3.0	2.5	2.5	2.0	8.0
96	3.5	3.5	3.0	2.5	2.0	2.0	8.0

* Spacing indicated in feet. Spacing may be limited by load rating of idler. See idler load ratings in Tables 5-7—5-11.

Idler Spacing

Return Idler Spacing

The suggested normal spacing of return idlers for general belt conveyor work is also given in Table 5-2. For conveyor belts with heavy carcasses, and with a width of 48 inches or more, it is recommended that the return idler spacing be determined by the use of the idler load ratings and belt sag considerations. (See Chapter 6.)

Carrying Idler Spacing at Loading Points

At loading points, the carrying idlers should be spaced to keep the belt steady and to hold the belt in contact with the rubber edging of the loading skirts along its entire length. Careful attention to the spacing of the carrying idlers at the loading points will minimize material leakage under the skirtboards and, at the same time, will also minimize wear on the belt cover.

Normally, carrying idlers in the loading zone are spaced at half (or less) the normal spacing suggested in Table 5-2.

Caution: If impact idlers are used at loading zones, impact idler ratings are no higher than standard idler ratings.

Good practice dictates that the spacing of idler rolls under the loading area be such that the major portion of the load engages the belt between idlers.

Rubber disc idlers are commonly used before, between, and at the end of impact bars. Though this spacing is directed by bar length, idler rating selection and elevation should be considered in these designs.

Troughing Idler Spacing Adjacent to Terminal Pulleys

In passing from the last troughing idler to the terminal pulley, the belt edges are stretched and tension is increased at the outer edges. If the belt edge stress exceeds the elastic limit of the carcass, the belt edge will be stretched permanently and will cause belt training difficulties. On the other hand, if the troughing idlers are placed too far from the terminal pulleys, spillage of the load is likely.

Distance is important in the change (transition) from troughed to flat form. This is especially significant when deeply troughed idlers are used.

Depending on the transition distance, one, two, or more transition type troughing idlers can be used to support the belt between the last standard troughing idler and the terminal pulley. These idlers can be positioned either at a fixed angle or at an adjustable concentrating angle. Table 5-3 shows recommended transition distances for various troughing angles, belt tensions, and types of belting. In no case should the rating of the idler be exceeded.

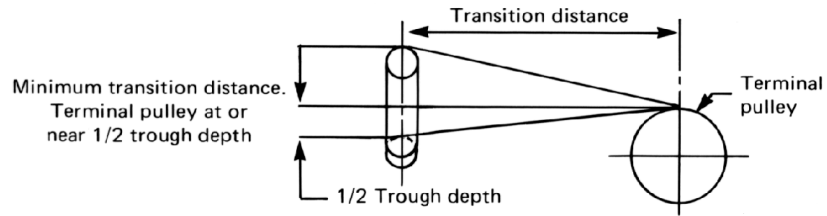


Table 5-3. A — Half trough recommended minimum transition distances.

Idle Angle	% Rated Tension	Fabric Belts	Steel Cable Belts
20°	Over 90	$0.9b$	$2.0b$
	60 to 90	$0.8b$	$1.6b$
	Less than 60	$0.6b$	$1.0b$
35°	Over 90	$1.6b$	$3.4b$
	60 to 90	$1.3b$	$2.6b$
	Less than 60	$1.0b$	$1.8b$
45°	Over 90	$2.0b$	$4.0b$
	60 to 90	$1.6b$	$3.2b$
	Less than 60	$1.3b$	$2.3b$

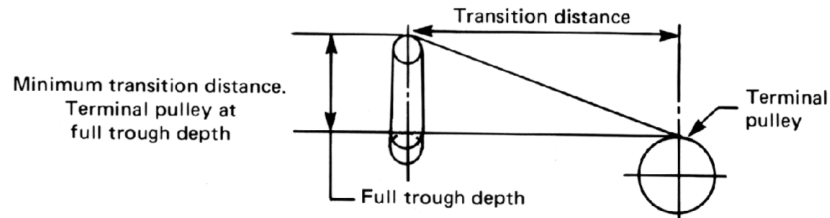


Table 5-3. B — Full trough recommended minimum transition distances.

Idler Angle	% Rated Tension	Fabric Belts	Steel Cable Belts
20°	60 to 90	$1.6b$	$3.2b$
	Less than 60	$1.2b$	$2.8b$
35°	Over 90	$3.2b$	$6.8b$
	60 to 90	$2.4b$	$5.2b$
	Less than 60	$1.8b$	$3.6b$
45°	Over 90	$4.0b$	$8.0b$
	60 to 90	$3.2b$	$6.4b$
	Less than 60	$2.4b$	$4.4b$

b = Belt width (transition distance will be in the same units as those used for b)

The Selection of Idlers

Previous to 10/1/96, CEMA ratings were based on 90,000 hours B_u (useful bearing life) at 500 rpm. B_u values were approximately 3 times L_{10} . The B_u (useful bearing life) theory was technically correct. However, L_{10} bearing life is more commonly used and accepted for bearing life calculations and rating. Previous to CEMA publication 502-1996, the CEMA idler selection procedure used idler life (K) factors to calculate an adjusted idler load. Some of these (K) factors were entirely independent of idler load and bearing L_{10} life. This procedure provided a conservative selection based on load but did not necessarily provide clear data relative to expected idler life.

Rating and Idler Life

Idler life is determined by a combination of many factors, such as seals, bearings, shell thickness, belt speed, lump size/material density, maintenance, environment, temperature, and the proper CEMA series of idler to handle the maximum calculated idler load. While bearing life is often used as an indicator of idler life, it must be recognized that the effect of other variables (e.g., seal effectiveness) may be more important in determining idler life than the bearings. Nevertheless, since bearing rating is the only variable for which laboratory tests have provided standard values, CEMA uses bearing L_{10} life as a guide for establishing idler ratings.

The definition of L_{10} for belt conveyor idlers: The basic rated life (number of operating hours at 500 rpm) based on a 90 percent statistical model which is expressed as the total number of revolutions 90 percent of the bearings in an apparently identical group of bearings subjected to identical operating conditions will attain or exceed before a defined area of material fatigue (flaking, spalling) occurs on one of its rings or rolling elements. The L_{10} life is also associated with 90 percent reliability for a single bearing under a certain load.

Tables 5.7 through 5.10 show load ratings for CEMA B, C, D, and E idlers. These load ratings are based on the following:

- CEMA B load rating based on minimum L_{10} of 30,000 hours at 500 rpm
- CEMA C load rating based on minimum L_{10} of 30,000 hours at 500 rpm
- CEMA D load rating based on minimum L_{10} of 60,000 hours at 500 rpm
- CEMA E load rating based on minimum L_{10} of 60,000 hours at 500 rpm

These loads and L_{10} life ratings are minimum ratings for CEMA rated idlers. Actual values for load ratings and L_{10} life for specific series and belt sizes supplied by CEMA manufacturers may be higher. In some cases the idler frame design could be the limiting factor for load with L_{10} life being a higher value.

Idler Selection Procedure

There are many conditions that affect idler life. Those considered in this selection procedure are:

1. Type of material handled
2. Idler load
3. Effect of load on predicted bearing L_{10} life
4. Belt speed
5. Roll diameter
6. Environmental, maintenance and other special conditions

In addition to information provided in the Idler Selection Procedure, the above items are summarized as follows:

Type of Material Handled

The characteristics of the material handled have a direct bearing on the idler selection. The weight of the material governs the idler load and spacing, and lump size modifies the effect of weight by introducing an impact factor. Table 5-6 combines the unit weight and the lump size into a group of empirical factors referred to as $K1$. Note that in the table “lump size” means the largest lump which may occasionally be carried rather than the average lump.

Lump Size Considerations

The lump size influences the belt specifications and the choice of carrying idlers. There is also an empirical relationship between lump size and belt width. The recommended maximum lump size for various belt widths is as follows:

For a 20 degree surcharge, with 10 percent lumps and 90 percent fines, the recommended maximum lump is $\frac{1}{3}$ the belt width ($bw/3$). With all lumps, the recommended maximum lump is $\frac{1}{5}$ belt width ($bw/5$).

For a 30 degree surcharge, with 10 percent lumps and 90 percent fines, the recommended maximum lump is $\frac{1}{6}$ the belt width ($bw/6$). With all lumps, maximum lump is $\frac{1}{10}$ the belt width ($bw/10$).

Idler Load

To select the proper CEMA class (series) of idler, it is necessary to calculate the idler load. This procedure is shown in “Idler Selection Procedure” (Step No. 1) for troughing idlers and (Step No. 2) for return idlers.

The idler load should be calculated for peak or maximum conditions. The belt conveyor designer should thoroughly investigate all conditions relative to calculating idler misalignment load (IML), in addition to structure misalignment. The idler height deviation between standard fixed idlers and training idlers (or other special types of idlers) must be accounted for either by idler series selection or by conveyor design and installation control.

Effect of Load on Predicted Bearing L_{10} Life

When calculated idler load (CIL) is less than CEMA load rating of series idler selected, the bearing L_{10} life will increase. Figure 5.17 (Step No. 3) shows this relationship for either a tapered roller bearing or a ball bearing idler design. This chart can be used in conjunction with the type of service or life expectancy of the conveyor system. If the specified design life of the conveyor system exceeds the CEMA L_{10} life rating at rated load, it may still meet specification based on percent of rated idler load vs. calculated idler load (CIL).

Belt Speed

Bearing life (L_{10}) is based on the number of revolutions of the bearing race. The faster the belt speed, the more revolutions per minute and consequently, a shorter life for a given number of revolutions. All CEMA L_{10} life ratings are based on 500 rpm. The following table (5-4) lists belt speed at 500 rpm for standard roll diameters.

Table 5-4. Belt speeds at 500 rpm.

Roll Diameter (inches)	Belt Speed (fpm)
4"	524
5"	654
6"	785
7"	916

Figure 5.18 (Step No. 4) shows the effect of belt speed on predicted bearing L_{10} life. However, suitable belt conveyor speeds also depend upon the characteristics of the material to be conveyed, the capacity desired, and the belt tensions employed. This subject is covered in more detail in Chapter 4.

Roll Diameter

For a given belt speed, using larger diameter rolls will increase idler bearing L_{10} . Figure 5.19 (Step No. 5) shows this relationship. In addition, since larger diameter rolls will be contacting the belt less due to a slower rpm, the wear life of the shell will be increased.

Environmental, Maintenance and Other Special Conditions

Step No. 6 in the idler selection procedure identifies conditions that will affect potential idler life. All of these conditions do not have an exact mathematical basis and therefore can be very subjective. The most important phase of this step is identifying the idler life condition for the application and then arriving at solutions to obtain maximum idler life for that application. Since idler roll configuration, type of bearing, and seal design can vary with each idler manufacturer, it is logical to state that idler life can also vary for a given environmental and maintenance condition.

Figures 5.20, 5.21, and 5.22 show general conditions which will affect idler life. Those conditions are independent of idler load but can cause idler failure before obtaining predicted L_{10} life rating. CEMA recommends contacting your CEMA idler manufacturer for assistance in establishing guidelines for "Potential Idler Life" for the various conditions shown or any unusual conditions not listed.

Special Conditions

Idler roll shell material usually used throughout the industry is electric resistance welded steel mechanical tubing. For most belt conveyor applications, this material provides sufficient idler life most economically. For severe abrasive or corrosive conditions, covered idler rolls are available in a variety of materials. CEMA has not compiled a relative wear index or corrosion compatibility index for these various materials. This information can be supplied by your CEMA idler manufacturer. However, the economic issue vs. increased life should be investigated thoroughly. Some of the generically available materials are listed below. There are numerous grades available in each of these materials which will affect performance.

1. Steel sleeves
2. Rubber lagging
3. Neoprene lagging

4. Polyethylene sleeves/rolls
5. Carboxylated nitrile
6. Urethane
7. Ceramic

Another consideration for increasing shell wear life is to use thicker metal shells. Some idler manufacturers customarily supply larger diameter rolls with thicker metal shells and usually offer optional shell thickness for all roll diameters. Idler shell wear life is more of a factor for the return idlers since it normally contacts the “dirty” side of the belt resulting in abrasive wear of the shell. The exception to this would be a conveyor system with a belt turnover system. With normal conveyor systems, materials build up on the roll and increase its effective diameter. Because the buildup is never uniform and usually is less at the belt edges, the clean sections of the return roll travel at a slower surface speed than that of the belt. This results in relative slippage, thereby accelerating wear of both the belt cover and the surface of the roll. Thus the life of the roll shell is usually shorter on return belt idlers than on carrying idlers. The material buildup can also aggravate belt training.

Impact forces at conveyor loading points are not covered by this selection procedure. This is due to the many variables and different designs utilized at loading points. Your CEMA idler manufacturer should be consulted for impact idler recommendations.

Idler Selection Procedure

Preface to Selection Procedure Figures and Tables

Initial Selection, Steps 1 and 2:

Select idler class by comparing calculated idler load with idler load ratings (CIL and CIL_R) from Tables 5-7 through 5-10. CEMA idler manufacturers have standard designs meeting these load ratings and dimensional standards shown in tables listed in this publication.

Bearing L_{10} Life Correction, Steps 3, 4, and 5:

Factors K_2 (Fig. 5.17) and K_{3A} (Fig. 5.18) are multiplying factors used to adjust basic L_{10} life rating of idler class selected. Factor K_2 is based on percent of idler load and K_{3A} is factor for actual roll speed (rpm). Factor K_{3B} (Fig. 5.19), step 5 is an optional step showing the advantage of using larger diameter rolls. It can be used as a multiplier to save repeating step 4 if a larger diameter roll is used.

Determine Potential Idler Life, Step 6:

Factors K_{4A} (Fig. 5.20), K_{4B} (Fig. 5.21) and K_{4C} (Fig. 5.22) show conditions which will affect idler life and are independent of bearing L_{10} life, idler load, and idler class. Use these figures to evaluate the potential expected idler life. Contact your CEMA idler manufacturer for recommendations.

Step No. 1 Troughing Idler Series Selection

Calculated Idler Load (lbs) = $CIL = ((WB + (WM \times KI)) \times SI) + IML$

Where:

WB = Belt weight (lbs/ft) use actual or estimate from Table 5-5

WM = Material weight (lbs/ft) = $(Q \times 2000) / (60 \times Vee)$

Q = Quantity of material conveyed (tons per hour)

Vee = Design belt speed (fpm)

SI = Spacing of idlers (ft)

KI = Lump adjustment factor (see Table 5-6)

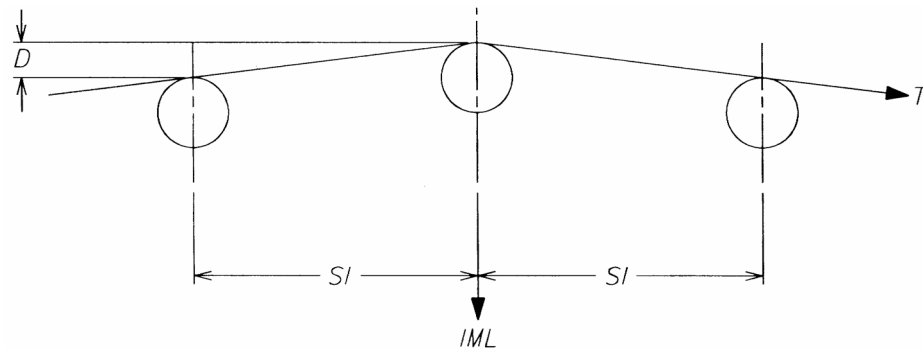
NOTE: The actual weight of the lump should be compared with the WM value. In certain situations it may be necessary to use actual lump weight as WM . Contact your CEMA idler manufacturer if you have doubts as to which value to use.

IML = Idler misalignment load (lbs) due to idler height deviation and belt tension = $(D \times T) / (6 \times SI)$ where:

D = Misalignment (in.)

T = Belt tension (lbs)

SI = Idler spacing (ft)



When an idler is higher than the adjacent idler, a component of belt tension will add load to that idler. The amount of height deviation can vary with the installation and type of idler. CEMA publication on conveyor installation standards (Appendix D, Belt Conveyors for Bulk Material, fifth edition or later) list recommendations on structure misalignment).

Use CIL and select proper series of idler from Tables 5-7 through 5-10.

CIL value should be equal to or less than idler rating.

This troughing idler selection procedure for calculated idler load does not include the following:

1. Impact force on idler at conveyor loading points.
2. Effect of belt transitions (head and tail pulley) on idler load.

Idler series selection for these load conditions should be verified by your CEMA idler manufacturer.

Table 5-5. WB-Estimated average belt weight, multiple- and reduced-ply belts, lbs/ft.

Belt Width (inches (b))	Material Carried, lbs/cu ft		
	30-74	75-129	130-200
18	3.5	4.0	4.5
24	4.5	5.5	6.0
30	6.0	7.0	8.0
36	9.0	10.0	12.0
42	11.0	12.0	14.0
48	14.0	15.0	17.0
54	16.0	17.0	19.0
60	18.0	20.0	22.0
72	21.0	24.0	26.0
84	25.0	30.0	33.0
96	30.0	35.0	38.0

1. Steel cable belts increase the above value by 50%.
2. Actual belt weights vary with different constructions, manufacturers, cover gauges, etc. Use the above values for estimating. Obtain actual values from the belt manufacturer whenever possible.

Table 5-6. K1-Lump adjustment factor.

Maximum Lump Size (inches)	Material Weight, lbs/cu ft						
	50	75	100	125	150	175	200
4	1.0	1.0	1.0	1.0	1.1	1.1	1.1
6	1.0	1.0	1.0	1.1	1.1	1.1	1.1
8	1.0	1.0	1.1	1.1	1.2	1.2	1.2
10	1.0	1.1	1.1	1.1	1.2	1.2	1.2
12	1.0	1.1	1.1	1.2	1.2	1.2	1.3
14	1.1	1.1	1.1	1.2	1.2	1.3	1.3
16	1.1	1.1	1.2	1.2	1.3	1.3	1.4
18	1.1	1.1	1.2	1.2	1.3	1.3	1.4

Step No. 2 Return idler series selection

$$\text{Calculated Idler Load (lbs)} = CIL_R = (WB \times SI) + IML$$

Use CIL_R and select proper series of idler from Tables 5-7 through 5-10. CIL_R should be equal to or less than return idler rating.

Step No. 3 K_2 = Effect of load on predicted bearing L_{10} life

When Calculated Idler Load (CIL) is less than the CEMA load rating of a series idler selected, the bearing L_{10} life will increase.

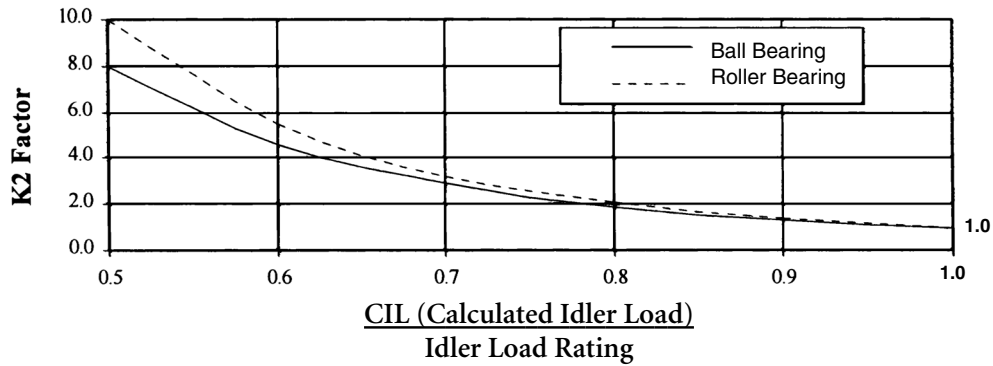


Figure 5.17 K_2 = Effect of load on predicted bearing L_{10} life.

Step No. 4 K_{3A} = Effect of belt speed on predicted bearing L_{10} life

CEMA L_{10} life ratings are based on 500 rpm. Slower speeds increase life and faster speeds decrease life. Figure 5.18 shows this relationship.

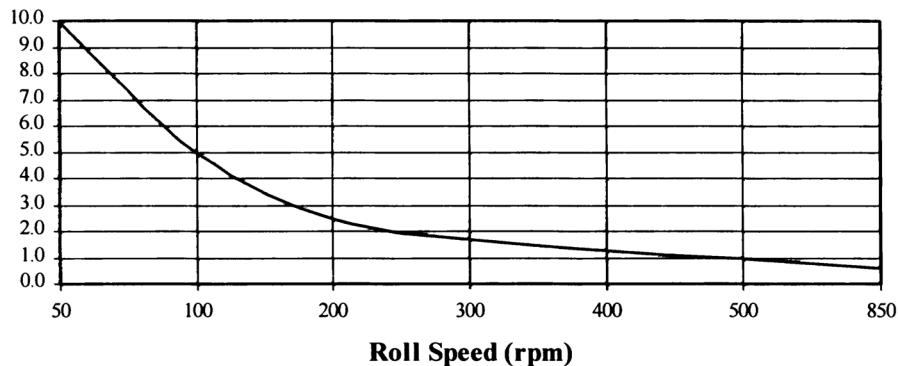


Figure 5.18 K_{3A} = Effect of belt speed on predicted bearing L_{10} life.

$$rpm = \frac{\text{Belt Speed (fpm)} \times 12}{\text{Roll Diameter (in)} \times \pi}$$

Step No. 5 $K3B$ = Effect of roll diameter on predicted bearing L_{10} life

For a given belt speed, using larger diameter rolls will increase idler L_{10} life. Figure 5.19 depicts L_{10} life adjustments for various roll diameters using 4" diameter as a value of 1.0. Percent life increase can be calculated for each roll diameter increase.

Example: $\frac{1.5 \text{ for 6 in. diam.}}{1.25 \text{ for 5 in. diam.}} = 1.20$, or 20% increase in L_{10} life

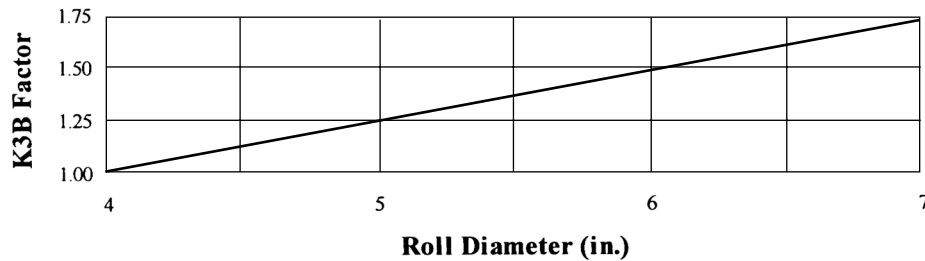


Figure 5.19 $K3B$ = Effect of roll diameter on predicted bearing L_{10} life (based on same belt speed).

NOTE: In addition to increased predicted bearing L_{10} life, larger diameter rolls can increase idler wear life.

Step No. 6 $K4$ = Environmental, maintenance, and other special conditions

Figure 5.20. $K4A$ = Effect of maintenance on potential idler life

Figure 5.21. $K4B$ = Effect of environment on potential idler life

Figure 5.22. $K4C$ = Effect of operating temperature on potential idler life

Based on collective application experience by CEMA idler manufacturers, these conditions are very important in determining potential idler life. However, the exact mathematical basis is very subjective, so contact your CEMA idler manufacturer for assistance or for any unusual conditions not listed.

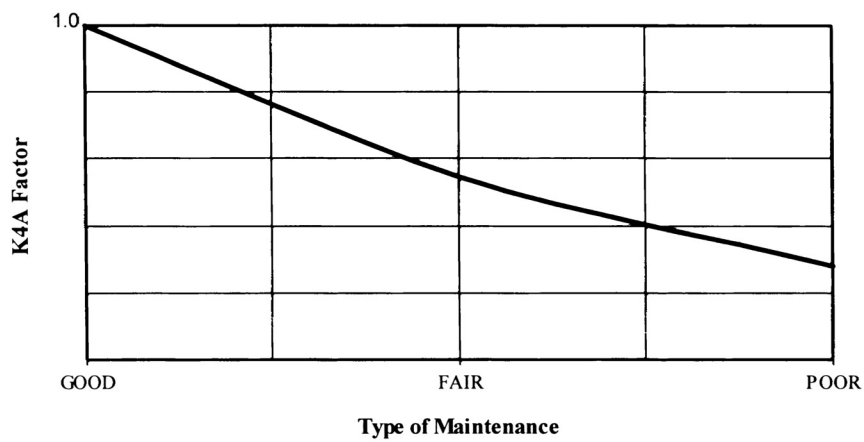


Figure 5.20 K4A = Effect of maintenance on potential idler life.

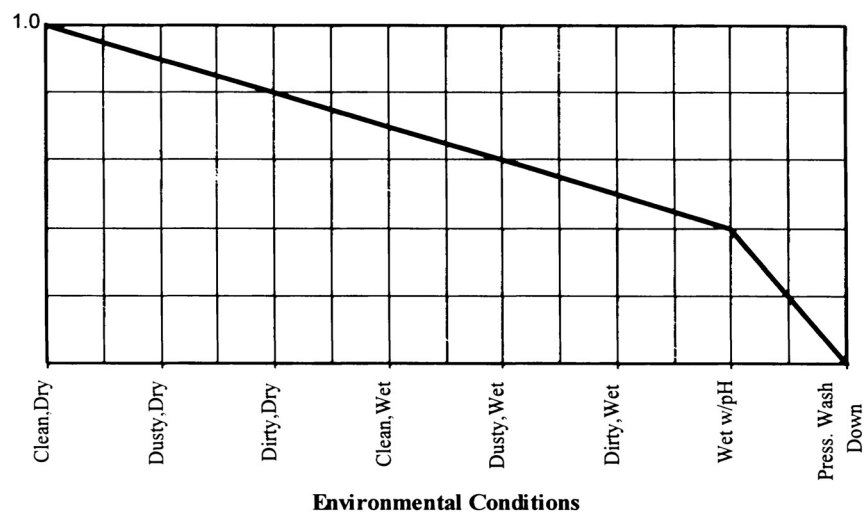


Figure 5.21 K4B = Effect of environmental conditions on potential idler life.

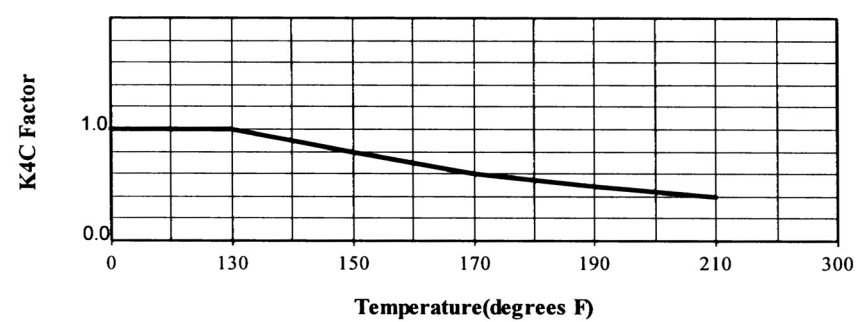


Figure 5.22 K4C = Effect of operating temperature on potential idler life.

Table 5-7. Load ratings for CEMA B idlers, lbs (rigid frame).

Belt Width (inches)	Trough Angle			Single Roll Return
	20°	35°	45°	
18	410	410	410	220
24	410	410	410	190
30	410	410	410	165
36	410	410	396	155
42	390	363	351	140
48	380	353	342	130

Ratings based on minimum L_{10} of 30,000 hours at 500 rpm.

Table 5-8. Load ratings for CEMA C idlers, lbs (rigid frame).

Belt Width (inches)	Trough Angle			Single Roll Return	Two Roll “Vee” Return
	20°	35°	45°		
18	900	900	900	475	
24	900	900	900	325	500
30	900	900	900	250	500
36	900	837	810	200	500
42	850	791	765	150	500
48	800	744	720	125	500
54	750	698	675	*	500
60	700	650	630	*	500
66				*	500

Ratings based on minimum L_{10} of 30,000 hours at 500 rpm.

*Use CEMA D return idler

Table 5-9. Load ratings for CEMA D idlers, lbs (rigid frame).

Belt Width (inches)	Trough Angle			Single Roll Return	Two Roll “Vee” Return
	20°	35°	45°		
24	1200	1200	1200	600	
30	1200	1200	1200	600	
36	1200	1200	1200	600	850
42	1200	1200	1200	500	850
48	1200	1200	1200	425	850
54	1200	1116	1080	375	850
60	1150	1070	1035	280	850
66				215	850
72	1050	977	945	155	850
78				125	850

Ratings based on minimum L_{10} of 60,000 hours at 500 rpm.

Table 5-10. Load ratings for CEMA E idlers, lbs (rigid frame and catenary where applicable).

Belt Width (inches)	Trough Angle			Single Roll Return	Two Roll “Vee” Return
	20°	35°	45°		
36	1800	1800	1800	1000	1300
42	1800	1800	1800	1000	1300
48	1800	1800	1800	1000	1300
54	1800	1800	1800	925	1300
60	1800	1800	1800	850	1300
66				775	1300
72	1800	1800	1800	700	1300
78				625	1300
84	1800	1674	1620	550	1300
90				475	1300
96	1750	1628	1575	400	1300
102				250	1300

Ratings based on minimum L_{10} of 60,000 hours at 500 rpm.

Notes:

1. Troughing idler load ratings (Tables 5.7–5.10) are for three equal length rolls.
2. Load ratings also apply for impact rolls.
3. Troughing idler load ratings are based on a load distribution of 70% on center roll and 15% on each end roll for all trough angles.
4. Unequal length rolls or picking idlers are not covered by this standard.

Example Idler Selection

Customer Furnished Data

Peak Load: 3,000 tph coal at 55/60 pcf minus 8" size

60" BW @ 650 fpm

T1 (Belt tension carrying side) _____ 37,000 lbs

T2 (Belt tension return side) _____ 12,000 lbs

Belt weight _____ 19 lbs/ft

D (Misalignment due to installation tolerances) _____ 1/4"

Conveyor system component design life 50,000 hours

Requested Information:

Recommended Idler Series and Spacing:

$$WM = \frac{3000 \times 2000}{60 \times 650} = 154 \text{ lb/ft}$$

Optional verification of customer data

A. Page 53, Table 4-3 - 35° troughed belt cross section of load

60" BW @ 20° Surcharge = 2.876 ft²

Full belt load: 2.876 ft² x 55 pcf = 158 lbs/ft

$$\text{Percent full load: } (<100) = 100 \times \frac{154}{158} = 97.47\%$$

Since this has been identified as Peak Load, the belt width, belt speed, and trough angle shown are good selections.

B. Page 64, Table 5-2: Suggested normal spacing of belt conveyor idlers (SI).

60" BW @ 50 pcf = 4.0 ft

60" BW @ 75 pcf = 3.5 ft

NOTE: Factors to be considered when selecting idler spacing are belt weight, material weight, idler rating, idler life, belt rating, and belt tension. For general conveyor design and selection, limit belt sag to 2 percent of idler spacing at minimum tension conditions. Sag limits during conveyor starting and stopping should also be considered in overall selection.

Idler Selection

Step No. 1: Carrying/troughing idler series selection based on Item B above. Use preliminary selection of 4 ft.

$$CIL = ((19 + (154 \times 1.0)) \times 4) + \frac{37,000 \times 0.25}{6 \times 4} = 1,077 \text{ lb}$$

$K1 = 1.0$ for 8-in lump

Per Table 5-9, D x 35° = 1070 lb rating

Per Table 5-10, E x 35° = 1800 lb rating

NOTE: Although it is recommended that CIL be equal to or less than CEMA Idler Load Rating, there is a certain amount of judgment involved in final selection. In this example, an experienced belt designer would know that the maximum IML load based on belt tension occurs at the head or discharge for a level or incline conveyor. Since belt tension would be decreasing from this point towards the tail or loading end, the number of idlers that slightly exceeded the CEMA Idler Load Rating could be determined. D series x 35 degrees could be used and verification requested from CEMA idler manufacturer.

Other choices are:

- A. D series at less than 4 ft spacing
- B. E series at greater than 4 ft spacing
- C. Increase belt speed which will decrease WM. This option would also decrease $T1$ belt tension which would decrease IML.
- D. Customer to maintain less than ¼" height deviation due to installation tolerances.

Some of these choices would require recalculating belt tensions, etc., and then weighing the economics with the expected performance of each selection. For this example, we will select D series x 35 degree troughing idlers at 4 ft spacing, although optional choices C & D have great merit. The rated bearing L_{10} life is 60,000 hours.

Step No. 2: Return Idler Series Selection

Option: From Chapter 5, page 64, Table 5-2: suggested normal spacing of belt conveyor idler (SI)

Return idlers 60" $BW = 10$ ft

$$CILR = (19 \times 10) + \frac{12,000 \times 0.25}{6 \times 10} = 240 \text{ lb}$$

Based on the above option, use a preliminary selection of 10 ft spacing.

NOTE: Quite often it is desirable to have return the idler spacing at a multiple of the troughing idler spacing to simplify stringer or truss design. However, this should not be the control for selection.

Per Table 5-9: D series single roll return = 280 lb rating

Per Table 5-8: C series two roll "Vee" return = 500 lb rating

If this conveyor has long centers, consideration should be given to using two roll "Vee" returns and increasing spacing. With this choice it would not be necessary to use training idlers. For this example, select D series single roll return at 12'-0". The rated bearing L_{10} life is 60,000 hours.

$$CILR = (19 \times 12) + \frac{12,000 \times 0.25}{6 \times 12} = 270 \text{ lbs}$$

Steps 1 and 2 have selected idlers based on load. Steps 3, 4, and 5 deal with predicted bearing L_{10} life, and Step 6 covers conditions affecting potential idler life.

Step No. 3: K2 Effect of Load on Predicted Bearing L_{10} Life

$$\text{Troughing Idler} = \frac{CIL}{\text{Idler Load Rating}} = \frac{1,077}{1,070} = 1.007$$

K_2 (from Figure 5.17, Tapered Roller Bearing) = 1.0

Bearing $L_{10} = (60,000 \times 1.0) = 60,000$ hours

$$\text{Return Idler} = \frac{CIL}{\text{Idler Load Rating}} = \frac{270}{280} = 0.96$$

K_2 (from Figure 5.17, Tapered Roller Bearing) = 1.15

Bearing $L_{10} = (60,000 \times 1.15) = 69,000$ hours

Step No. 4: K3A Effect of Belt Speed on Predicted Bearing L_{10} Life

650 fpm belt speed specified.

Select minimum roll dia. For <500 rpm at 650 fpm

From Table 5.4, pg. 69, 5" dia. = 654 fpm

K_{3A} (from Figure 5.18) = 1.0

Bearing L_{10} life for D5 series idlers at 650 fpm

Troughing idler = $(60,000 \times 1.0) = 60,000$ hours

Return idler = $(69,000 \times 1.0) = 69,000$ hours

Step No. 5: K3B Effect of Roll Diameter on Predicted Bearing Life

Compare bearing L_{10} life increase for 6" diameter roll.

Example Idler Selection

$K3B$ (Figure 5.19) = $\frac{6 \text{ inch diameter roll} = 1.50}{5 \text{ inch diameter roll} = 1.25} = 1.20$ or 20% increase in bearing L_{10} life

Troughing idler = $(60,000 \times 1.2) = 72,000$ hours

Return idler = $(69,000 \times 1.2) = 82,800$ hours

NOTE: In addition, the 6" roll would have longer wear life, and roll resistance would be less, which would decrease belt tension and reduce IML.

D6 idlers are recommended. Idler selection is based on customer furnished data.

Troughing idlers D6 x 35° at 4 ft spacing with 72,000 hours predicted bearing L_{10} life.

Return idlers D6 at 12 ft spacing with 82,800 hours predicted bearing L_{10} life.

Step No. 6: $K4$ Effect of Environmental, Maintenance, and Temperature on Potential Idler Life

For the purpose of example we will assume the following conditions:

$K4A$ (Figure 5.20) Maintenance: _____ Good to Fair

$K4B$ (Figure 5.21) Environmental: _____ Dirty

$K4C$ (Figure 5.22) Temperature: _____ <120 F

Hostile environmental conditions and the level of commitment to the belt conveyor installation and maintenance will affect idler life. With the above assumed conditions it is apparent that potential idler life will be less than the predicted bearing L_{10} life. These conditions should be discussed with your CEMA idler manufacturer.

Expected or potential idler life may also be limited by shell wear. Shell wear can vary considerably with each installation. In addition to conveyed material characteristics, environmental, and maintenance factors, idler alignment and belt cleaning can have a significant effect on shell wear and idler life.

NOTE: Calculated idler loads should be repeated for training idlers (if used). Height deviation of training idlers must be included for IML calculation or controlled by shimming and maintaining closer installation tolerances at these areas of conveyor.

Conclusion: There are numerous options available to the belt conveyor designer in regard to idler selection. Through involving your CEMA idler manufacturer in this selection process these options can be explored, resulting in a reliable, cost effective installation.

Belt Alignment

A belt conveyor must be designed, constructed, and maintained so that the belt consistently runs centrally on its mechanical system of idlers and pulleys. To accomplish this, the following conditions must prevail:

1. Square the tail and head pulleys with the conveyor frame.
2. Square all idlers and returns with the conveyor frame; be sure they are in line and lie in the same horizontal plane; and tighten the attachment belts.
3. Level all frames to ensure a cross-section parallel to the ground plane. If one side of the conveyor frame is lower than the other, gravity will force the belt off-center.
4. The belt must be straight and the belt splice square. If side creep occurs only in the vicinity of the belt splice, the splice may not be square with the belt. In general, if the creep follows the belt, there is a problem with the belt. If it remains in one general vicinity, there is a problem with the system.
5. The belt should have good contact with all troughing rolls.
6. Load material centrally on the belt.

There may be times when the above procedure is not sufficient and the belt persistently runs to one side. The following corrective measures may be initiated to prevent side creep:

1. While running the belt at the lowest speed possible, find the point of maximum side creep. The idler preceding this point along the direction of belt travel can be adjusted to minimize side creep. Facing the conveyor from the tail end, the idler must be pivoted clockwise to correct side creep on the left and counterclockwise to correct side creep on the right. Once the belt is centered, change to a higher speed (if possible) and load the belt with material. Continue adjusting until normal operating conditions do not cause the belt to misalign.
2. If creep persists, insure that the head and tail pulleys are perfectly aligned. Steer the belt with the carrying or return idlers.
3. Training idlers can be installed to replace troughing or return idlers. They should only be used in problem systems and should be at least 50 feet from any terminal or bend pulleys. Do not use a training idler in a vertical curve. Reversible belt training idlers are available for reversible belt conveyors. Free rotation of the training idler's vertical bearing is essential for satisfactory tracking results.
4. If creep still persists, some or all of the troughing idlers may be tilted not more than 2 degrees from the vertical, in the direction of belt travel. This can be accomplished by using a steel flat washer between the conveyor frame and the troughing idler foot plate. **Caution:** Troughing idlers which have this tilt built in should not be additionally tilted. Reversible belts should not use tilted idlers, as the misalignment of the belt would be accentuated when it runs in the reverse direction. The effect on belt tension and idler roll shell wear should be considered when tilting idlers.

5. If none of the above steps solve a belt misalignment condition, the conveyor should be laser aligned and corrective action taken based upon the survey data.

NOTE: Maximum tracking effect occurs when the distance between the corrective component and the following component is maximized.

Table 5-11. Average weight (lbs) of troughing idler rotating parts—steel rolls.

Belt Width (inches)	CEMA Idler Class								
	B4	B5	C4	C5	C6	D5	D6	E6	E7
18	15.0	19.2	14.5	19.1	26.7				
24	18.3	24.2	17.5	23.2	32.6	23.2	32.6		
30	21.8	28.3	20.5	26.8	38.0	26.8	38.0		
36	25.3	33.0	23.5	31.3	43.6	31.3	43.6	64.8	81.8
42	30.8	38.1	26.5	35.2	49.2	35.2	49.2	73.3	91.7
48	32.9	41.6	29.5	39.3	54.8	39.3	54.8	81.9	101.3
54				45.9	62.3	45.9	62.3	93.6	121.8
60				50.1	68.3	50.1	68.3	102.2	132.7
72						57.9	77.8	119.4	154.5
84								132.0	164.0
96								145.3	173.0

Table 5-12. Average weight (lbs) of return idler rotating parts—steel rolls.

Belt Width (inches)	CEMA Idler Class								
	B4	B5	C4	C5	C6	D5	D6	E6	E7
18	13.1	16.3	12.2	16.6	21.6				
24	16.3	20.9	15.2	20.1	27.1	20.9	30.1		
30	19.5	24.5	18.2	24.0	32.3	25.8	35.4		
36	22.7	28.5	21.2	28.0	37.6	30.1	40.5	59.0	70.0
42	26.0	33.0	24.6	32.1	43.3	34.3	47.2	67.4	80.1
48	27.4	36.1	27.6	36.1	48.4	38.7	54.4	75.6	89.9
54						43.4	60.8	83.2	99.9
60						49.2	68.1	92.2	109.4
72						55.1	74.9	109.4	129.0
84								114.0	136.2
96								122.0	149.8

Table 5-13. WK^2 (lb-in²) average for three equal-roll troughing idlers.

Belt Width (inches)	CEMA Idler Class								
	B4	B5	C4	C5	C6	D5	D6	E6	E7
18	43	90	45	90	179				
24	53	116	56	116	224	116	224		
30	65	140	67	140	269	140	269		
36	78	161	79	161	313	161	313	446	801
42	90	185	91	185	358	185	358	502	876
48	104	207	105	207	400	207	400	564	1017
54				223	447	223	447	628	1127
60				245	493	245	493	689	1234
72						266	538	811	1451
84								985	1598
96								1114	1804

Table 5-14. WK^2 (lb-in²) average for single steel return idlers.

Belt Width (inches)	CEMA Idler Class								
	B4	B5	C4	C5	C6	D5	D6	E6	E7
18	41	85	42	85	165				
24	50	106	51	106	210	106	210		
30	61	128	61	128	254	128	254		
36	72	152	72	152	312	152	312	419	750
42	83	174	83	174	348	174	348	479	857
48	94	196	94	196	385	196	385	539	963
54						218	429	597	1070
60						234	473	659	1177
72						256	513	779	1391
84								933	1581
96								1074	1737

Belt Tension, Power, and Drive Engineering

Basic power requirements
Belt tension calculations
CEMA horsepower formula
Drive pulley relationships
Drive arrangements
Maximum and minimum belt tensions
Tension relationships and belt sag between idlers
Acceleration and deceleration forces
Analysis of acceleration and deceleration forces
Design considerations
Conveyor horsepower determination — graphical method
Examples of belt tension and horsepower calculations — six problems
Belt conveyor drive equipment
Backstops
Brakes
Brakes and backstops in combination
Devices for acceleration, deceleration, and torque control
Brake requirement determination (deceleration calculations)

The earliest application engineering of belt conveyors was, to a considerable extent, dependent upon empirical solutions that had been developed by various manufacturers and consultants in this field. The belt conveyor engineering analysis, information, and formulas presented in this manual represent recent improvements in the concepts and data which have been developed over the years, using the observations of actual belt conveyor operation and the best mathematical theory.

Horsepower (*hp*) and tension formulas, incorporating successively all the factors affecting the total force needed to move the belt and its load, are presented here in a manner that permits the separate evaluation of the effect of each factor. These formulas represent the consensus of all CEMA member companies.

In recent years, CEMA member companies have developed computer programs capable of complete engineering analysis of the most complex and extensive belt conveyor systems. These programs are more comprehensive and include more extensive analysis and calculations than can be included in this manual. Although the programs are treated as proprietary information, each CEMA member company welcomes an opportunity to assist in the proper application of belt conveyor equipment. One advantage of using computer programs is the speed and accuracy with which they provide information for alternate conveyor designs.

Basic Power Requirements

The horsepower, *hp*, required at the drive of a belt conveyor, is derived from the pounds of the effective tension, T_e , required at the drive pulley to propel or restrain the loaded conveyor at the design velocity of the belt V , in fpm:

(1)

$$hp = \frac{T_e \times V}{33,000}$$

To determine the effective tension, T_e , it is necessary to identify and evaluate each of the individual forces acting on the conveyor belt and contributing to the tension required to drive the belt at the driving pulley. T_e is the final summarization of the belt tensions produced by forces such as:

1. The gravitational load to lift or lower the material being transported.
2. The frictional resistance of the conveyor components, drive, and all accessories while operating at design capacity.
3. The frictional resistance of the material as it is being conveyed.
4. The force required to accelerate the material continuously as it is fed onto the conveyor by a chute or a feeder.

The basic formula for calculating the effective tension, T_e , is:

$$T_e = LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) + T_p + T_{am} + T_{ac} \quad (2)$$

Belt Tension Calculations

The following symbols will be used to assist in the identification and evaluation of the individual forces that cumulatively contribute to T_e and that are therefore components of the total propelling belt tension required at the drive pulley:

A_i = belt tension, or force, required to overcome frictional resistance and rotate idlers, lbs (see page 91)

C_l = friction modification factor for regenerative conveyor

H = vertical distance that material is lifted or lowered, ft

K_t = ambient temperature correction factor (see Figure 6.1)

K_x = factor used to calculate the frictional resistance of the idlers and the sliding resistance between the belt and idler rolls, lbs per ft (see equation 3, page 91)

K_y = carrying run factor used to calculate the combination of the resistance of the belt and the resistance of the load to flexure as the belt and load move over the idlers (see equation 4, page 94, and Table 6-2). For return run use constant 0.015 in place of K_y . See T_{yr} .

L = length of conveyor, ft

Q = tons per hour conveyed, tph, short tons of 2,000 lbs

S_i = troughing idler spacing, ft

T_{ac} = total of the tensions from conveyor accessories, lbs:

$$T_{ac} = T_{sb} + T_{pl} + T_{tr} + T_{bc}$$

T_{am} = tension resulting from the force to accelerate the material continuously as it is fed onto the belts, lbs

T_b = tension resulting from the force needed to lift or lower the belt, lbs (see page 116):

$$T_b = \pm H \times W_b$$

T_{bc} = tension resulting from belt pull required for belt-cleaning devices such as belt scrapers, lbs

T_e = effective belt tension at drive, lbs

T_m = tension resulting from the force needed to lift or lower the conveyed material, lbs:

$$T_m = \pm H \times W_m$$

T_p = tension resulting from resistance of belt to flexure around pulleys and the resistance of pulleys to rotation on their bearings, total for all pulleys, lbs

T_{pl} = tension resulting from the frictional resistance of plows, lbs

T_{sb} = tension resulting from the force to overcome skirtboard friction, lbs

T_{tr} = tension resulting from the additional frictional resistance of the pulleys and the flexure of the belt over units such as trippers, lbs

T_x = tension resulting from the frictional resistance of the carrying and return idlers, lbs:

$$T_x = L \times K_x \times K_t$$

T_{yb} = total of the tensions resulting from the resistance of the belt to flexure as it rides over both the carrying and return idlers, lbs:

$$T_{yb} = T_{yc} + T_{yr}$$

T_{yc} = tension resulting from the resistance of the belt to flexure as it rides over the carrying idlers, lbs:

$$T_{yc} = L \times K_y \times W_b \times K_t$$

T_{ym} = tension resulting from the resistance of the material to flexure as it rides with the belt over the carrying idlers, lbs:

$$T_{ym} = L \times K_y \times W_m$$

T_{yr} = tension resulting from the resistance of the belt to flexure as it rides over the return idlers, lbs:

$$T_{yr} = L \times 0.015 \times W_b \times K_t$$

V = design belt speed, fpm

W_b = weight of belt in pounds per foot of belt length. When the exact weight of the belt is not known, use average estimated belt weight (see Table 6-1)

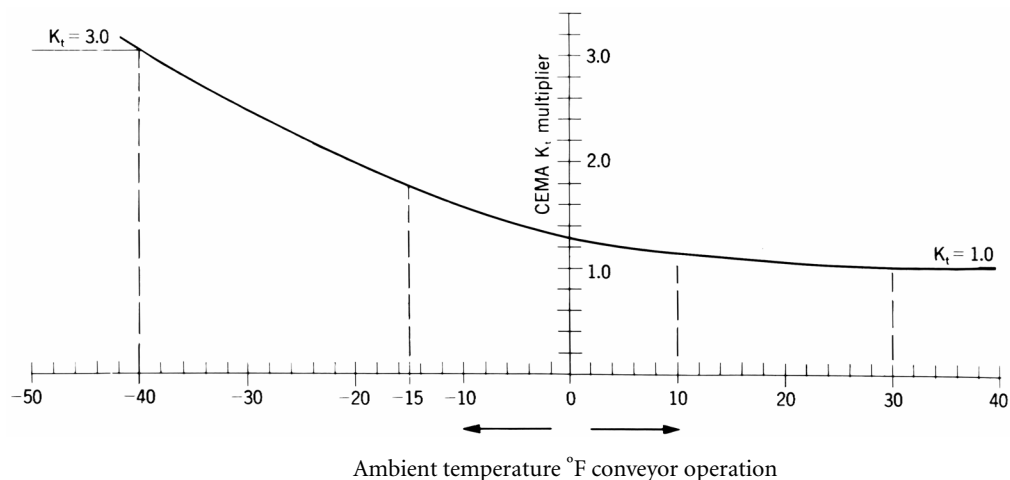
W_m = weight of material, lbs per foot of belt length:

$$W_m = \frac{Q \times 2,000}{60 \times V} = \frac{33.33 \times Q}{V}$$

Three multiplying factors, K_t , K_x , and K_y , are used in calculations of three of the components of the effective belt tension, T_e .

K_t — Ambient Temperature Correction Factor

Idler rotational resistance and the flexing resistance of the belt increase in cold weather operation. In extremely cold weather the proper lubricant for idlers must be used to prevent excessive resistance to idler rotation.



Operation at temperatures below -15°F involves problems in addition to horsepower considerations. Consult conveyor manufacturer for advice on special belting, greasing, and cleaning specifications and necessary design modification.

Figure 6.1 Variation of temperature correction factor, K_t , with temperature.

K_t is a multiplying factor that will increase the calculated value of belt tensions to allow for the increased resistances that can be expected due to low temperatures. Figure 6.1 provides values for factor K_t .

K_x — Idler Friction Factor

The frictional resistance of idler rolls to rotation and sliding resistance between the belt and the idler rolls can be calculated by using the multiplying factor K_x . K_x is a force in lbs/ft of conveyor length to rotate the idler rolls, carrying and return, and to cover the sliding resistance of the belt on the idler rolls. The K_x value required to rotate the idlers is calculated using equation (3).

The resistance of the idlers to rotation is primarily a function of bearing, grease, and seal resistance. A typical idler roll equipped with antifriction bearings and supporting a load of 1,000 lbs will require a turning force at the idler roll periphery of from 0.5 to 0.7 lbs to overcome the bearing friction. The milling or churning of the grease in the bearings and the bearing seals will require additional force. This force, however, is generally independent of the load on the idler roll.

Under normal conditions, the grease and seal friction in a well-lubricated idler will vary from 0.1 to 2.3 lbs/idler, depending upon the type of idler, the seals, and the condition of the grease.

Sliding resistance between the belt and idler rolls is generated when the idler rolls are not exactly at 90 degrees to the belt movement. After initial installation, deliberate idler misalignment is often an aid in training the belt. Even the best installations have a small requirement of this type. However, excessive idler misalignment results in an extreme increase in frictional resistance and should be avoided.

Table 6-1. Estimated average belt weight, multiple- and reduced-ply belts, lbs/ft.

Belt Width inches (<i>b</i>)	Material Carried, lbs/ft ³		
	30-74	75-129	130-200
18	3.5	4.0	4.5
24	4.5	5.5	6.0
30	6.0	7.0	8.0
36	9.0	10.0	12.0
42	11.0	12.0	14.0
48	14.0	15.0	17.0
54	16.0	17.0	19.0
60	18.0	20.0	22.0
72	21.0	24.0	26.0
84	25.0	30.0	33.0
96	30.0	35.0	38.0

1. Steel-cable belts — increase above value by 50 percent.

2. Actual belt weights vary with different constructions, manufacturers, cover gauges, etc. Use the above values for estimating. Obtain actual values from the belt manufacturer whenever possible.

Some troughing idlers are designed to operate with a small degree of tilt in the direction of belt travel, to aid in belt training. This tilt results in a slight increase in sliding friction that must be considered in the horsepower formula.

Values of K_x can be calculated from the equation:

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}, \text{ lbs tension per foot of belt length} \quad (3)$$

$A_i = 1.5$ for 6" diameter idler rolls, CEMA C6, D6

$A_i = 1.8$ for 5" diameter idler rolls, CEMA B5, C5, D5

$A_i = 2.3$ for 4" diameter idler rolls, CEMA B4, C4

$A_i = 2.4$ for 7" diameter idler rolls, CEMA E7

$A_i = 2.8$ for 6" diameter idler rolls, CEMA E6

For regenerative declined conveyors, $A_i = 0$.

The A_i values tabulated above are averages and include frictional resistance to rotation for both the carrying and return idlers. Return idlers are based on single roll type. *If two roll V return idlers are used, increase A_i value by 5%.* In the case of long conveyors or very high belt speed (over 1,000 fpm) refer to CEMA member companies for more specific values of A_i .

K_y — Factor for Calculating the Force of Belt and Load Flexure over the Idlers

Both the resistance of the belt to flexure as it moves over idlers and the resistance of the load to flexure as it rides the belt over the idlers develop belt-tension forces. K_y is a multiplying factor used in calculating these belt tensioning forces.

Table 6-2 gives values of K_y for carrying idlers as they vary with differences in the weight/ft of the conveyor belt, W_b ; load, W_m ; idler spacing, S_i ; and the percent of slope or angle that the conveyor makes with the horizontal. When applying idler spacing, S_i , other than specified in Table 6-2, use Table 6-3 to determine a corrected K_y value.

Example 1. For a conveyor whose length is 800 ft and $(W_b + W_m) = 150$ lbs/ft having a slope of 12%, the K_y value (Table 6-2) is .017. This K_y value is correct only for the idler spacing of 3.0 ft. If a 4.0-foot idler spacing is to be used, using Table 6-3 and the K_y reference values at the top of the table, the K_y of .017 lies between .016 and .018. Through interpolation and using the corresponding K_y values for 4.0-foot spacing, the corrected K_y value is .020.

Example 2. For a conveyor whose length is 1,000 ft and $(W_b + W_m) = 125$ lbs/ft with a slope of 12%, the K_y value (Table 6-2) is .0165. This value is correct only for 3.5-foot spacing. If 4.5-foot spacing is needed, Table 6-3 shows that .0165 lies between .016 and .018 (reference K_y). Through interpolation and using the corresponding K_y values for 4.5-foot spacing, the corrected K_y value is .0194.

Table 6-2. Factor K_y values.

Conveyor Length (ft)	$W_b + W_m$ (lbs/ft)	Percent Slope						
		0	3	6	9	12	24	33
		Approximate Degrees						
		0	2	3.5	5	7	14	18
250	20	0.035	0.035	0.034	0.031	0.031	0.031	0.031
	50	0.035	0.034	0.033	0.032	0.031	0.028	0.027
	75	0.035	0.034	0.032	0.032	0.030	0.027	0.025
	100	0.035	0.033	0.032	0.031	0.030	0.026	0.023
	150	0.035	0.035	0.034	0.033	0.031	0.025	0.021
	200	0.035	0.035	0.035	0.035	0.032	0.024	0.018
	300	0.035	0.035	0.035	0.035	0.032	0.019	0.018
400	20	0.035	0.034	0.032	0.030	0.030	0.030	0.030
	50	0.035	0.033	0.031	0.029	0.029	0.026	0.025
	75	0.034	0.033	0.030	0.029	0.028	0.024	0.021
	100	0.034	0.032	0.030	0.028	0.028	0.022	0.019
	150	0.035	0.034	0.031	0.028	0.027	0.019	0.016
	200	0.035	0.035	0.033	0.030	0.027	0.016	0.014
	300	0.035	0.035	0.034	0.029	0.024	0.018	0.018
500	20	0.035	0.033	0.031	0.030	0.030	0.030	0.030
	50	0.034	0.032	0.030	0.028	0.028	0.024	0.023
	75	0.033	0.032	0.029	0.027	0.027	0.021	0.019
	100	0.033	0.031	0.029	0.028	0.026	0.019	0.016
	150	0.035	0.033	0.030	0.027	0.024	0.016	0.016
	200	0.035	0.035	0.030	0.027	0.023	0.016	0.016
	300	0.035	0.035	0.029	0.024	0.019	0.018	0.018
600	20	0.035	0.032	0.030	0.029	0.029	0.029	0.029
	50	0.033	0.030	0.029	0.027	0.026	0.023	0.021
	75	0.032	0.030	0.028	0.026	0.024	0.020	0.016
	100	0.032	0.030	0.027	0.025	0.022	0.016	0.016
	150	0.035	0.031	0.026	0.024	0.019	0.016	0.016
	200	0.035	0.031	0.026	0.021	0.017	0.016	0.016
	300	0.035	0.031	0.023	0.018	0.018	0.018	0.018
800	20	0.035	0.031	0.030	0.029	0.029	0.029	0.029
	50	0.032	0.029	0.028	0.026	0.025	0.021	0.018
	75	0.031	0.029	0.026	0.024	0.022	0.016	0.016
	100	0.031	0.028	0.025	0.022	0.020	0.016	0.016
	150	0.034	0.028	0.023	0.019	0.017	0.016	0.016
	200	0.035	0.027	0.021	0.016	0.016	0.016	0.016
	300	0.035	0.026	0.020	0.017	0.016	0.016	0.016

Idle spacing: The above values of K_y are based on the following idler spacing (for other spacing, see Table 6-3).

$(W_b + W_m)$, lbs per ft	S_i , ft	$(W_b + W_m)$, lbs per ft	S_i , ft
Less than 50	4.5	100 to 149	3.5
50 to 99	4.0	150 and above	3.0

Table 6-2. Factor K_y values.

Conveyor Length (ft)	$W_b + W_m$ (lbs/ft)	Percent Slope						
		0	3	6	9	12	24	33
		Approximate Degrees						
		0	2	3.5	5	7	14	18
1000	50	0.031	0.028	0.026	0.024	0.023	0.019	0.016
	75	0.030	0.027	0.024	0.022	0.019	0.016	0.016
	100	0.030	0.026	0.022	0.019	0.017	0.016	0.016
	150	0.033	0.024	0.019	0.016	0.016	0.016	0.016
	200	0.032	0.023	0.017	0.016	0.016	0.016	0.016
	250	0.033	0.022	0.017	0.016	0.016	0.016	0.016
1400	300	0.033	0.021	0.018	0.018	0.018	0.018	0.018
	50	0.029	0.026	0.024	0.022	0.021	0.016	0.016
	75	0.028	0.024	0.021	0.019	0.016	0.016	0.016
	100	0.028	0.023	0.019	0.016	0.016	0.016	0.016
	150	0.029	0.020	0.016	0.016	0.016	0.016	0.016
	200	0.030	0.021	0.016	0.016	0.016	0.016	0.016
2000	250	0.030	0.020	0.017	0.016	0.016	0.016	0.016
	300	0.030	0.019	0.018	0.018	0.018	0.018	0.018
	50	0.027	0.024	0.022	0.020	0.018	0.016	0.016
	75	0.026	0.021	0.019	0.016	0.016	0.016	0.016
	100	0.025	0.020	0.016	0.016	0.016	0.016	0.016
	150	0.026	0.017	0.016	0.016	0.016	0.016	0.016
2400	200	0.024	0.016	0.016	0.016	0.016	0.016	0.016
	250	0.023	0.016	0.016	0.016	0.016	0.016	0.016
	300	0.022	0.018	0.018	0.018	0.018	0.018	0.018
	50	0.026	0.023	0.021	0.018	0.017	0.016	0.016
	75	0.025	0.021	0.017	0.016	0.016	0.016	0.016
	100	0.024	0.019	0.016	0.016	0.016	0.016	0.016
3000	150	0.024	0.016	0.016	0.016	0.016	0.016	0.016
	200	0.021	0.016	0.016	0.016	0.016	0.016	0.016
	250	0.021	0.016	0.016	0.016	0.016	0.016	0.016
	300	0.020	0.018	0.018	0.018	0.018	0.018	0.018
	50	0.024	0.022	0.019	0.017	0.016	0.016	0.016
	75	0.023	0.019	0.016	0.016	0.016	0.016	0.016

Idler spacing: The above values of K_y are based on the following idler spacing (for other spacing, see Table 6-3).

$(W_b + W_m)$, lbs per ft	S_i , ft	$(W_b + W_m)$, lbs per ft	S_i , ft
Less than 50	4.5	100 to 149	3.5
50 to 99	4.0	150 and above	3.0

K_y values in Tables 6-2 and 6-3 are applicable for conveyors up to 3,000 ft long with a single slope and a 3% maximum sag of the belt between the troughing and between the return idlers. The return idler spacing is 10 ft nominal and loading of the belt is uniform and continuous.

Equation (4) provides K_y values for the carrying idlers of belt conveyors whose length, number of slopes, and/or average belt tensions exceed the limitations specified above for the conveyors covered by Tables 6-2 and 6-3. This equation is applicable for conveyors in which the average belt tension is 16,000 lbs or less. To determine the K_y factor for use in calculating conveyors of this class, it is necessary, first, to assume a tentative value for the average belt tension. The graphical method for determining conveyor horsepower (pages 141 through 145) may be of assistance in estimating this initial tentative value of average belt tension.

After estimating the average belt tension and selecting an idler spacing, refer to Table 6-4 to obtain values for A and B for use in the following equation:

$$K_y = (W_m + W_b) \times A \times 10^{-4} + B \times 10^{-2} \quad (4)$$

By using equation (4), an initial value for K_y can be determined and an initial average belt tension can be subsequently calculated. The comparison of this calculated average belt tension with the original tentative value will determine the need to select another assumed belt tension. Recalculate K_y and calculate a second value for the average belt tension. The process should be repeated until there is reasonable agreement between the estimated and final calculated average belt tensions.

There are no tabulated K_y values or mathematical equations to determine a K_y for conveyors having an average belt tension exceeding 16,000 lbs. A reasonably accurate value that can be used for calculations is K_y equals 0.016. It is suggested that this value for K_y be considered a minimum, subject to consultation with a CEMA member company on any specific applications.

The force that results from the resistance of the belt to flexure as it moves over the idlers for the return run is calculated in the same manner as the resistance to flexure for the carrying run, except a constant value of 0.015 is used in place of K_y . The resistance of the belt flexure over idler rolls is a function of the belt construction, cover thickness and indentation by the idler rolls, type of rubber compound, idler roll diameter, temperature, and other factors. The belt flexing resistance increases at lower temperatures.

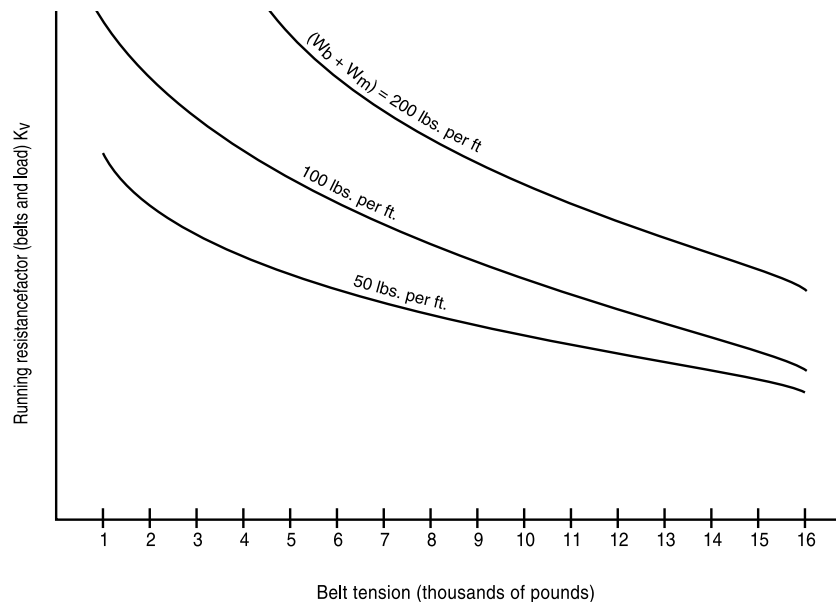


Figure 6.2 Effect of belt tension on resistance of material to flexure over idler rolls.

Table 6-3. Corrected factor K_y values when other than tabular carrying idler spacings are used.

$W_b + W_m$ (lbs/ft)	S_p (ft)	Reference Values of K_y for Interpolation									
		0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034
Less than 50	3.0	0.0160	0.0160	0.0160	0.0168	0.0183	0.0197	0.0212	0.0227	0.0242	0.0257
	3.5	0.0160	0.0160	0.0169	0.0189	0.0207	0.0224	0.0241	0.0257	0.0274	0.0291
	4.0	0.0160	0.0165	0.0182	0.0204	0.0223	0.0241	0.0259	0.0278	0.0297	0.0316
	4.5	0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034
	5.0	0.0174	0.0195	0.0213	0.0236	0.0254	0.0273	0.0291	0.0311	0.0329	0.0348
50 to 99	3.0	0.0160	0.0162	0.0173	0.0186	0.0205	0.0221	0.0239	0.026	0.0274	0.029
	3.5	0.0160	0.0165	0.0185	0.0205	0.0222	0.024	0.0262	0.0281	0.030	0.0321
	4.0	0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034
	4.5	0.0175	0.0193	0.0214	0.0235	0.0253	0.0272	0.0297	0.0316	0.0335	0.035
	5.0	0.0184	0.021	0.023	0.0253	0.027	0.029	0.0315	0.0335	0.035	0.035
100 to 149	3.0	0.0160	0.0164	0.0186	0.0205	0.0228	0.0246	0.0267	0.0285	0.0307	0.0329
	3.5	0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034
	4.0	0.0175	0.0197	0.0213	0.0234	0.0253	0.0277	0.0295	0.0312	0.033	0.035
	4.5	0.0188	0.0213	0.0232	0.0253	0.0273	0.0295	0.0314	0.033	0.0346	0.035
	5.0	0.0201	0.0228	0.0250	0.0271	0.0296	0.0316	0.0334	0.035	0.035	0.035
150 to 199	3.0	0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034
	3.5	0.0172	0.0195	0.0215	0.0235	0.0255	0.0271	0.0289	0.031	0.0333	0.0345
	4.0	0.0187	0.0213	0.0235	0.0252	0.0267	0.0283	0.0303	0.0325	0.0347	0.035
	4.5	0.0209	0.023	0.0253	0.0274	0.0289	0.0305	0.0323	0.0345	0.035	0.035
	5.0	0.0225	0.0248	0.0272	0.0293	0.0311	0.0328	0.0348	0.035	0.035	0.035
200 to 249	3.0	0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034
	3.5	0.0177	0.0199	0.0216	0.0235	0.0256	0.0278	0.0295	0.031	0.0327	0.0349
	4.0	0.0192	0.0216	0.0236	0.0256	0.0274	0.0291	0.0305	0.0322	0.0339	0.035
	4.5	0.021	0.0234	0.0253	0.0276	0.0298	0.0317	0.0331	0.0347	0.035	0.035
	5.0	0.0227	0.0252	0.0274	0.0298	0.0319	0.0338	0.035	0.035	0.035	0.035

To use this table to correct the value of K_y for idler spacing other than shown in bold type, apply the procedure shown in the two examples on page 91.

The resistance of the material load to flexure over idler rolls is a function of belt tension, type of material, shape of the load cross section, and idler spacing. Measurements indicate that the most important factor is belt tension, because this controls the amount of load flexure. Figure 6.2 shows this relationship for a typical idler spacing.

For a given weight per foot of belt and load, the running resistance, in pounds per ft of load, decreases with increases in belt tension. For a given belt tension, running resistance, in pounds per ft of load, increases with increases in the amount of load. However, the running resistance is not proportional to the weight of the load.

Table 6-4. *A* and *B* values for equation $K_y = (W_m + W_b) \times A \times 10^{-4} + B \times 10^{-2}$

Average Belt Tension, lbs	Idler Spacing, ft									
	3.0		3.5		4.0		4.5		5.0	
	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>
1,000	2.150	1.565	2.1955	1.925	2.200	2.250	2.2062	2.584	2.1750	2.910
2,000	1.8471	1.345	1.6647	1.744	1.6156	1.982	1.5643	2.197	1.5429	2.331
3,000	1.6286	1.237	1.4667	1.593	1.4325	1.799	1.4194	1.991	1.4719	2.091
4,000	1.4625	1.164	1.3520	1.465	1.3295	1.659	1.3250	1.825	1.3850	1.938
5,000	1.2828	1.122	1.1926	1.381	1.1808	1.559	1.1812	1.714	1.2283	1.839
6,000	1.1379	1.076	1.0741	1.318	1.0625	1.472	1.0661	1.627	1.0962	1.761
7,000	1.0069	1.039	0.9448	1.256	0.9554	1.404	0.9786	1.549	1.0393	1.657
8,000	0.9172	0.998	0.8552	1.194	0.8643	1.337	0.8875	1.472	0.9589	1.583
9,000	0.8207	0.958	0.8000	1.120	0.7893	1.272	0.8339	1.388	0.8911	1.507
10,000	0.7241	0.918	0.7362	1.066	0.7196	1.216	0.7821	1.314	0.8268	1.430
11,000	0.6483	0.885	0.6638	1.024	0.6643	1.167	0.7375	1.238	0.7768	1.340
12,000	0.5828	0.842	0.5828	0.992	0.6232	1.100	0.6750	1.180	0.7411	1.242
13,000	0.5207	0.798	0.5241	0.938	0.5732	1.040	0.6179	1.116	0.6821	1.169
14,000	0.4690	0.763	0.4810	0.897	0.5214	0.996	0.5571	1.069	0.6089	1.123
15,000	0.4172	0.718	0.4431	0.841	0.4732	0.935	0.5179	1.006	0.5607	1.063
16,000	0.3724	0.663	0.3966	0.780	0.4232	0.875	0.4589	0.958	0.5054	1.009

A minimum K_y value of .016 should be used when tensions exceed 16,000 lbs. Refer to page 92 for further explanations.

Information similar to that in Figure 6.2 has been developed by analyzing a series of field tests on belt conveyors of different widths carrying different materials. Many investigators, both in the United States and abroad, have analyzed similar series of field tests and have obtained similar results. Although the exact expressions differ, all investigators agree that changes in belt tension affect the force required to flex the material over idler rolls to a substantially greater degree than changes in the material handled. The latter does have a noticeable effect, and thus appears to be of less importance in the overall calculation.

Compilation of Components of T_e

The preceding pages describe the methods and provide the data for calculating factors K_t , K_x , and K_y . These factors must be evaluated as the first step to calculating certain components of belt tension that will be summarized to determine the effective tension, T_e , required at the driving pulley.

The procedures for calculating the belt tension components are as follows:

1. T_x — from the frictional resistance of the carrying and return idlers, lbs

$$T_x = L \times K_x \times K_t$$

(References: K_x — page 90, K_t — page 89)

2. T_{yb} — from the resistance of the belt to flexure as it moves over the idlers, lbs

$$T_{yc} \text{ — for carrying idlers: } T = L \times K_y \times W_b \times K_t$$

$$T_{yr} \text{ — for return idlers: } T_{ty} = L \times 0.015 \times w_b \times K_t$$

$$T_{yb} = T_{yc} + T_{yr}$$

$$T_{yb} = L \times K_y \times W_b \times K_t + L \times 0.015 \times W_b \times K_t$$

$$= L \times W_b \times K_t (K_y + 0.015)$$

(References: K_y — page 91, K_t — page 89)

3. T_{ym} — from resistance of the material to flexure as it rides the belt over the idlers, lbs

$$T_{ym} = L \times K_y \times W_m$$

(Reference: K_y — page 91)

4. T_m — from force needed to life or lower the load (material), lbs

$$T_m = \pm H \times W_m$$

5. T_p — from resistance of belt to flexure around pulleys and the resistance of pulleys to rotate on their bearings, lbs

Pulley friction arises from two sources. One source is the resistance of the belt to flexure over the pulleys, which is a function of the pulley diameter and the belt stiffness. The belt stiffness depends upon the ambient temperature and the belt construction.

The other source of pulley friction is the resistance of the pulley to rotate, which is a function of pillow block bearing friction, lubricant, and seal friction. The pillow block bearing friction depends upon the load on the bearings, but the lubricant and seal frictions generally are independent of load.

Since the drive pulley friction does not affect belt tension, it is not introduced into the mathematical calculation for belt tension; however, it must be included when determining the total horsepower at the motor shaft.

Table 6-5 provides conservative values for the pounds of belt tension required to rotate each of the pulleys on a conveyor. However, if a more precise value for belt tension to rotate pulleys is desired refer to **Appendix C**, page 352. Examples of belt tension and horsepower calculations shown in this book use values from Table 6-5.

$$T_p = \text{total of the belt tensions required to rotate each of the pulleys on the conveyor}$$

6. T_{am} — from force to accelerate the material continuously as it is fed onto the belt

Table 6-5. Belt tension to rotate pulleys.

Location of Pulleys	Degrees Wrap of Belt	Pounds of Tension at Belt Line
Tight side	150° to 240°	200 lbs/pulley
Slack side	150° to 240°	150 lbs/pulley
All other pulleys	less than 150°	100 lbs/pulley
Note: Double the above values for pulley shafts that are not operating in antifriction bearings.		

When material is discharged from chutes or feeders to a belt conveyor, it cannot be assumed that the material is moving in the direction of belt travel, at belt speed, although this may be the case in some instances. Normally, the material loaded onto the belt is traveling at a speed considerably lower than belt speed. The direction of material flow may not be fully in the direction of belt travel. Therefore, the material must be accelerated to the speed of the belt in the direction of belt travel, and this acceleration requires additional effective tension.

The belt tension T_{am} can be derived from the basic equation $F = MV_c$

where:

$$T_{am} = F = MV_c$$

M = mass of material accelerated per second, slugs

W = weight of material accelerated

$$= \frac{Q \times 2000}{3600}, \text{ lbs/sec}$$

Q = tph

g = 32.2 ft/sec²

$$M = \frac{W}{g} = \frac{Q \times 2000}{3600 \times 32.2}$$

V_c = velocity change, fps

$$= \frac{V - V_o}{60}$$

V = design belt speed, fpm

V_o = initial velocity of material as it is fed onto belt, fpm

$$T_{am} = \frac{Q \times 2000}{3600 \times 32.2} \times \frac{V - V_o}{60}$$

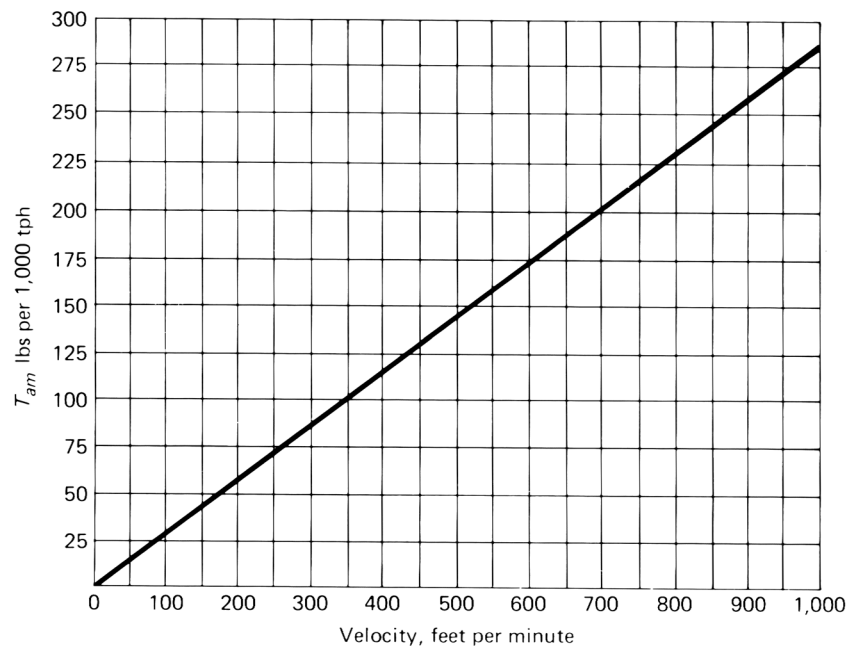
$$= 2.8755 \times 10^{-4} \times Q \times (V - V_o)$$

The graph in Figure 6.3 provides a convenient means of estimating the belt tension, T_{am} , for accelerating the material as it is fed onto the belt.

7. T_{ac} — from the resistance generated by conveyor accessories

Conveyor accessories such as trippers, stackers, plows, belt cleaning equipment, and skirtboards usually add to the effective tension, T_e . The additional belt tension requirements may come from frictional losses caused by the accessory. If the accessory lifts the conveyed material a force will be added to belt tension.

T_{tr} — from trippers and stackers



To use this chart:

- Enter chart at belt velocity and read T_{am} per 1,000 tph.
- Again enter chart at material velocity in direction of belt travel and read T_{am} per 1,000 tph. This may be positive, zero, or negative.
- Subtract the second T_{am} reading from the first T_{am} reading and convert the difference from 1,000 tph to the value for the actual tonnage. This will be the T_{am} desired, lbs.

Figure 6.3. Effective tension required to accelerate material as it is fed onto a belt conveyor.

The additional belt pull to flex the belt over the pulleys and rotate the pulleys in their bearings can be calculated from Table 6-5 or Tables C-1 and C-2.

The force needed to lift the material over the unit can be calculated from the formula, $T_m = H \times W_m$ lbs.

Frictional resistance of the idlers, belt, and material should be included with that of the rest of the conveyor.

T_{pl} — from frictional resistance of plows

The use of a plow on a conveyor will require additional belt tension to overcome both the plowing and frictional resistances developed.

While a flat belt conveyor may be fitted with a number of plows to discharge material at desired locations, seldom is more than one plow in use at one time on one run of the belt conveyor. However, when proportioning plows are used — with each plow taking a fraction of the load from the belt — two or even three separate plows may be simultaneously in contact with the carrying run of the belt.

To approximate the amount of additional belt pull that normally will be required by well-adjusted, rubber-shod plows, the values given in Table 6-6 can be used.

Table 6-6. Discharge plow allowance.

Type of Plow	Additional Belt Pull per Plow, at Belt Line (lbs/in belt width)
Full V or single slant plow, removing all material from belt	5.0
Partial V or single slant plow, removing half material from belt	3.0

T_{bc} — from belt-cleaning devices

Belt scraper cleaning devices add directly to the belt pull. The additional belt pull required for belt cleaning devices can vary from 2 to 14 lbs/in. width of scraper blade contact. This wide variance is due to the different types of cleaning blades and single cleaner system vs. multiple cleaner systems that are available. In lieu of data on specific cleaning system being used, use 5 lbs/in. width of scraper blade contact for each blade or scraper device in contact with the belt.

Rotary brushes and similar rotating cleaning devices do not impose appreciable belt pull, if independently driven and properly adjusted. If such devices are driven from the conveyor drive shaft, suitable additional power should be incorporated in the drive to operate them. Consult a CEMA member company for horsepower required.

T_{sb} — From skirtboard friction

The force required to overcome skirtboard friction is normally larger per foot of skirtboarded conveyor than the force to move the loaded belt over the idlers. In some cases, this force can be significant. When the total conveyor length is many times that portion of the length provided with the skirtboards, the additional power requirements for the skirtboards is relatively small, and in some cases negligible. However, if a large portion of the conveyor is equipped with skirtboards, the additional belt pull

required may be a major factor in the effective tension, T_e , required to operate the conveyor.

When the spacing of the skirtboards is two-thirds the width of the belt, the depth of the material rubbing on the skirtboards will not be more than 10 percent of the belt width, providing no more than a 20-degree surcharge load is carried on 20-degree troughing idlers.

Once the cross section of the load on the belt conveyor has been determined, the skirtboard friction can be calculated by determining the total pressure of the material against the skirtboard, then multiplying this value by the appropriate coefficient of friction of the material handled. The pressure of the material against the skirtboard can be calculated by assuming that the wedge of material contained between a vertical skirtboard and the angle of repose of the material is supported equally by the skirtboard and the belt.

This results in the following formula for conveyors whereon the material assumes its natural surcharge angle:

$$P = \frac{L_b d_m h_s^2}{288} \times \frac{(1 - \sin \phi)}{(1 + \sin \phi)}$$

where:

- P = total force against one skirtboard, lbs
- L_b = skirtboard length, ft one skirtboard
- d_m = apparent density of the material, lbs/cu ft
- h_s = depth of the material touching the skirtboard, in
- ϕ = angle of repose of material, degrees

Combining the apparent density, coefficient of friction, the angle of repose, and the constant into one factor for one type of material, the formula can be expressed:

$$C_s = \frac{2d_m}{288} \times \left(\frac{1 - \sin \phi}{1 + \sin \phi} \right)$$

$$T = (C_s)(L_b h_s^2) = C_s L_b h_s^2$$

where:

- T = belt tension to overcome skirtboard friction of two parallel skirtboards, lbs
- C_s = factor for the various materials in Table 6-7

Table 6-7. Skirtboard friction factor, C_s .

Material	Factor C_s	Material	Factor C_s	Material	Factor C_s
Alumina, pulv. dry	0.1210	Coke, ground fine	0.0452	Limestone, pulv., dry	0.1280
Ashes, coal, dry	0.0571	Coke, lumps and fines	0.0186	Magnesium chloride, dry	0.0276
Bauxite, ground	0.1881	Copra, lumpy	0.0203	Oats	0.0219
Beans, navy, dry	0.0798	Cullet	0.0836	Phosphate rock, dry, broken	0.1086
Borax	0.0734	Flour, wheat	0.0265	Salt, common, dry, fine	0.0814
Bran, granular	0.0238	Grains, wheat, corn or rye	0.0433	Sand, dry, bank	0.1378
Cement, Portland, dry	0.2120	Gravel, bank run	0.1145	Sawdust, dry	0.0086
Cement clinker	0.1228	Gypsum, 1/2" screenings	0.0900	Soda ash, heavy	0.0705
Clay, ceramic, dry fines	0.0924	Iron ore, 200 lbs/cu ft	0.2760	Starch, small lumps	0.0623
Coal, anthracite, sized	0.0538	Lime, burned, 1/8"	0.1166	Sugar, granulated dry	0.0349
Coal, bituminous, mined	0.0754	Lime, hydrated	0.0490	Wood chips, hogged fuel	0.0095

To this skirtboard friction must be added 3 lbs for every linear foot of each skirtboard, to overcome friction of the rubber skirtboard edging, when used, with the belt. Then,

$$\begin{aligned}
 T_{sb} &= T + 2L_b \times 3 \\
 &= C_s L_b h_s^2 + 2L_b \times 3 \\
 T_{sb} &= L_b (C_s h_s^2 + 6)
 \end{aligned}$$

Summary of Components of T_e

Once they have been compiled, the components of belt tension can be summarized to determine the effective tension, T_e required at the driving pulley.

T_e equals the total of the following:

$$\begin{aligned}
 &T_x, \text{ idler friction} &&= L \times K_x \times K_t \\
 + &T_{yc}, \text{ belt flexure, carrying idlers} &&= L \times K_y \times W_b \times K_t \\
 + &T_{yr}, \text{ belt flexure, return idlers} &&= L \times 0.015 \times W_b \times K_t \\
 &\text{Subtotal (A)} &&= LK_t(K_x + K_y W_b + 0.015 W_b) \\
 + &T_{ym}, \text{ material flexure} &&= L \times K_y \times W_m \\
 + &T_m, \text{ lift or lower} &&= \pm H \times W_m \\
 &\text{Subtotal (B)} &&= W_m(LK_y \pm H)
 \end{aligned}$$

$$\begin{aligned}
 &+ T_p, \text{ pulley resistance} \\
 &+ T_{am}, \text{ accelerated material} \\
 &+ T_{ac}, \text{ accessories} \\
 &\quad (T_{tr} + T_{pl} + T_{bc} + T_{sb}) \\
 &\text{Subtotal (C)} = (T_p + T_{am} + T_{ac}) \\
 T_e &= \Sigma \text{Subtotals (A), (B), and (C)}
 \end{aligned}$$

$$= LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) + T_p + T_{am} + T_{ac}$$

CEMA Horsepower Formula

Equation 1, page 86, provides the means for calculating the horsepower (hp) required by a belt conveyor having an effective tension, T_e , at the drive pulley and a design velocity, V , of the belt, as follows:

$$\text{hp} = \frac{T_e \times V}{33,000}$$

Combining equations (1) and (2) on pages 86-87, the hp load can be expressed as follows:

$$\text{hp} = \frac{V}{33,000} [LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) + T_p + T_{am} + T_{ac}]$$

The motor that will drive a fully loaded belt conveyor without becoming overheated may not be able to accelerate the loaded conveyor from rest to the design speed. To insure adequate starting capabilities, the following conditions must exist. First, the locked rotor torque of the motor should exceed the sum of the torque required to lift the material, plus approximately twice the torque required to overcome total conveyor friction, despite any possible voltage deficiencies that may exist during the acceleration period. This may not be true for long, horizontal conveyors or for declined conveyors.

Second, the motor speed-torque curve should not drop below a line drawn from the locked rotor torque requirement to the torque of the running horsepower requirement at full speed. This is further explained in Chapter 13, "Motors and Controls."

For examples illustrating the use of the equations in determining the effective belt tension, T_e , at the drive pulley and the horsepower to operate the belt conveyor, refer to the two problems on pages 145 through 152.

It is also possible to arrive at a close approximation of the horsepower required to operate a belt conveyor by means of a graphical solution. This method, used under proper circumstances, is quick and relatively simple. Generally, a graphical solution will provide a somewhat conservative value of required horsepower. However, it must be recognized that it is impractical to incorporate all elements of belt conveyor design into a simple graphical solution. Therefore, the graphs should be used based on a

complete understanding of all aspects of the analytical method of calculating belt conveyor tension and horsepower, in order to allow for adjustment of the results to account for unusual situations. It is recommended that final design be based on calculations made by the analytical method. The graphical method of designing belt conveyors is described on pages 141-145.

Drive Pulley Relationships

The force required to drive a belt conveyor must be transmitted from the drive pulley to the belt by means of friction between their two surfaces. The force required to restrain a downhill regenerative conveyor is transmitted in exactly the same manner. In order to transmit power, there must be a difference in the tension in the belt as it approaches and leaves the drive pulley. This difference in tensions is supplied by the driving power source. Figures 6.4 and 6.5 illustrate typical arrangements of single pulley drives.

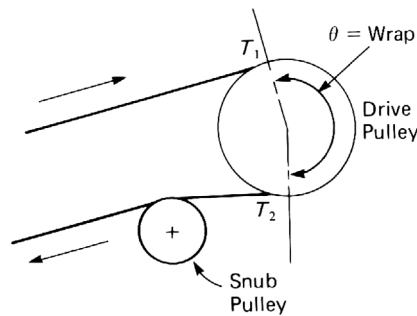


Figure 6.4 Inclined or horizontal conveyor, pulley driving belt.

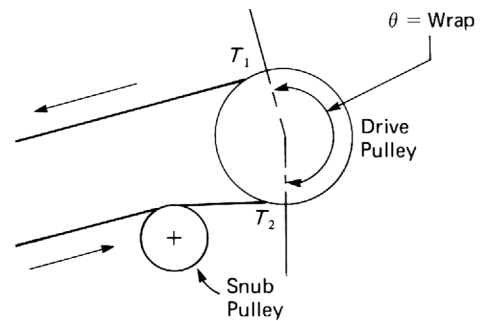


Figure 6.5 Declined conveyor. Lowering load with regeneration, belt driving pulley.

It should be noted that if power is transmitted from the pulley to the belt, the approaching portion of the belt will have the larger tension, T_1 , and the departing portion will have the smaller tension, T_2 . If power is transmitted from the belt to the pulley, as with a regenerative declined conveyor, the reverse is true. Wrap is used here to refer to the angle or arc of contact the belt makes with the pulley's circumference.

Wrap Factor, C_w

The wrap factor, C_w , is a mathematical value used in the determination of the effective belt tension, T_e , that can be dependably developed by the drive pulley. The T_e that can be developed is governed by the coefficient of friction existing between the pulley and the belt, wrap, and the values of T_1 and T_2 .

The following symbols and formulas are used to evaluate the drive pulley relationships:

$$T_e = T_1 - T_2 = \text{effective belt tension, lbs}$$

$$T_1 = \text{tight-side tension at pulley, lbs}$$

$$T_2 = \text{slack-side tension at pulley, lbs}$$

$$e = \text{base of naperian logarithms} = 2.718$$

$$f = \text{coefficient of friction between pulley surface and belt surface (0.25 rubber surfaced belt driving bare steel or cast iron pulley; 0.35 rubber sur-$$

faced belt driving rubber lagged pulley surface). Values apply to normal running calculations.

θ = wrap of belt around the pulley, radians (one degree = 0.0174 radians)

C_w = wrap factor (see Table 6-8)

$$= \frac{T_2}{T_e} = \frac{1}{e^{f\theta} - 1}$$

It should be noted that the wrap factors do not determine T_2 but only establish its safe minimum value for a dry belt. A wet belt and pulley will substantially reduce the power that can be transmitted from the one to the other because of the lower coefficient of friction of the wet surfaces. Various expedients, such as grooving the lagging on the pulley, lessen this problem. However, the best solution is to keep the driving side of the belt dry. If this is impractical, increasing the wrap is helpful, or providing some means of increasing the slack side tension, T_2 . This can be done, for example, by increasing the counterweight in a gravity takeup.

Wrap Factor with Screw Takeup

When a screw takeup is used, Table 6-8 indicates an increased wrap factor. This increased wrap factor is necessary to provide sufficient slack side tension, T_2 , to drive the conveyor in spite of the amount of stretch in the conveyor belt, for which the screw takeup makes no automatic provision.

Table 6-8. Wrap factor, C_w (Rubber-surfaced belt).

Type of Pulley Drive	θ Wrap	Automatic Takeup		Manual Takeup	
		Bare Pulley	Lagged Pulley	Bare Pulley	Lagged Pulley
Single, no snub	180°	0.84	0.50	1.2	0.8
Single with snub	200°	0.72	0.42	1.0	0.7
	210°	0.66	0.38	1.0	0.7
	220°	0.62	0.35	0.9	0.6
	240°	0.54	0.30	0.8	0.6
Dual*	380°	0.23	0.11	0.5	0.3
	420°	0.18	0.08	—	—

*Dual values based on ideal distribution between primary and secondary drive.

For wet belts and smooth lagging, use bare pulley factor.

For wet belts and grooved lagging, used lagged pulley factor.

If wrap is unknown, assume the following:

Type of Drive	Assumed Wrap
Single—no snub	180°
Single —with snub	210°
Dual	380°

Wrap θ (Arc of Contact)

So far, it has been shown that the relationship between the values known as T_1 , the tight side tension (and generally the tension for which the belt must be designed and built), and T_2 , the slack side tension (the minimum value that must be available for driving the belt successfully), is influenced by the angle of wrap of the belt around the drive pulley and by the coefficients of friction established by the belt and pulley surfaces as they make contact. It has been indicated that the coefficient of friction may vary when driving a rubber surfaced belt by a bare steel or cast iron pulley, or by a rubber-lagged pulley surface.

The angle of wrap of the belt around the drive pulley can be varied by the use of a snub pulley or, for larger angles of wrap, by supplying power, under the proper conditions, to more than one drive pulley.

The wrap limits for various types of pulley drives can be determined from Table 6-9.

Table 6-9. Wrap limits.

Type of Pulley Drive	Wrap Limits*	
	From	To
Single—no snub	180°	180°
Single—with snub	180°	240°
Dual	360°	480°

*The above wrap angles apply to either bare or lagged pulleys.

For most cases, the belt will have an angle of wrap around the drive pulley of about 180 degrees to 240 degrees. Often, it will be necessary to arrange a drive that uses an angle of wrap greater than 180 degrees. This is accomplished by the appropriate positioning of a snub pulley, which can extend the angle of wrap to 240 degrees. However, its use is subject to the following limitations: (1) Snub pulley diameter is limited by the belt specifications; (2) In order to thread through a new or replacement belt between the pulleys, a suitable clearance between pulley rims should be allowed; (3) The departing direction of the belt from the snub pulley (plus clearance for belt fasteners, etc.) must be below the deck plate or on the underside of the carrying idlers. These limitations will be found to restrict a snubbed drive to an angle of wrap not exceeding 240 degrees, in most cases. If a greater angle of wrap is necessary, it may be necessary to use a dual-pulley drive.

Dual-Pulley Drives

A dual-pulley drive uses two or more separate motors, one or more driving the primary drive pulley and one or more driving the secondary drive pulley. Table 6-8, Wrap Factor C_w , shows the major increase in wrap that becomes available when using a dual-pulley drive. This increase in available wrap can often provide for lower maximum belt tension and a more efficient and lower cost conveyor design.

In any such system where two drive pulleys are involved, the secondary pulley starts out with a certain value of T_2 . Contingent on its angle of wrap and the applicable friction coefficient, the secondary pulley can produce a value, T_3 , such that:

$$T_1 - T_3 = T_{ep} \text{ (primary)}$$

$$T_3 - T_2 = T_{es} \text{ (secondary)}$$

$$T_{ep} \text{ (primary)} + T_{es} \text{ (secondary)} = T_e \text{ (total for conveyor)}$$

The value, T_3 , for the secondary pulley, clearly is the only value available to be used as the slack side tension in the preceding primary drive. This value of T_3 , added to the T_{ep} for the primary pulley, yields T_1 . The sum of the secondary T_{es} and the primary T_{ep} yields a total T_e that the combined two-pulley drive can produce.

For the maximum efficiency of a two-pulley or dual drive, as described above, it is evident that the proportionate size of the two motors employed must be related appropriately to the angles of wrap and the coefficients of friction at the respective pulleys.

The ratio of tight side tension/slack side tension for each of the drive components, when multiplied together, gives the constant that would apply to the combined or total drive. Or, putting this in another way, T_1/T_3 multiplied by T_3/T_2 will equal T_1/T_2 , provided the drive conditions are the same for both pulleys. However, if the primary drive utilizes the clean side of the belt while the secondary drive is permitted to operate on the carrying or dirty side of the belt, the friction coefficient and wrap factor for the secondary pulley will vary and the tension relationship should be investigated.

For any conveyor drive that utilizes more than one drive pulley, a snub pulley arrangement is preferable, so that both pulleys drive on the same clean side of the belt.

The following symbols and formulas will be of assistance in evaluating the drive pulley relationships for dual pulley drives:

T_3 = belt tension between the primary and secondary drive pulleys

C_{ws} = wrap factor for the secondary drive pulley

C_{wp} = wrap factor for the primary drive pulley

C_w = the combined wrap factor for both drive pulleys

T_{es} = effective tension on the secondary drive pulley

T_{ep} = effective tension on the primary drive pulley

$$T_2 = T_e C_w$$

$$T_1 = T_2 + T_e$$

$$T_1 = T_{ep} + T_3$$

$$T_3 = T_2 + T_{es}$$

$$C_{wp} = \frac{T_3}{T_{ep}}$$

$$C_{ws} = \frac{T_3}{T_{es}} - 1$$

$$T_e = T_{es} + T_{ep}, \text{ whence}$$

$$T_2 = (T_{es} + T_{ep})(C_w)$$

$$T_{es} = \frac{T_2}{C_{ws}} \text{ by definition}$$

$$T_{ep} = \frac{T_3}{C_{wp}} \text{ by definition}$$

$$T_3 = T_2 + T_{es}, \text{ whence}$$

$$T_{ep} = \frac{T_2 + T_{es}}{C_{wp}}$$

Substituting:

$$T_2 = \left[\frac{T_2}{C_{ws}} + \frac{(T_2 + T_{es})}{C_{wp}} \right] C_w$$

$$= \left(\frac{T_2}{C_{ws}} + \frac{T_2}{C_{wp}} + \frac{T_{es}}{C_{wp}} \right) C_w$$

$$T_2 = \left(\frac{T_2 C_{wp} + T_2 C_{ws} + T_{es} C_{ws}}{C_{ws} C_{wp}} \right) C_w$$

solving for C_w , and noting that $T_{es} C_{ws} = T_2$,

$$C_w = \frac{T_2 C_{ws} C_{wp}}{T_2 C_{wp} + T_2 C_{ws} + T_2}, \text{ and because } T_{es} C_{ws} = T_2,$$

$$C_w = \frac{C_{ws} C_{wp}}{C_{wp} + C_{ws} + 1}$$

For example, if the angles of wrap of the primary and secondary drive pulleys are 180 degrees and 220 degrees, respectively, the factors are as follows for lagged pulleys (see Table 6-8):

$$C_{ws} = 0.35 \text{ for } 220^\circ \text{ angle of wrap}$$

$$C_{wp} = 0.50 \text{ for } 180^\circ \text{ angle of wrap}$$

$$C_w = 0.095 \text{ for } 400^\circ \text{ total angle of wrap by interpolating between } 380^\circ \text{ and } 420^\circ, \text{ or:}$$

$$C_w = \frac{C_{wp}C_{ws}}{1 + C_{wp} + C_{ws}} = \frac{(0.5)(0.35)}{1 + 0.35 + 0.50} = 0.0945$$

The tensions exterior to a dual-pulley drive are the same as those for a single-pulley drive.

A part of the effective tension, T_e is taken on the primary drive pulley and a part on the secondary drive pulley. Using two motors, the ratio of T_{ep} to T_{es} is the ratio of the horsepower ratings of the two motors.

For example, if the total calculated horsepower is 250, this could be supplied, allowing for drive losses, by using a 200-horsepower primary drive and a 75-horsepower secondary drive, with a drive efficiency of 90 percent.

The primary pulley would take

$$(200/275)(250) = 182\text{hp}$$

The secondary pulley would take

$$(75/275)(250) = 68\text{hp}$$

If the belt velocity, V , is 400 fpm, then

$$T_{ep} = \frac{(182)(33,000)}{400} = 15,000 \text{ lbs}$$

$$T_{es} = \frac{(68)(33,000)}{400} = 5,625 \text{ lbs}$$

and

$$\frac{T_{ep}}{T_{es}} = \frac{15,000}{5,625} = 2.67$$

(see Problem 1, page 145.)

Drive Arrangements

The final selection and design of a conveyor drive arrangement is influenced by many factors, including the performance requirements, the preferred physical location, and relative costs of components and installation.

Figures 6.6A through 6.7F illustrate some of the drive combinations that have been furnished. Other arrangements may be better suited to a particular conveyor in a particular location. CEMA member companies can assist in final recommendations.

Note that the illustrated arrangements that are on a substantially downhill run are usually regenerative and are so indicated in the title.

Figure 6.6 Single-pulley/drive arrangements.

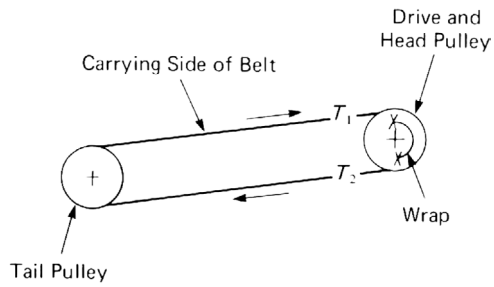


Figure 6.6A Single-pulley drive at head end of conveyor without snub pulley.

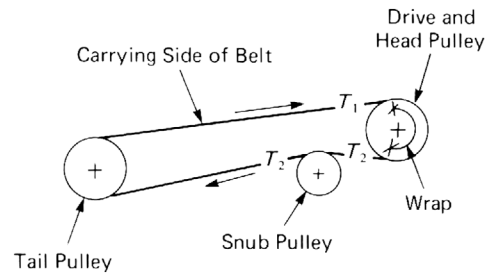


Figure 6.6B Single-pulley drive at head end of conveyor with snub pulley.

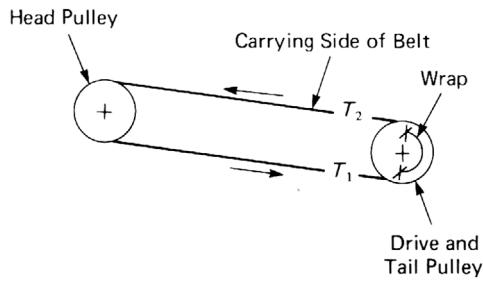


Figure 6.6C Single-pulley drive at tail end of conveyor without snub pulley. Used when head end drive cannot be applied.

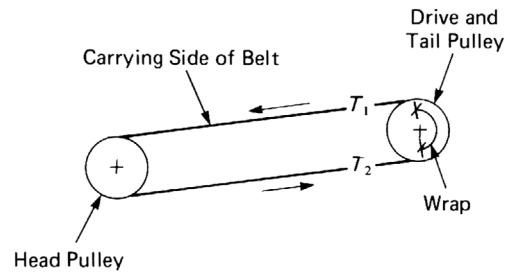


Figure 6.6D Single-pulley drive at tail end of conveyor without snub pulley; regenerative.

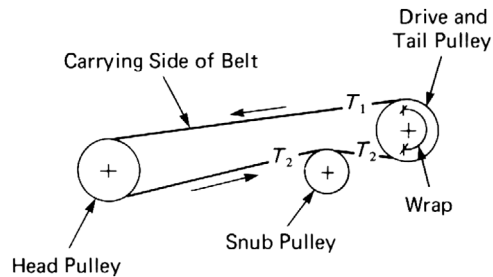


Figure 6.6E Single-pulley drive at tail end of conveyor with snub pulley; regenerative.

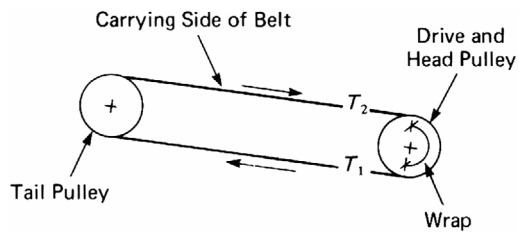


Figure 6.6F Single-pulley drive at head end of conveyor without snub pulley; regenerative.

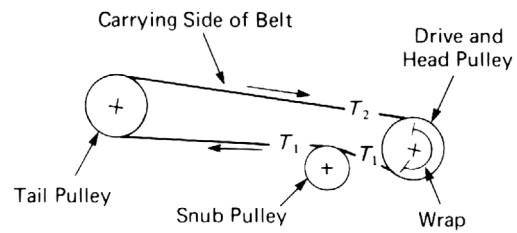


Figure 6.6G Single-pulley drive at head end of conveyor with snub pulley; regenerative.

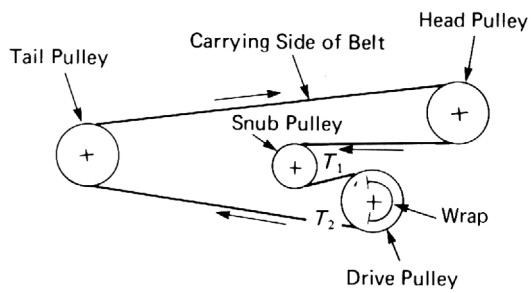


Figure 6.6H Single-pulley drive on return run.

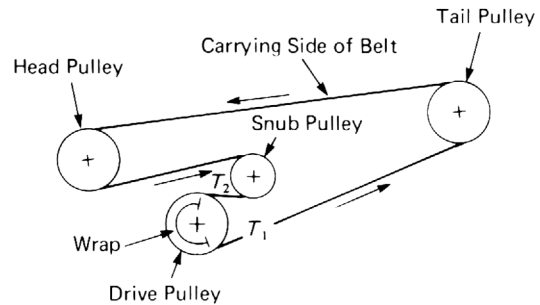


Figure 6.6I Single-pulley drive on return run; regenerative.

Figure 6.7 Dual-pulley drive arrangements.

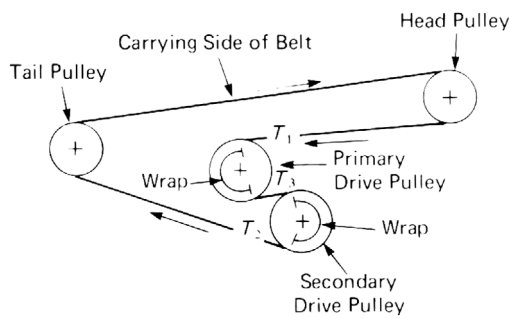


Figure 6.7A Dual-pulley drive on return run.

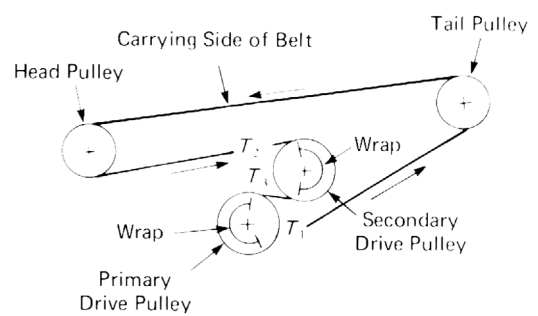


Figure 6.7B Dual-pulley drive on return run; regenerative.

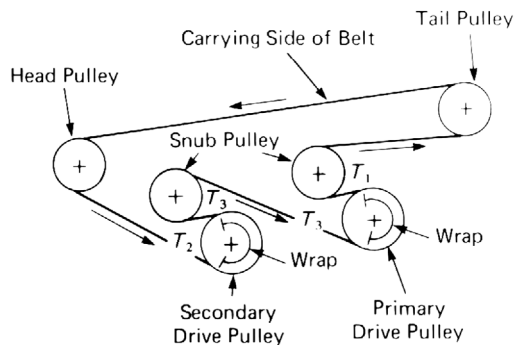


Figure 6.7C Dual-pulley drive on return run; regenerative.

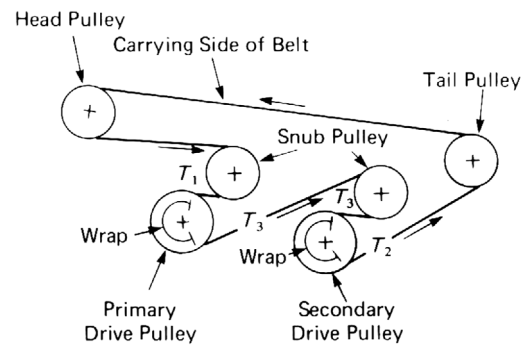


Figure 6.7D Dual-pulley drive on return run. Drive pulleys engage clean side of belt.

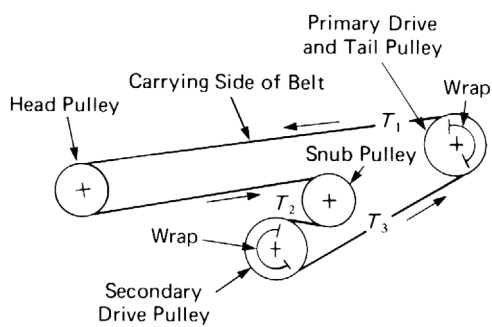


Figure 6.7E Dual-pulley drive with primary drive on tail pulley of conveyor; regenerative.

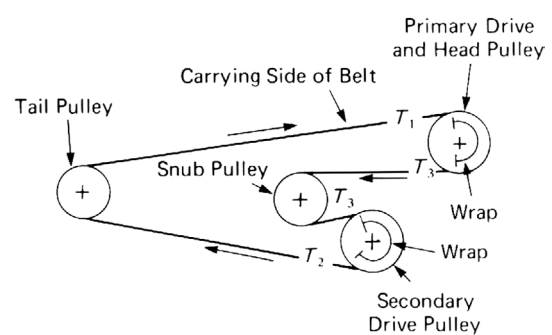


Figure 6.7F Dual-pulley drive with primary drive on head pulley of conveyor.

Maximum and Minimum Belt Tensions

For the illustrated common conveyor profiles and drive arrangements, minimum and maximum tensions will be discussed and procedures given for calculating the belt tension at any point in the conveyor. The applicable formulas are indicated with the various profile and drive arrangements where single-pulley drives are involved. The tensions involved in multiple-pulley drives are treated separately.

Maximum Belt Tension

Operating Maximum Belt Tension. The operating maximum belt tension is defined as the maximum belt tension occurring when the belt is conveying the design load from the loading point continuously to the point of discharge. Operating maximum tension usually occurs at the discharge point on horizontal or inclined conveyors and at the loading point on regenerative declined conveyors. On compound conveyors, the operating maximum belt tension frequently occurs elsewhere. Because the operating maximum belt tension must be known to select a belt, its location and magnitude must be determined. For details on belt tensions, refer to Figures 6.8 through 6.16.

Conveyors having horizontal and lowering, or horizontal and elevating, sections can have maximum tensions at points other than a terminal pulley. In this case, belt tensions can be calculated by considering the horizontal and sloping sections as separate conveyors.

Temporary Operating Maximum Belt Tension. A temporary operating maximum belt tension is that maximum tension which occurs only for short periods. For example, a conveyor with a profile that contains an incline, a decline, and then another incline, may generate a higher operating tension when only the inclines are loaded and the decline is empty. These temporary operating maximum belt tensions should be considered in the selection of the belt and the conveyor machinery.

Starting and Stopping Maximum Tension

The starting torque of an electric motor may be more than $2\frac{1}{2}$ times the motor full-load rating. Such a torque transmitted to a conveyor belt could result in starting tensions many times more than the chosen operating tension. To prevent progressive weakening of splices and subsequent failure, such starting maximum tensions should be avoided. Refer to Chapter 13. Likewise, if the belt is brought to rest very rapidly, especially on decline conveyors, the inertia of the loaded belt may produce high tensions.

The generally recommended maximum for starting belt tension is 150 percent of the allowable belt working tension. On conveyors with tensions under 75 lbs/ply in or the equivalent, the maximum can be increased to as high as 180 percent. For final design allowances, conveyor equipment or rubber belt manufacturers should be consulted.

Minimum Belt Tension, T_{min}

For conveyors that do not overhaul the drive, the minimum belt tension on the carrying run will usually occur at the tail (feed) end. For conveyors that do overhaul their drive, the minimum belt tension will usually occur at the head (discharge) end. The locations and magnitude of minimum belt tensions are given in connection with the conveyor profiles and drives shown in Figures 6.8 through 6.16.

It will be seen that the minimum tension is influenced by the T_2 tension required to drive, without slippage of the belt on the pulley, and by the T_0 tension required to

limit the belt sag at the point of minimum tension. The minimum tension is calculated both ways and the larger value used. If T_0 to limit belt sag is larger than the T_{min} produced by the T_2 tension necessary to drive the belt without slippage, a new T_2 tension is calculated, using T_0 and considering the slope tension, T_b , and the return belt friction, T_{yr} . Formulas for calculating T_2 , having T_0 , T_b , and T_{yr} are given for each of the conveyor profiles and drive arrangements.

Tension Relationships and Belt Sag Between Idlers

Chapter 5, “Belt Conveyor Idlers,” presents the basic facts on the subject of idler spacing. A major requirement, noted in Chapter 5, is that the sag of the belt between idlers must be limited to avoid spillage of conveyed material over the edges of the belt. The sag between idlers is closely related to the weight of the belt and material, the idler spacing, and the tension in the belt.

Graduated Spacing of Troughing Idlers

For belt conveyors with long centers, it is practical to vary the idler spacing so as to equalize the catenary sag of the belt as the belt tension increases.

The basic equation for the sag in a catenary can be written:

$$\text{Sag, ft} = \frac{WS_i^2}{8T}$$

where :

W = weight, $(W_b + W_m)$, lbs/ft of belt and material

S_i = idler spacing, ft

T = tension in belt, lbs

The basic sag formula can also be expressed as a relation of belt tension, T , idler spacing, S_i , and the weight per foot of belt and load, $(W_b + W_m)$, in the form:

y = vertical drop (sag) between idlers, ft

$$y = \frac{S_i^2(W_b + W_m)}{8T}$$

Experience has shown that when a conveyor belt sags more than 3 percent of the span between idlers, load spillage often results. For 3 percent sag the equation becomes:

$$\frac{S_i^2(W_b + W_m)}{8T} = \frac{3S_i}{100}$$

While pure catenary equations are used, the allowable percent sag takes into account such factors as stiffness of the belt carcass, strength of the belt span due to the “channel” shape of a troughed belt, etc.

Simplifying for minimum tension to produce various percentages of belt sag yields the following formulas:

$$\text{For 3 percent sag, } T_0 = 4.2S_i(W_b + W_m)$$

$$\text{For 2 percent sag, } T_0 = 6.25S_i(W_b + W_m)$$

$$\text{For } 1\frac{1}{2} \text{ percent sag, } T_0 = 8.4S_i(W_b + W_m)$$

See Table 6-10 for recommended belt sag percentages for various full load conditions.

The graduated spacing should be calculated to observe the following limitations: (1) A maximum of 3 percent sag should be maintained when belt is operating with a normal load. (2) A maximum of 4.5 percent sag should be maintained when the loaded belt is standing still. (3) The idler spacing should not exceed twice the suggested normal spacing of the troughing idlers listed in Table 5-2. (4) The load on any idler should never exceed the idler load ratings given in Chapter 5.

Moreover, the number of spacing variations must be based on practical considerations, such as the number of different stringer sections in the conveyor support structure, so that the fabrication cost of the support structure does not become excessive. Usually, the spacing of troughing idlers is varied in 6-inch increments.

Limiting the calculated belt sag to 3 percent of the idler spacing, at any point on the conveyor, will generally prevent spillage of the material from conveyor belts operating over 20 degrees troughing idlers.

When handling lumpy material on belts operating on 35 degrees (or deeper) troughing idlers, belt tension should be increased to reduce the percent of sag. Deep-troughed conveyor belts normally carry a relatively large cross-sectional loading and corresponding heavy weight of material per foot of length. Therefore, the material exerts a greater pressure against the side of the trough, tending to cause greater transverse belt flexure. The purpose of increasing the minimum belt tension in belts operating on idlers of greater than 20 degrees troughing angle is to keep this transverse belt flexure to an acceptable minimum and thus prevent spillage.

Similarly, when frequent surge loads are encountered or a substantial percentage of large lumps is expected, the material weight per foot of conveyor will be increased. Consideration of increased minimum belt tension at, or closely adjacent to, the loading points is recommended.

Table 6-10. Recommended belt sag percentages for various full load conditions.

Idler Troughing Angle	Material		
	All Fines	One-half the Maximum Lump Size*	Maximum Lump Size*
20°	3%	3%	3%
35°	3%	2%	2%
45°	3%	2%	1½%

*See Figure 4.1

Note: Reduced load cross sections will permit an increase in the sag percentage, resulting in a decreased minimum tension. Such a choice may lead to a more economical belt selection, as the maximum tension will be reduced correspondingly.

Slack Side Tension, T_2 . The minimum tension required to drive the belt without slippage is the product of T_e and C_w . However, the value to be used for minimum belt tension on the carrying run is either T_0 (calculated as above) plus or minus the tension T_b and plus or minus the return belt friction T_{yr} , or the minimum tension to drive without slippage $T_e \times C_w$. By rearranging and substituting terms,

$$T_2 = T_0 \pm T_b \pm T_{yr}$$

or, by the above definition,

$$T_2 = T_e C_w$$

Use the larger value of T_2 .

Tension, T_b . The weight of the carrying and/or return run belt for a sloped conveyor is carried on the pulley at the top of the slope. This must be considered in calculating the T_2 tension, as indicated above.

$$T_b = HW_b$$

where:

W_b = weight of belt, lbs/ft

H = net change in elevation, ft

Return Belt Friction Tension, T_{yr} . The return belt friction is the belt tension resulting from the empty belt moving over the return run idlers:

$$T_{yr} = 0.015LW_bK_t$$

where :

L = length, ft, of conveyor to center of terminal pulleys

K_t = temperature correction factor as defined on page 89

For temperatures above 32°F, $K_t = 1.0$

Belt Tensions for Conveyors of Marginal Decline. Allowances made for frictional losses in a conveyor are intended as conservative assumptions. When a declined conveyor is involved, such allowances should be discounted when considered in conjunction with maximum possible regenerative tensions (or hp).

Belt Tensions for Typical Conveyors. When calculating tensions at any point in these conveyor profiles, the portions of the conveyor on zero slope, inclines, or declines should be considered as separate conveyors.

Belt Tension at Any Point, X, on Conveyor Length. In order to understand clearly the formulas for evaluating the belt tension at any point, X, on the belt conveyor length, it is necessary to establish the following nomenclature:

L_x = distance, ft, from tail pulley to point X along the conveyor

H_x = vertical distance, ft, from tail pulley to point X

T_{cx} = belt tension, lbs, at point X on the carrying run

T_{rx} = belt tension, lbs, at point X on the return run

T_t = belt tension, lbs, at tail pulley

T_{hp} = belt tension, lbs, at head pulley

T_{wcx} = tension, lbs, at point X on the carrying run, resulting from the weight of belt and material carried

T_{fcx} = tension, lbs, at point X on the carrying run, resulting from friction

T_{wrx} = tension, lbs, at point X on the return run, resulting from the weight of the empty belt

T_{frx} = tension, lbs, at point X on the return run, resulting from friction

$T_{wcx} = H_x (W_b + W_m)$

$T_{fcx} = L_x [K_t K_x + K_y W_b] + L_x K_y W_m$

$T_{wrx} = H_x W_b$

$T_{frx} = 0.015 L_x W_b K_t$

Formulas for T_{cx} and T_{rx} are given for all the belt conveyor profiles and drives in Figures 6.8 through 6.16. [Tensions from conveyor accessories (T_{ac}) have been omitted for clarity; however, these should be accounted for in the final design.]

Analysis of Belt Tensions

In addition to calculation of the effective belt tension, T_e , which occurs at the drive pulley, a designer must consider the belt tension values that occur at other points of the conveyor's belt path.

Figures 6.8 through 6.16 illustrate various possible conveyor layouts and profiles and the appropriate tension analysis. Some of these examples are more commonly applied than others; the order of presentation is not intended to infer preference of design. Many of these diagrams illustrate the takeup, TU , in alternate locations. It is most unusual for a conveyor to employ more than one takeup; a preferred single location should be chosen.

Figure 6.8 Head pulley drive — horizontal or elevating.*

$$T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_t + T_b - T_{yr}$$

Use the larger value of T_2

$$T_t = T_0 \text{ or}$$

$$T_t = T_2 - T_b + T_{yr}$$

Use the larger value of T_t

$$T_t = T_{min} \quad T_1 = T_{max}$$

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

$$T_{rx} = T_t + T_{wrx} - T_{frx}$$

*With decline conveyors the T_e tension required for an empty conveyor may sometimes be greater than the T_e for the loaded conveyor.

Figure 6.8A Inclined conveyor with head pulley drive.

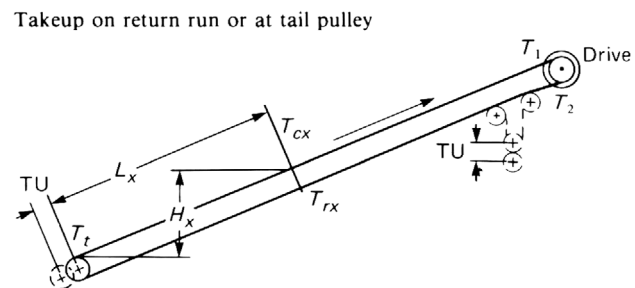


Figure 6.8B Horizontal belt conveyor with concave vertical curve, and head pulley drive.

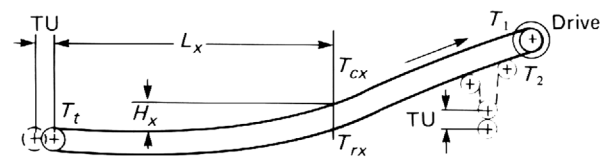
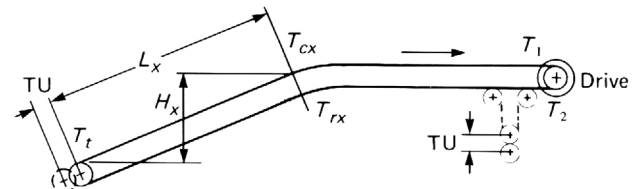


Figure 6.8C Horizontal belt conveyor with convex vertical curve, and head pulley drive.



NOTE: Two takeups are shown only to illustrate alternative. Two automatic takeups cannot function properly on the same conveyor.

Figure 6.9 Head pulley drive — lowering without regenerative load.*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o - T_e \text{ or } T_2 = T_o - T_b - T_{yr}$$

Use the larger value of T_2

$$T_1 = T_e + T_2$$

$$T_t = T_2 + T_b + T_{yr} \text{ or } T_t = T_o + T_b + T_{yr} - T_e \text{ or } T_t = T_o$$

$$T_{max} = T_t \text{ or } T_1$$

$$T_{min} = T_t + T_1$$

$$T_{cx} = T_t - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_t - T_{wrx} - T_{frx}$$

*With decline conveyors the T_e tension required for an empty conveyor may sometimes be greater than the T_e for the loaded conveyor.

Figure 6.9A Declined belt conveyor with head pulley drive. Lowering without regenerative load.

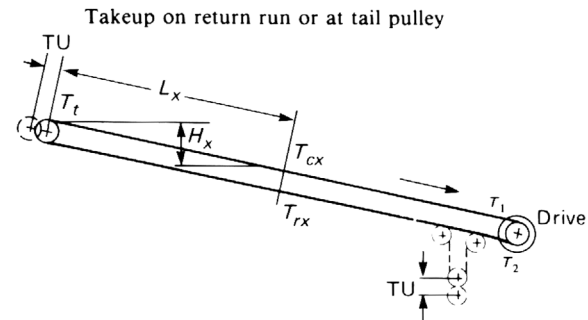


Figure 6.9B Conveyor with convex vertical curve, head pulley drive. Lowering without regenerative load.

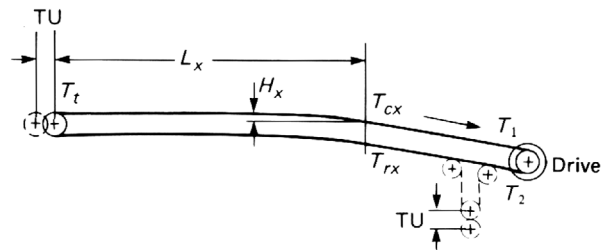
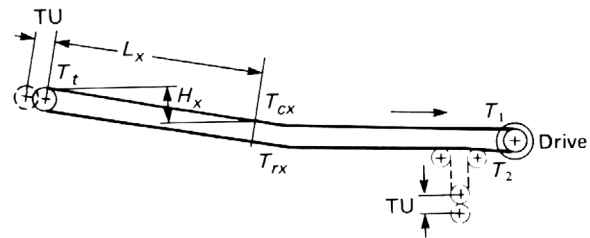


Figure 6.9C Conveyor with concave vertical curve, head pulley drive. Lowering without regenerative load.



NOTE: Two takeups are shown only to illustrate alternative. Two automatic takeups cannot function properly on the same conveyor.

Figure 6.10 Head pulley drive — lowering with regenerative load.

$$^* T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o$$

Use the larger value of T_2 .

$$T_t = T_{max} = T_1 + T_b + T_{yr} \text{ or } T_t = T_e + T_o + T_b + T_{yr}$$

$$T_{min} = T_2$$

$$T_{cx} = T_t - T_{wcx} + T_{frx}$$

$$T_{rx} = T_t - T_{yw_{rx}} - T_{frx}$$

*See page 119.

Takeup on return run not recommended to avoid driving through the takeup.

Figure 6.10A Declined belt conveyor with head pulley drive. Lowering with regenerative load.

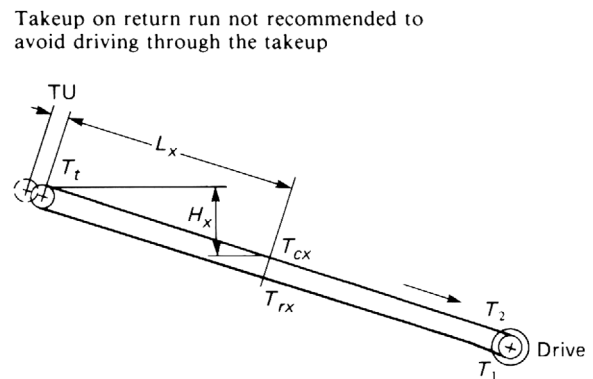


Figure 6.10B Conveyor with convex vertical curve, head pulley drive. Lowering with regenerative load.

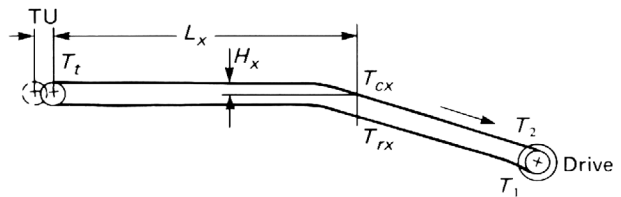


Figure 6.10C Conveyor with concave vertical curve, head pulley drive. Lowering with regenerative load.

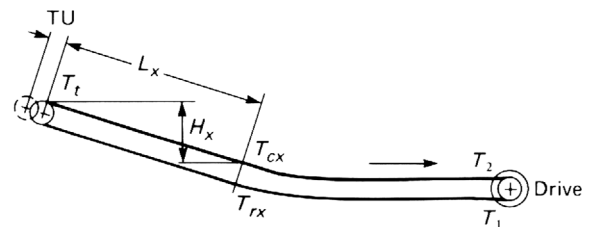


Figure 6.11 Tail pulley drive — horizontal or elevating.

$$T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_0$$

Use the larger value of T_2

$$T_t = T_2$$

$$T_{min} = T_2$$

$$T_{hp} = T_1 - T_{yr} + T_b$$

$$T_{max} = T_1 \text{ or } T_{max} = T_{hp} \text{ Use the larger value of } T_{max}$$

$$T_{cx} = T_2 + T_{wcx} + T_{fcx}$$

$$T_{rx} = T_1 + T_{wrx} - T_{frx}$$

With a tail pulley drive more of the belt is under high tension and it is usually not practical to locate takeup at T_2 just before the loading point. Calculate belt tension at takeup during acceleration and make adequate to avoid belt slip.

Figure 6.11A Inclined conveyor with tail pulley drive.

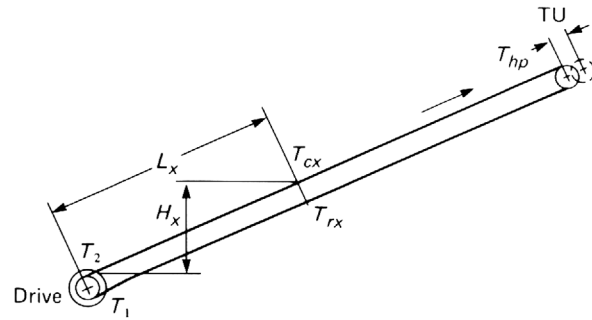


Figure 6.11B Horizontal belt conveyor with concave vertical curve and tail pulley drive.

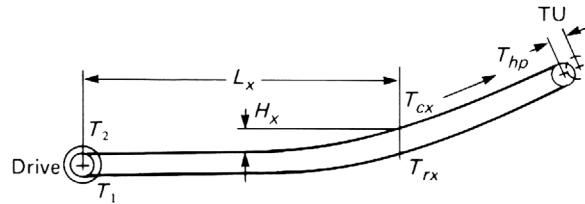


Figure 6.11C Horizontal belt conveyor with convex vertical curve and tail pulley drive.

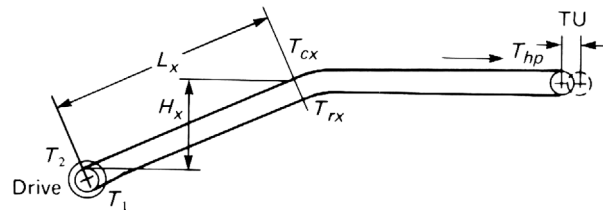


Figure 6.12 Tail pulley drive — lowering without regenerative load.*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o + T_b + T_{yr} - T_e$$

Use the larger value of T_2

$$T_{hp} = T_o \text{ or } T_{hp} = T_1 - T_b - T_{yr}$$

Use the larger value of T_{hp}

$$T_{hp} = T_{min}$$

$$T_1 = T_e + T_2 = T_{max}$$

$$T_{cx} = T_2 - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_1 - T_{wrx} - T_{frx}$$

*See page 119.

Takeup on return run not recommended to avoid driving through the takeup.

Figure 6.12A Declined belt conveyor with tail pulley drive. Lowering without regenerative load.

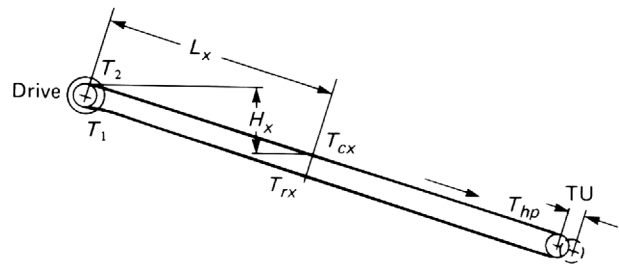


Figure 6.12B Conveyor with convex vertical curve, tail pulley drive. Lowering without regenerative load.

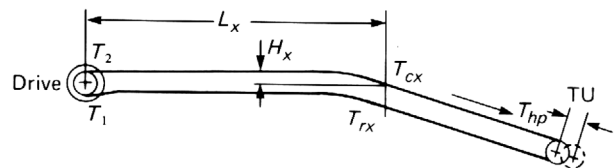


Figure 6.12C Conveyor with concave vertical curve, tail pulley drive. Lowering without regenerative load.

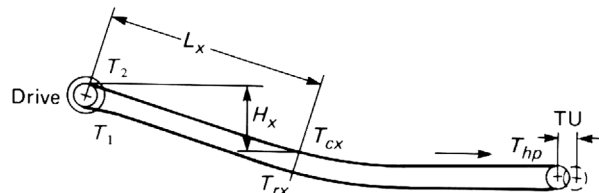


Figure 6.13 Tail pulley drive — lowering with regenerative load.*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o + T_b + T_{yr}$$

Use the larger value of T_2

$$T_{hp} = T_2 - T_b - T_{yr} \text{ or } T_{hp} = T_o$$

$$T_{hp} = T_{min}$$

$$T_1 = T_{max} = T_e + T_2$$

$$T_{cx} = T_1 - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_2 - T_{wrx} - T_{frx}$$

*Calculate belt tension required at takeup during acceleration and make takeup adequate to prevent lift-up. See page 119.

Figure 6.13A Declined belt conveyor with tail pulley drive. Lowering with regenerative load.

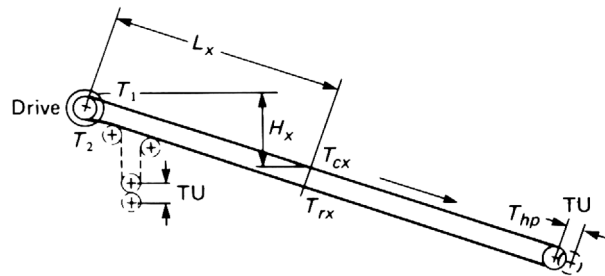


Figure 6.13B Conveyor with concave vertical curve, tail pulley drive. Lowering with regenerative load.

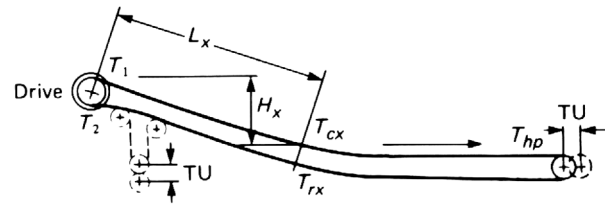
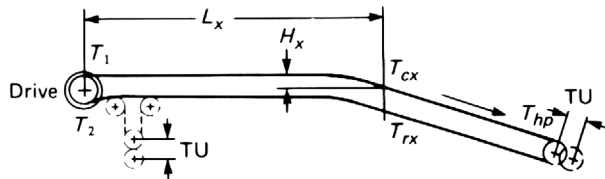


Figure 6.13C Conveyor with convex vertical curve, tail pulley drive. Lowering with regenerative load.



NOTE: Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.

Figure 6.14 Drive on return run — horizontal or elevating.*

$$T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or}$$

$$T_2 = T_o - 0.015 W_b L_s + W_b H_d$$

here

H_d = lift to the drive pulley

Use the larger value of T_2

$$T_t = T_{min} \text{ and } T_t = T_o$$

$$T_t = T_2 + 0.015 W_b L_s - W_b H_d$$

Use the larger value of T_t

$$T_{hp} = T_e + T_2 + \frac{L - L_s}{L} (T_b - T_{yr}) \text{ or } T_{hp} = T_e + T_o + T_b - T_{yr}$$

$$T_{hp} = T_{max}$$

$$T_{cx} = T_t + T_{fcx} + T_{wcx}$$

$$T_{rx} = T_t - T_{frx} + T_{wrx}$$

Takeups on return run or at tail pulley. *See page 119

Figure 6.14A Inclined conveyor with drive on return run.

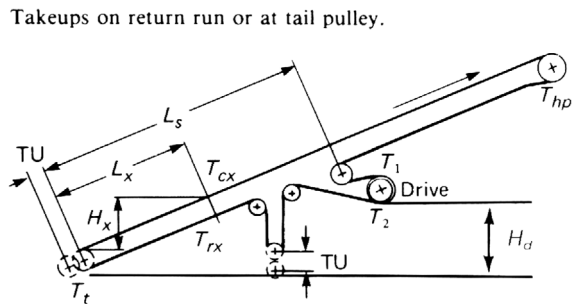


Figure 6.14B Horizontal belt conveyor with concave vertical curve, and drive on return run.

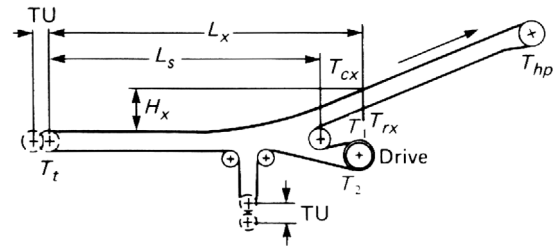


Figure 6.14C Horizontal belt conveyor with convex vertical curve, and drive on return run.

NOTE: Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.

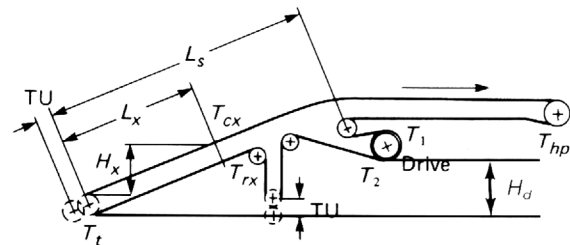


Figure 6.15 Drive on return run — lowering without regenerative load.*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o - T_e \text{ or } T_2 = T_o - 0.015 W_b(t_s) - W_b H_d$$

Use the larger value of T_2

$$T_1 = T_e + T_2$$

$$T_t = T_2 + 0.015 W_b L_s + W_b H_d$$

here

$$H_d = \text{lift to the drive pulley, or } T_t = T_o$$

Use the larger value of T_t

$$T_{max} = T_t \text{ or } T_1$$

$$T_{min} = T_t \text{ or } T_1$$

$$T_{hp} = T_t + T_{fcx} - T_{wcx}$$

here

$$L_x = L$$

$$T_{cx} = T_t + T_{fcx} - T_{wcx}$$

$$T_{rx} = T_t - T_{frx} - T_{wrx}$$

Takeup on return run or at tail pulley. *See page 119.

Figure 6.15A Declined conveyor, with drive on return run. Lowering without regenerative load.

Takeup on return run or at tail pulley

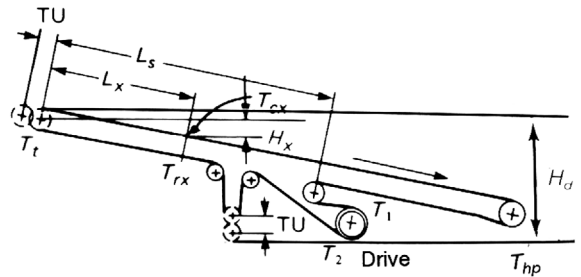


Figure 6.15B Conveyor with convex vertical curve, drive on return run. Lowering without regenerative load.

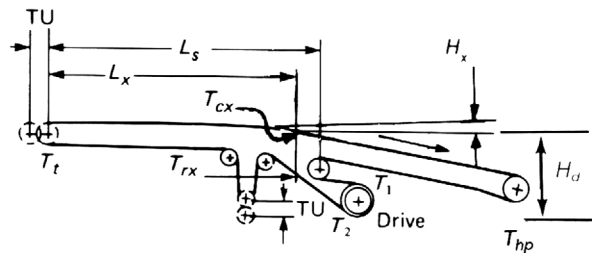
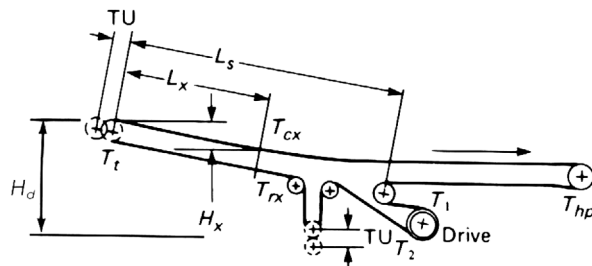


Figure 6.15C Conveyor with concave vertical curve, drive on return run. Lowering without regenerative load.



NOTE: Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.

Figure 6.16 Drive on return run — lowering with regenerative load.*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o + 0.015 W_b(L - L_s) + W_b(H - H_d)$$

Use the larger value of T_2

$$T_1 = T_e + T_2$$

$$T_t = T_{max} = T_1 + 0.015 W_b L_s + W_b H_d$$

here

H_d = lift to the drive pulley

$$T_{hp} = T_o$$

$$T_{hp} = T_2 - 0.015 W_b(L - L_s) - W_b(H - H_d)$$

Use the larger value of T_{hp}

$$T_{cx} = T_t - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_t - T_{wrx} - T_{frx}$$

*See page 119

Figure 6.16A Declined conveyor with drive on return run. Lowering with regenerative load.

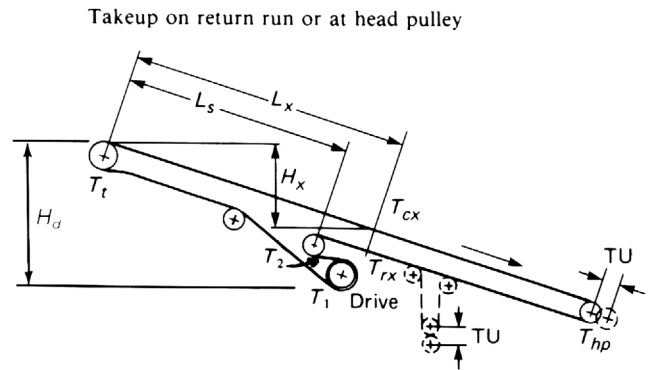


Figure 6.16B Conveyor with concave vertical curve, drive on return run. Lowering with regenerative load.

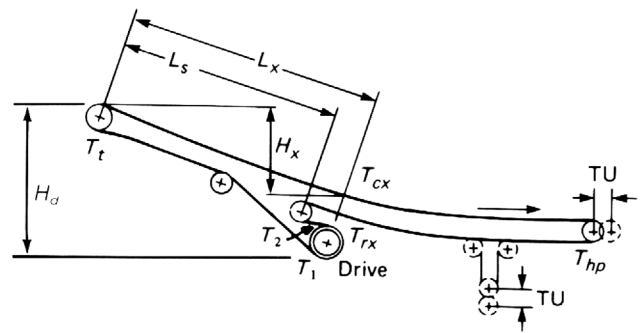
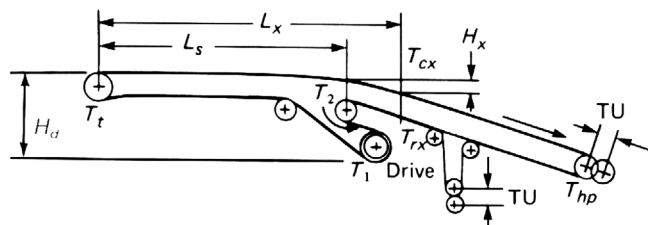


Figure 6.16C Conveyor with convex vertical curve, drive on return run. Lowering with regenerative load.



NOTE: Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.

Belt Tension Examples

A typical calculation of the various tensions in a conveyor belt with single-pulley drive is given below.

Example 1

Calculate the various belt tensions for a 30-inch belt conveyor per Figure 6.8A, with 300-ft centers, and a lift of 50-ft. Capacity is 500 tph, of material weighing 100 pounds per cubic ft, at a belt speed of 350 fpm. The belt is carried on 5-inch diam. Class C5 idlers (see Chapter 5) with $\frac{3}{4}$ -in. shafts, 35 degrees end roll angle. Idlers are spaced every $3\frac{1}{2}$ ft. The material contains 50 percent lumps. Belt weight is 15.0 lbs/ft, W_b . Material weight is 47.5 lbs/ft, W_m . Temperature is 60°F. T_e has been calculated and is equal to 3,030 lbs.

Step 1: Determine C_w . Assume lagged pulley, gravity takeup, and 180 degree wrap. Table 6-8 gives $C_w = 0.50$.

Step 2: Determine belt tension, T_2 . Minimum T_2 to drive = $T_e \times C_w (3,030)(0.50) = 1,515$ lbs. T_0 , the minimum allowable tension for a 2 percent belt sag, per page 120, is as follows:

$$T_0 = 6.25 S_i (W_b + W_m) = (6.25)(3.5)(15.0 + 47.5) = 1,367 \text{ lbs}$$

Using the formula for determining return belt friction tension, pages 87-88,

$$\begin{aligned} L &= 300 \\ W &= 15 \\ K_t &= 1.0, \text{ at } 60^\circ\text{F.} \\ T_{yr} &= 0.015 L W_b K_t = (0.015)(300)(15)(1) = 68 \text{ lbs} \\ T_2 \text{ (considering } T_0) &= T_0 + T_b - T_{yr} \\ T_b &= H W_b = (50)(15) = 750 \text{ lbs} \end{aligned}$$

$$\text{Therefore: } T_2 = 1,367 + 750 - 68 = 2,049 \text{ lbs}$$

Because this is larger than the 1,515 lbs minimum T_2 to drive, use $T_2 = 2,049$ lbs.

Step 3: Calculate the T_1 , T_{max} , and takeup tensions.

$$T_{max} = T_1 = T_e + T_2 = 3,030 + 2,049 = 5,079 \text{ lbs}$$

Takeup tension depends upon the location of the gravity takeup. If located near the head end, the belt tension at takeup is T_2 less the weight of nearly 3 ft of belt (45 lbs), or $2,049 - 45 = 2,004$ lbs. If located near the tail pulley, the takeup tension will be approximately the same as T_0 :

$$T_0 = T_2 + T_{yr} - T_b = 2,049 + 68 - 750 = 1,367 \text{ lbs}$$

(The non-driving pulley frictions have been omitted.)

A typical calculation of the belt tensions of a two-pulley (dual) drive is given below. The conveyor outline is per Figure 6.14A, but with a dual-pulley head drive, per Figure 6.7F.

Example 2

Conveyor length = 1,200 ft
Belt speed = 400 fpm
 T_e at the drive pulleys = 20,625 lbs
Required hp at the drive pulleys = 250 hp
Total motor horsepower = 275 hp
Primary drive motor = 200 hp
Secondary drive motor = 75 hp
Belt weight, W_b = 20 lbs/ft
 C_w = 0.11, according to Table 6-8 (380° wrap, lagged pulleys)

Step 1: Calculate T_{ep} and T_{es} :

$$T_{ep} = \left(\frac{200}{275}\right)(250)\left(\frac{33,000}{400}\right) = 15,000 \text{ lbs}$$

$$T_{es} = \left(\frac{75}{275}\right)(250)\left(\frac{33,000}{400}\right) = 5,625 \text{ lbs}$$

Step 2: Calculate T_2 , which is the minimum value that avoids slippage of the belt on the secondary pulley:

$$T_2 = T_e C_w = (20,625)(0.11) = 2,269 \text{ lbs}$$

Step 3: Calculate T_3 : $T_3 = T_2 + T_{es} = 2,269 + 5,625 = 7,894 \text{ lbs}$

Step 4: Calculate T_1 : $T_1 = T_3 + T_{ep} = 7,894 + 15,000 = 22,894 \text{ lbs}$

Step 5: Calculate C_{wp} and C_{ws} :

$$C_{wp} = \frac{T_3}{T_{ep}} = \frac{7,894}{15,000} = 0.53, \text{ requiring } 180^\circ \text{ wrap angle}$$

$$C_{ws} = \frac{T_3}{T_{es}} - 1 = \frac{7,894}{5,625} - 1 = 0.40, \text{ requiring } 205^\circ \text{ wrap angle}$$

Step 6: Check T_2 , using formula in Figure 6.14A. Assume conveyor is 1,200 ft long, the lift is 60 ft, $W_b = 20$, $W_m = 80$, idler spacing 3½ ft, drive at head of the conveyor.

$$T_0 = T_{min} = 6.25(W_b + W_m)S_i = 6.25(20 + 80)3.5 = 2,188 \text{ lbs}$$

(See “Minimum Belt Tension,” page 113.)

$$T_{yr} = 0.015LW_bK_t = (.015)(1,200)(20)(1) = 360 \text{ lbs}$$

(See “Return Belt Friction Tension,” pages 116-117; assume temperature above 32°, $K_t = 1.0$)

$$T_b = HW_b = (60)(20) = 1,200 \text{ lbs}$$

(See “Tension, T_b ” page 116)

Then $T_2 = T_{min} + T_b - T_{yr} = 2,188 + 1,200 - 360 = 3,028$ lbs

Therefore, as T_2 based on T_{min} is larger than T_2 minimum to prevent slippage (3,028 is greater than 2,269), use $T_2 = 3,028$ lbs.

Step 7: Calculate corrected values of T_3 , T_1 , C_{ws} , and C_{wp} :

$$T_3 = T_2 + T_{es} = 3,028 + 5,625 = 8,653 \text{ lbs}$$

$$T_{max} = T_1 = T_3 + T_{ep} = 8,653 + 15,000 = 23,653 \text{ lbs}$$

$$C_{ws} = \frac{T_3}{T_{es}} - 1 = \frac{8,653}{5,625} - 1 = 0.54$$

$$C_{wp} = \frac{T_3}{T_{ep}} = \frac{8,653}{15,000} = 0.58$$

Based on wrap factors from Step 5, which provide the minimum T_2 tension to drive without slippage, the secondary drive pulley would require a 205-degree angle of wrap, and the primary drive pulley would require a 180 degree angle of wrap.

The revised value of T_2 and the correspondingly revised values of C_{wp} and C_{ws} , per Step 7, indicate that both drives could have 180 degree angles of wrap. In order to have equal resistance to slip, both drives should have approximately the same wrap angle.

Belt Tension Calculations

Five illustrative examples are offered to make clear the use of the formulas in determining the belt tensions at point X on the belt conveyor.

Example 1

The basis for this example is the conveyor profile in Figure 6.8A from page 118, repeated below.

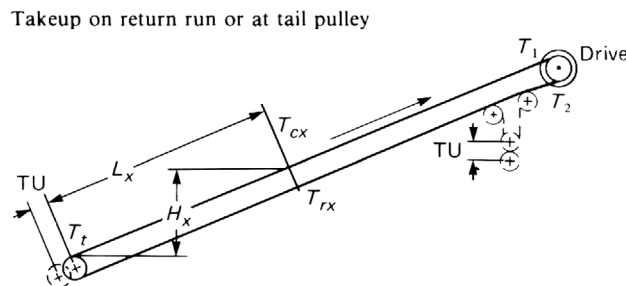


Figure 6.8A.

48-in. belt conveyor

W_b = weight of belt = 15 lbs/ft

W_m = weight of material = 106.6 lbs/ft

Troughing idlers, 20-degree angle, Class E6, 6-inch diameter, spaced at 3½ ft, factor $A_i = 2.8$

Return idlers, Class C6, 6-inch diameter, spaced at 10 ft

K_t = temperature correction factor = 1.0

T_t = $T_0 = 1,788$ lbs, as $T_0 = T_{min}$ here

L_x = 1,000 ft

H_x = 31.3 ft

To find the belt tension at point X on the carrying run:

T_{cx} = tension at point X on carrying run

T_{wcx} = tension resulting from weight of belt and material at point X

T_{fcx} = tension resulting from friction on carrying run at point X

$T_{cx} = T_t + T_{wcx} + T_{fcx}$

$T_{wcx} = H_x(W_b + W_m) = (31.3)(121.6) = 3,806$ lbs

$T_{fcx} = L_x K_t (K_x + K_y W_b) + L_x K_y W_m$ when $K_t = 1.0$, then

$T_{fcx} = L_x [K_x + K_y (W_b + W_m)]$

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}$$

$$= 0.00068(15 + 106.6) + \frac{2.8}{3.5}$$

$$= 0.883 \text{ (for value of } A_i, \text{ see tabulation on page 91)}$$

$K_y = 0.025$ when conveyor length is 1,000 ft (see Table 6-2). At 3.13% slope, $W_b + W_m = 121.6$ (use 125 in tables); and 3½ ft standard idler spacing.

$T_{fcx} = 1,000 [0.883 + 0.025(121.6)] = 3,923$ lbs

$T_{cx} = 1,788 + 3,806 + 3,923 = 9,517$ lbs

To find the belt tension at point X on the return run:

T_{rx} = tension at point X on the return run

T_{wrx} = tension at point X on the return run resulting from the weight of the belt

T_{frx} = tension at point X on the return run resulting from return run friction

$T_{rx} = T_t + T_{wrx} - T_{frx}$

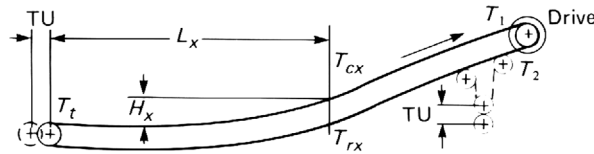
$T_{wrx} = H_x W_b = (31.3)(15) = 470$ lbs

$T_{frx} = L_x 0.015 W_b K_t = (1,000)(0.015)(15)(1) = 225$ lbs

$T_{rx} = 1,788 + 470 - 225 = 2,033$ lbs

Example 2

The basis for this example is the conveyor profile shown in Figure 6.8B from page 118, repeated below; otherwise, the data is the same as for Example 1, above.



$$\begin{aligned} L_x &= 1,915 \text{ ft} \\ H_x &= 31.3 \text{ ft} \\ T_t &= 1,788 \text{ lbs} \end{aligned}$$

Horizontal portion 1,565 ft long, inclined portion 835 ft long, on 9 percent slope.

To find the T_{cx} tension in the carrying run at point X:

$$\begin{aligned} T_{cx} &= T_t + T_{wcx} + T_{fcx} \\ T_{wcx} &= H_x(W_b + W_m) = (31.3)(121.6) = 3,806 \text{ lbs} \\ T_{fcx} &= L_x[K_x + K_y(W_b + W_m)] \text{ Since } K_t = 1.0 \\ T_{fcx} &\text{ is figured in two parts, first for the horizontal portion and then for the inclined portion} \end{aligned}$$

For the horizontal portion of the carrying run:

$$\begin{aligned} K_x &= 0.883 \\ K_y &= 0.0277 \text{ when conveyor length is 1,565 ft at 0 degree slope; } W_b + W_m = 121.6, \text{ (use 125 in tables); } 3\frac{1}{2} \text{ ft standard idler spacing} \\ T_{fcx}, \text{ horizontal} &= 1,565 [(0.883) + (0.0277)(121.6)] = 6,653 \text{ lbs} \end{aligned}$$

For the inclined portion of the carrying run:

$$\begin{aligned} K_x &= 0.883 \\ K_y &= 0.0217 \text{ when conveyor length is 1,915 ft at 1.63 percent average slope; } W_b + W_m = 121.6, \text{ use 125 in tables; } 3\frac{1}{2} \text{ ft standard idler spacing} \\ T_{fcx}, \text{ incline} &= 352 [(0.883) + (0.0217)(121.6)] = 1,240 \text{ lbs, where 352 ft is the distance along the inclined portion to point X. Total } T_{fcx}, \text{ horizontal plus incline} = 6,653 + 1,240 = 7,893 \text{ lbs} \\ T_{cx} &= 1,788 + 3,806 + 7,893 = 13,487 \text{ lbs} \end{aligned}$$

To find the T_{rx} tension in the return belt at point X:

$$\begin{aligned} T_{rx} &= T_t + T_{wrx} - T_{frx} \\ T_{wrx} &= (H_x)(W_b) = (31.3)(15) = 470 \text{ lbs} \\ T_{frx} &= L_x(0.015 W_b) K_t = 1,915(0.015)(15)(1.0) = 431 \text{ lbs} \\ T_{rx} &= 1,788 + 470 - 431 = 1,827 \text{ lbs} \end{aligned}$$

Example 3

The basis for this example is the conveyor profile shown in Figure 6.8C from page 118, repeated following. The data is the same as for Examples 1 and 2 above.

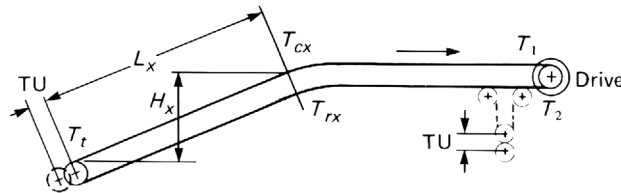


Figure 6.8C.

$$L_x = 350 \text{ ft, on 9\% slope}$$

$$H_x = 31.3 \text{ ft}$$

$$T_t = 1,788$$

To find T_{cx} at point X on carrying run:

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

$$T_{wcx} = H_x(W_b + W_m) = (31.3)(121.6) = 3,806 \text{ lbs}$$

$$T_{fcx} = L_x[K_x + K_y(W_b + W_m)]$$

$$K_x = 0.883$$

$$K_y = 0.0293 \text{ when conveyor length is 350 ft at 9\% slope; } W_b + W_m = 121.6 \text{ (use 125 in tables); } 3\frac{1}{2} \text{ ft standard idler spacing}$$

$$T_{fcx} = 350 [0.883 + (0.0293)(121.6)] = 1,556 \text{ lbs}$$

$$T_{cx} = 1,788 + 3,806 + 1,556 = 7,150 \text{ lbs}$$

To find T_{rx} at point X on return run:

$$T_{rx} = T_t + T_{wrx} - T_{frx}$$

$$T_{wrx} = H_x W_b = (31.3)(15) = 470 \text{ lbs}$$

$$T_{frx} = L_x(0.015 W_b) K_t = 350(0.015)(15)(1.0) = 79 \text{ lbs}$$

$$T_{rx} = 1,788 + 470 - 79 = 2,179 \text{ lbs}$$

Example 4

This example illustrates the problem of finding belt tensions when the drive is on the return run. The tension calculated at the head and tail pulleys is carried out as follows:

Takeups on return run or at tail pulley.

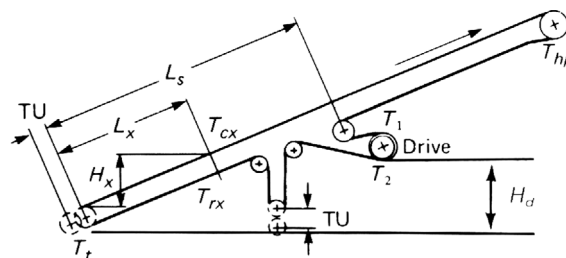


Figure 6.14A.

1. Calculate T_e , T_1 , T_2 , and T_t , the same as for a conveyor driven at a terminal pulley, using the horsepower formula on page 86, and appropriate tension formulas indicated with the conveyor profiles, Figure 6.14A. The T_e , T_1 , and T_2 tensions so calculated apply at the drive pulley regardless of its location along the return run.
2. Calculate the tension, T_{hp} , at the head pulley using the appropriate formula for T_{cx} as indicated for conveyor profile with drive on the return run, Figure 6.14. Calculate T_{wcx} and T_{fcx} from the formula for determining belt tension at any point. See page 117.

Conveyor data:

$$\begin{aligned}
 W_m &= 120 \text{ lbs/ft} \\
 W_b &= 15 \text{ lbs/ft} \\
 K_t &= 1.0 \\
 K_x &= 0.35 \\
 K_y &= 0.0243 \\
 C_w &= 0.35 \\
 S_i &= 3.5 \text{ ft, idler spacing}
 \end{aligned}$$

36-in. belt conveyor, 600-ft centers, drive located midway of return run, lift 54 ft, slope 9 percent

Calculate the head pulley tension, T_{hp} , and the tail pulley tension, T_t .

$$\begin{aligned}
 T_e &= LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) \\
 &= 600 [0.35 + (0.0243)(15) + (0.015)(15)] + 120[(600)(0.0243) + 54] \\
 &= 8,794 \text{ lbs}
 \end{aligned}$$

For 3 percent belt sag,

$$\begin{aligned}
 T_0 &= 4.2 S_i (W_b + W_m) = (4.2)(3.5)(135) = 1,985 \text{ lbs} \\
 T_2 \quad \text{minimum to drive} &= T_e C_w = (8,794)(0.35) = 3,078 \text{ lbs}
 \end{aligned}$$

$$\begin{aligned}
 \text{Corresponding } T_t &= T_2 + (L/2)(0.015 W_b) - (H/2) W_b \\
 &= 3,078 + (300)(0.015)(15) - (27)(15) = 2,741 \text{ lbs}
 \end{aligned}$$

This total tail pulley tension, 2,741 lbs, is greater than 1,985 lbs

Therefore:

$$\begin{aligned}
 T_t &= 2,741 \text{ lbs, and } T_2 = 3,078 \text{ lbs} \\
 T_1 &= T_e + T_2 = 8,794 + 3,078 = 11,872 \text{ lbs}
 \end{aligned}$$

The tension at any point on the carrying run is:

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

Now let

$$L_x = L$$

$$K_t = 1.0$$

Then, tension at the head pulley, $T_{hp} = T_t + T_{wcx} + T_{fcx}$

$$T_{wcx} = H_x(W_b + W_m) = (54)(135) = 7,290 \text{ lbs}$$

$$\begin{aligned} T_{fcx} &= L_x[K_t K_x + K_y W_b] + L_x K_y W_m, \text{ and since } L = L_x \\ &= 600 [0.35 + (0.0243)(15)] + (600)(0.0243)(120) \\ &= 2,178 \text{ lbs} \end{aligned}$$

$$\text{Therefore, } T_{hp} = 2,741 + 7,290 + 2,178 = 12,209 \text{ lbs}$$

This is the maximum belt tension. The T_I tension at the drive pulley may be checked as follows:

$$\begin{aligned} T_I &= T_{hp} - 27W_b + L/2(0.015W_b) = 12,209 - (27)(15) + (300)(0.015)(15) \\ &= 11,872 \text{ lbs} \end{aligned}$$

This checks with the 11,872 lbs calculated for T_I from the formula, $T_I = T_e + T_2$.

Example 5

This example calculates the belt tension at any point in a declined regenerative conveyor. The calculation is substantially the same as that for a non-regenerative conveyor except that $\frac{2}{3}K_y$ is used in place of K_y , and the factor A_i is eliminated in the formula for K_x . The value of K_y is for the length L_x . Figure 6.10A from page 120 is repeated below.

Takeup on return run not recommended to avoid driving through the takeup

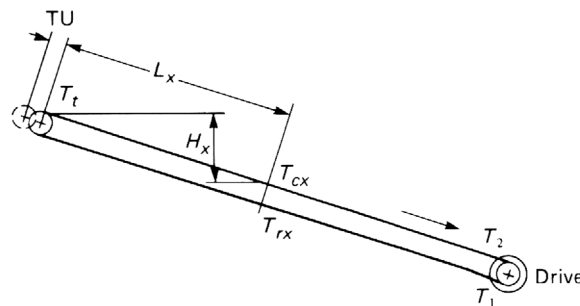


Figure 6.10A.

Conveyor data:

36-in. belt conveyor, 1,000-ft centers, head pulley drive, drop 90 ft, slope 9 per-cent

$$W_b = 15 \text{ lbs/ft}$$

$$\begin{aligned}
 W_m &= 120 \text{ lbs/ft} \\
 S_i &= 3.5 \text{ ft idler spacing} \\
 K_x &= 0.00068 (W_b + W_m) = 0.00068(135) = 0.0918 \\
 K_y &= (0.0169)(0.666) = 0.01126 \text{ for 1,000 ft, 135 for } (W_b + W_m) \text{ and 9 percent slope} \\
 K_t &= 1.0 \\
 C_w &= 0.35 \\
 T_{cx} &= T_t - T_{wcx} + T_{fcx} \\
 T_{rx} &= T_t - T_{wrx} - T_{frx} \\
 T_e &= LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) \\
 T_2 &= T_e C_w \text{ or } T_2 = T_0. \text{ If } T_0 \text{ is the greater} \\
 T_1 &= T_e + T_2 \\
 T_t &= T_1 + 0.015 W_b L + W_b H \\
 T_e &= 1,000 [0.0918 + (0.01126)(15) + (0.015)(15)] + (1,000)(0.01126)(120) - (90)(120) = 485.7 + 1,351.2 - 51.2 - 10,800 = -8,963 \text{ lbs}
 \end{aligned}$$

The minus sign merely means that the belt drives the pulley (the conveyor is regenerative).

$$\begin{aligned}
 T_2 &= (8,963)(0.35) = 3,137 \text{ lbs} \\
 T_0 \text{ for 3 percent sag} &= (4.2)(3.5)(135) = 1,985 \text{ lbs}
 \end{aligned}$$

Therefore, T_2 can be taken at 3,137 lbs

$$\begin{aligned}
 T_1 &= T_e + T_2 = 8,963 + 3,137 = 12,100 \text{ lbs} \\
 T_t &= 12,100 + (1,000)(0.015)(15) + (90)(15) = 13,675 \text{ lbs}
 \end{aligned}$$

To calculate T_{cx} at a point 500 ft from the tail shaft

$$\begin{aligned}
 T_{cx} &= T_t - T_{wcx} + T_{fcx} \\
 T_{wcx} &= H_x(W_b + W_m) = (45)(135) = 6,075 \text{ lbs} \\
 T_{fcx} &= L_x(K_x + K_y W_b) + L_x K_y W_m \text{ when } K_t = 1.0 \\
 &= 500 [0.0918 + (0.01126)(15)] + (500)(0.01126)(120) = 182 + 1,092 = 1,274 \text{ lbs when } K_y = \frac{2}{3} K_y \text{ for a 500-ft conveyor at 9 percent slope} \\
 \text{Then, } T_{cx} &= 13,675 - 6,075 + 1,274 = 8,874 \text{ lbs}
 \end{aligned}$$

To calculate T_{rx} at a point 500 ft from the tail shaft

$$\begin{aligned}
 T_{rx} &= T_t - T_{wrx} - T_{frx} \\
 T_{wrx} &= H_x W_b = (45)(15) = 675 \text{ lbs} \\
 T_{frx} &= L_x(0.015) W_b = (500)(0.015)(15) = 113 \text{ lbs when } K_t = 1.0 \\
 \text{Then } T_{rx} &= 13,675 - 675 - 113 = 12,887 \text{ lbs}
 \end{aligned}$$

For conveyor profiles per Figures 6.10B and 6.10C, the portion of the conveyor on a given slope is calculated separately, as in Example 2, page 131, and Problems 5 and 6, pages 172-176.

Acceleration and Deceleration Forces

Investigation of the acceleration and deceleration forces is necessary for the following reasons.

Belt Stress

Economy of design dictates the selection of a belt having a carcass strength at or near the normal operating tensions. Consequently, the additional forces resulting from acceleration or deceleration may overstress the belt or its splices, particularly if mechanical splices are used. While this problem is most likely to exist with respect to the belt, there also is the possibility of overstressing the mechanical components such as pulleys, shafts, bearings, takeups, etc.

Vertical Curves

Two different problems may be encountered with vertical curves.

In the case of concave curves (where the center of curvature lies above the belt) if belt tensions are too high during starting, the belt will lift off the troughing idlers. This is discussed in detail in Chapter 9. It is necessary to analyze this problem in regard to full, partial, and no loads.

In the case of convex vertical curves, where the center of the curvature lies below the belt, there is the possibility of overloading certain idlers.

Loss of Tension Ratio

During both acceleration and deceleration there exists the distinct possibility of losing the required T_1/T_2 ratio necessary to maintain the desired control of the engagement of belt and drive pulley. This particularly is true if the takeup is located far from the drive.

If a screw takeup is used and improperly adjusted or the travel of a gravity takeup is too limited, the necessary ratio T_1/T_2 may be lost during the attempt to accelerate the belt conveyor.

During deceleration, the effect of the inertia load may cause a loss of the T_1/T_2 ratio necessary to transmit braking forces from the braking pulley to the belt. This would permit the continued motion of the belt and load, after the pulley had been stopped.

Load Conditions on the Belt

The belt conveyor may operate satisfactorily during stopping or starting if completely loaded or if empty. This, however, may not be so if only portions of the conveyor length are loaded. The conveyor, therefore, has to be analyzed under various conditions of loading.

For example, when a belt conveyor contains a concave curve, a critical condition of starting may be the lifting of the belt at the curve during acceleration because the portion of the belt ahead of the concave vertical curve is loaded, while the remainder of the belt is not. This may not be true if the conveyor is regenerative. Such conditions require careful analysis.

Coasting

Where there is a system of belt conveyors transferring from one to another, sequence starting or stopping is almost always a prerequisite of design. As an example, a belt with very long centers may transfer to a belt of short centers, in which case the time required to decelerate the two belts must obviously be synchronized, despite the differences in the braking forces required. During the acceleration period, the same synchronization is necessary. In either case, the consequences of not making a

proper analysis and providing the necessary controls will result in a pile-up at the transfer point and possible destruction of the machinery and belt, plus an inoperative system.

Takeup Movement

During both the acceleration and deceleration cycles, where counterweighted takeups are used, the takeup travel may be insufficient unless these forces are considered. The engineer must consider not only the length of travel, but also the rate of travel, particularly where hydraulic, electric, or pneumatic controls are involved.

Effect on Material Carried

In certain instances, the rate of starting and stopping may exert influences on the material which result in intolerable conditions. Obviously, certain materials can be accelerated or decelerated more effectively by the belt than others. For example, if a declined belt conveyor handling pelletized iron ore is stopped too rapidly, the material may start to roll on the belt surface and result in a pile-up at the discharge point. Similarly, starting an inclined belt too rapidly may cause the material to roll backward.

Festooning

Without proper consideration of the starting and stopping forces, it is possible that belt tensions may drop to a point, at some spot in the line, where the belt will festoon (buckle). For example, a belt with a decline from the tail end, and an incline at the head, may be loaded at the tail end only. If braking is applied at the head pulley, the belt may have zero tension or even some slack on the carrying side. The obvious result is load spillage, entanglement, loss of alignment, etc.

Power Failure

In the event of power failure, the belt eventually will stop because of inherent friction forces. Depending upon the profile and conditions of loading, the time required for the friction forces to stop the belt may be intolerably long or short. In the case of a declined regenerative belt conveyor, it may completely unload itself. In a system of belt conveyors, a pile-up of material at transfer points is probable. Therefore, it is obvious that controlled stopping, in the event of a power failure, is very important.

Braking Tensions Taken by Return Run and Tail Pulley

When deceleration is accomplished by means of a brake, the belt tension resulting from the braking force is taken in a direction opposite to that for driving the belt.

For instance, if the drive is at the head end of a horizontal or lift conveyor, power is transmitted from the drive pulley to the carrying side of the belt when the motor is energized. When decelerating with a brake connected to the drive pulley and the motor de-energized, the braking force may be transmitted from the drive pulley to the return belt. The brake application, therefore, may be significant in determining the amount of the counterweight, the design of the takeup, and the shaft sizes.

These are some of the problems that result when acceleration and deceleration forces are ignored or are improperly evaluated. While other difficulties may also exist, those discussed above are sufficient to indicate the importance of proper consideration and analysis.

Analysis of Acceleration and Deceleration Forces

The accelerating and decelerating forces that act on a belt conveyor during the starting and stopping intervals are the same in either case. However, their magnitude and the algebraic signs governing these forces change, as do the means for dealing with them.

Acceleration

The acceleration of a belt conveyor is accomplished by some form of prime mover, usually by an electric motor. The resulting forces in a horizontal conveyor are determined by inertia plus friction; in an inclined conveyor, by inertia plus friction plus elevating of the load; in a declined conveyor, by inertia plus friction minus lowering of the load.

Deceleration

The deceleration of a belt conveyor is accomplished by some form of brake. The resulting forces in a horizontal conveyor are determined by inertia minus friction; in an inclined conveyor, by inertia minus friction minus elevating of the load; in a declined conveyor, by inertia minus friction plus lowering of the load.

If the conveyor contains several portions with different (positive or negative) slopes, a combination of these conditions may result.

Calculation of Acceleration and Deceleration Forces

The belt conveyor designer is then confronted with the necessity to compute for the conveyor in question the inertia of all its moving parts, the inertia of the load on the belt, total frictional forces, and forces caused by elevating or lowering the load and belt. To be useful, the first two quantities have to be converted to a pound force at the belt line.

Inasmuch as acceleration is defined as the second derivative of displacement with respect to time, and deceleration is simply negative acceleration, time is the basic variable in computing the force. To compute the time, Newton's second law is used. The basic approach is as follows:

$$F_a = Ma$$

where :

$$F_a = \text{accelerating or decelerating force, lbs}$$

$$M = \text{mass, in slugs} = \frac{W_e}{g}$$

$$W_e = \text{equivalent weight of moving parts of the conveyor and load, lbs}$$

$$g = \text{acceleration by gravity} = 32.2 \text{ ft/sec}^2$$

$$a = \text{acceleration, ft per sec per sec (ft/sec}^2\text{)}$$

The force necessary to achieve the acceleration or deceleration is always directly proportional to the mass (or the weight) of the parts and material in motion.

For purposes of calculation, it can be assumed that the belt and the load on it move in a straight line. Other important parts of the system, however, rotate. This is true for all pulleys (including those on takeups and belt trippers), all idlers, and all the rotating parts of the drive.

It appears convenient to use the equation for linear motion as the basis for calculating the acceleration and deceleration forces. This makes it necessary to convert the physical properties of the rotating components of the system to a form in which they can be used in the basic linear relationship:

$$F = \left(\frac{W_e}{g} \right) a$$

In other words, one must find the “Equivalent Weight” of the rotating parts.

For rotating bodies, the mass actually distributed around the center of rotation is equivalent in its effect to the whole mass concentrated at a distance, K , (the polar radius of gyration, in ft) from that center.

The WK^2 is the weight of the body multiplied by the square of the radius of gyration. If WK^2 is known for the rotating conveyor components, the Equivalent Weight of these components, at the belt line, can be found by solving the equation

$$\text{Equivalent Weight lbs} = WK^2 \left(\frac{2\pi \text{ rpm}}{V} \right)^2$$

where: V = belt velocity, fpm

Values of WK^2 (expressed in lb-ft²), which are difficult to compute, except for very simple shapes, must be obtained for each component from the manufacturers of the conveyor components, motors, transmission elements, etc.

So far considered have been the forces in the system caused by inertia of the moving parts of the conveyor, the moving parts of the drive, and the moving load. Two other forces also are involved, as mentioned on page 138. These are: (1) Forces resulting from friction. (2) Forces resulting from the elevating or lowering of the load and belt. These simply represent the components of the weight of the material and belt, in the direction of motion of the belt, in the various portions of the conveyor.

Design Considerations

The belt conveyor designer is confronted with two problems: (1) The necessity to provide a prime mover powerful enough to start the conveyor, sometimes under adverse conditions. (2) To make sure, for the reasons outlined under “Acceleration and Deceleration Forces,” page 138, that the maximum force exerted on the conveyor is within safe limits.

In long, level, high-speed conveyors, a motor large enough for continuous full-load operation may be unable to start the fully loaded conveyor, particularly in cold weather. On the other hand, a motor capable of continuous full-load operation of an inclined conveyor may overstress the belt during starting, unless preventive measures are taken.

The maximum permissible accelerating forces are determined by the factors listed in page 138 of this chapter. Minimum accelerating forces may be dictated by the time during which the prime mover, which usually is an electric motor, can exert its starting torque without being damaged. This limitation is also affected by the frequency of starting the conveyor system.

In the case of deceleration, maximums are governed by the same factors. A minimum deceleration may be dictated by safety or may be necessary because of the material flow at transfer points. In all deceleration calculations involving brakes, the energy-dissipating capacity of the brake will be an important factor to consider.

Necessary Assumptions

As in all engineering investigations of this type, the first question is, “To what degree of accuracy will the computations have to be carried out?” The answer is not simple. Important factors are the overall size, the importance of the installation, and the type and sensitivity of the equipment adjacent to it.

In any case, numerous simplifying assumptions will have to be made to keep the engineering work within reasonable limits. For examples of simplifying assumptions, refer to the problems connected with belt stretch (elastic elongation from accelerating or decelerating forces) and takeup reactions.

During both the acceleration and deceleration cycles, the transient forces imposed result in extra stretch not encountered during steady state operation. This may result in early splice failure, excessive takeup travel, and other difficulties. Because of the vast differences in carcass construction, from the standpoint of both materials used and methods of manufacture, no single numerical value can express belt stretch as a function of the applied force.

Most manufacturers have listed values of B_m (elastic constant) for their line of belts. These vary from 1.3×10^6 lbs per in. of belt width for steel-cable belt to 2.3×10^3 lbs per ply in. of belt width for cotton fabric belts. Other rubber manufacturers may list different values, but they also would vary over the same wide range.

For this reason, as well as many others, the calculations for acceleration and deceleration treat the system as a rigid body. This is a common practice in the solution of problems in dynamics. And while the results usually are quite satisfactory, there is more cause for concern over the accuracy of results in the case of belt conveyors.

No further attempt will be made to justify simplifying the assumption, because this usually is not of major significance. However, the belt conveyor designer should be aware that, for conveyor systems with very long center belts, stretch considerations should not be overlooked.

Calculations

While the calculations are relatively simple for a conveyor with only one slope, they become increasingly complex for belt conveyors which change slope several times, or which are loaded and unloaded at different points, or which have belt trippers operating on them.

All this results in a great number of possible combinations of load distribution, tripper position, etc. Although theoretically it suffices to investigate only the worst combination of conditions, without analysis, it is usually impossible for even the experienced designer to tell which combination of factors will lead to this extreme case.

In the more complicated cases, it will be necessary to divide the conveyor into portions or sections—within which neither the slope nor the conditions of loading change more than is permitted by the required accuracy of the calculations—and to determine the physical properties for each such portion discussed under “Analysis of Acceleration and Deceleration Forces,” page 138. Any really large rotating member of the belt conveyor, because of its very magnitude, may have to be considered a portion or section by itself.

A summation of these weights, forces, and stresses with proper consideration of their algebraic signs will indicate that portion of the system which will impose the most severe limitations to the allowable values for acceleration and deceleration. This, in turn, will permit the selection of the proper prime mover and the necessary control elements.

Conveyor Horsepower Determination — Graphical Method

The graphical method shown in Figures 6.17, 6.18, and 6.19 provides the means for estimating horsepower (hp). Belt tensions can be calculated from the resulting horsepower. This method is suitable for conveyors of moderate capacity having relatively straight paths of travel. The results will be sufficiently accurate to establish horsepower requirements when actual weights of belt and revolving parts per foot of conveyor centers are used in Figure 6.17. However, for use in determining tentative or approximate horsepower, a convenient table of typical weights is superimposed on Figure 6.17.

The graphical method is not suitable for final calculations of horsepower for conveyors having decline portions, high capacity, or complex arrangements of terminals, nor for the extended use of rubber skirting and plows that substantially increase the frictional drag on the conveyor belt. On the other hand, it is useful for tentative estimates of horsepower under most of these conditions.

An example of the use of the graphical method follows.

Determining Required Horsepower — Graphical Method

The following example illustrates a method for determining the required horsepower (hp) for a belt conveyor. The example is the same as Problem 1, page 145, calculated by the analytical method.

In this graphical solution, only the horsepower requirements to move the belt horizontally, elevate the material, and convey the material horizontally are considered. Additional accessory factors such as pulley friction, skirtboard friction, material acceleration, and auxiliary device frictions are included as averages.

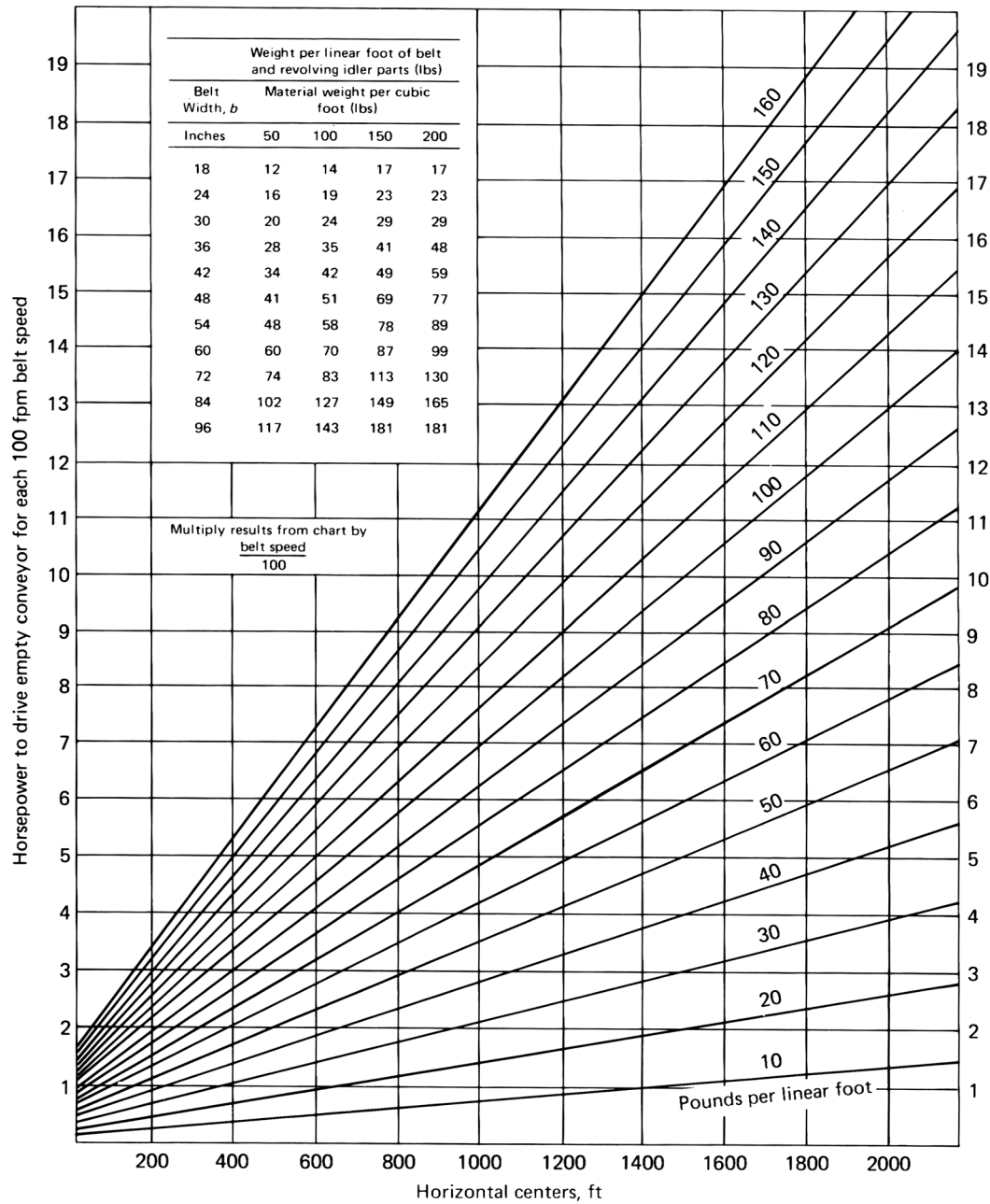


Figure 6.17 Horsepower required to drive empty conveyor.*

NOTE: *The table of weights is representative of average weights of revolving idler parts, as given in Chapter 5, and estimated belt weights, listed in Table 6-1, page 90. Where actual weights are known, these should be used in the graphical solution.

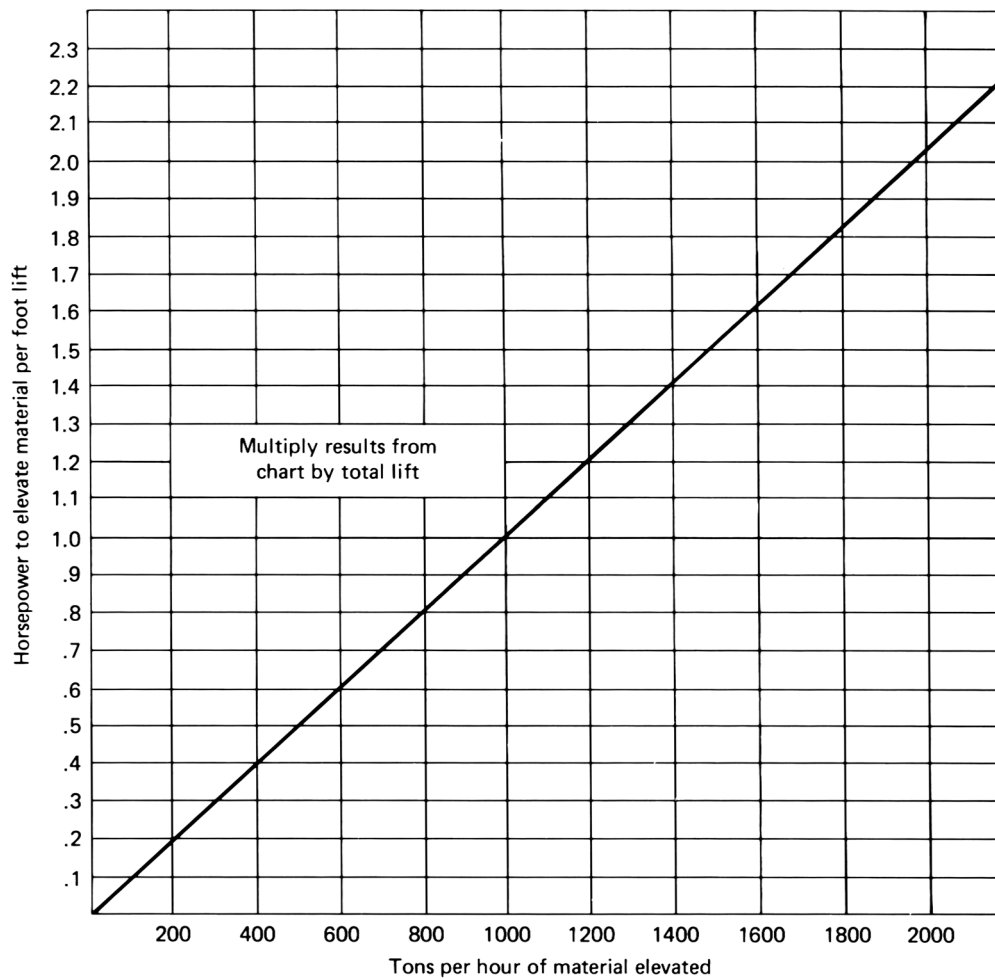


Figure 6.18 Horsepower required to elevate material.

Conveyor specifications:

Length, $L = 2,000$ ft

Lift, $H = 75$ ft

Capacity, $Q = 1,600$ tph

Belt speed, $V = 500$ fpm

Material density, $d_m = 100$ lbs/cu ft

Belt width, $b = 48$ in.

Graphical analysis. Referring to Figure 6.17, the weight per foot of belt and revolving idler parts for a 48-in. wide conveyor and 100 pounds per cubic foot of material is given as 51 pounds per foot. Using this value, the horsepower required to drive the empty conveyor at a speed of 100 fpm is 6.5.

Therefore, the horsepower to drive the empty conveyor at 500 fpm is equal to:

$$\frac{6.5 \times 500}{100} = 32.5 \text{ horsepower}$$

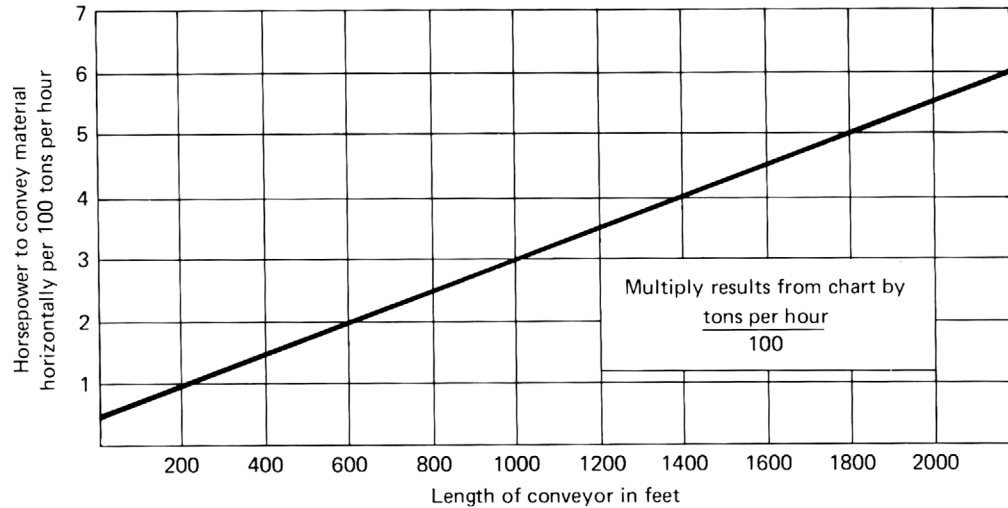


Figure 6.19 Horsepower required to convey material horizontally.

From using Figure 6.18, the horsepower required to elevate the material can be determined. The horsepower per foot of lift for the 1,600 tph capacity is 1.62. Therefore, the horsepower required to elevate the material 75 ft is given as:

$$1.62 \times 75 = 121.5 \text{ hp}$$

The horsepower needed to convey the material horizontally is determined through the use of Figure 6.19. Using the given conveyor specification of 2,000 ft of conveyor length, the horsepower required to convey 100 tph of material is equal to 5.5 hp. Therefore, the horsepower required for the 1,600 tph capacity is equal to:

$$\frac{5.5 \times 1,600}{100} = 88 \text{ horsepower}$$

The total required horsepower at the belt line is the sum of the above, and equals $32.5 + 121.5 + 88 = 242 \text{ hp}$.

Assuming a standard 5 percent loss of power through the drive components due to their inefficiencies, the required motor horsepower is:

$$\frac{242}{.95} = 254.7 \text{ horsepower}$$

A comparison of the horsepower derived by the analytical method, shown on pages 145 to 148 and the above illustrated graphical method shows that the results of the two methods are comparatively very close. This is coincidental inasmuch as the

degree of accuracy of determinations made with the graphical solution is dependable only for estimating purposes. Final design should be made by the analytical method for greatest accuracy.

Examples of Belt Tension and Horsepower Calculations — Six Problems

Application of the CEMA horsepower (hp) formula and analysis of belt tensions and power requirements will be illustrated by the following six problems:

Problem 1 — inclined conveyor;

Problem 2 — declined conveyor with regenerative characteristics;

Problem 3 — horizontal conveyor;

Problem 4 — conveyor with a horizontal section, an inclined section, and vertical curves;

Problems 5, 6 — comparison of tension and hp values on two similar conveyors.

Problems 3 and 4 also include calculation of acceleration and deceleration forces.

Problem 1

Inclined Belt Conveyor

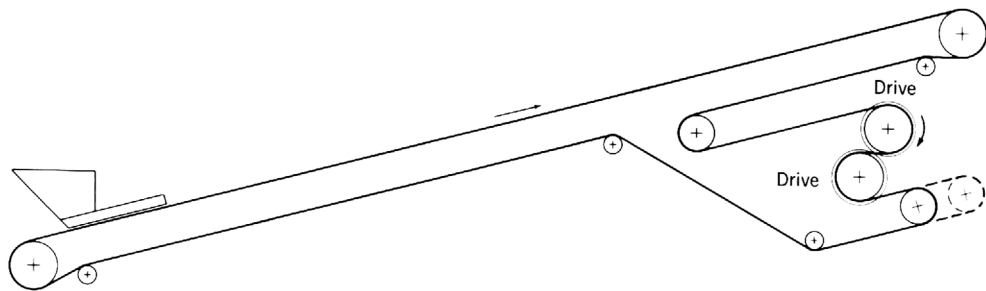


Figure 6.20 Inclined belt conveyor.

Problem:

Determine effective tension, T_e ; slack-side tension, T_2 ; maximum tension, T_1 ; tail tension, T_t ; belt and motor horsepower requirements; and type and location of drive.

In this problem, only two accessories are considered, pulley friction from non-driving pulleys and skirtboard friction. The belt speed is too low to involve any appreciable material acceleration force. The discharge is made freely over the head pulley and no cleaning devices are used.

Conveyor Specifications:

$W_b = 15$ lbs/ft from Table 6-1

$L = \text{length} = 2,000$ ft

$V = \text{speed} = 500$ fpm

$H = \text{lift} = 75$ ft

$Q = \text{capacity} = 1,600$ tph

$S_i = \text{spacing} = 3.5$ ft

Ambient temperature = 60°F

Belt width = 48 in

Material = phosphate rock at 80 lb/ft³, 15-in. maximum lump from a gyratory crusher

Drive = lagged head pulley or dual drive. Wrap is 240° or 380°, depending on which drive is to be used. See Figures 6.6B and 6.7A; also, Example 2, page 131, and comments.

Troughing idlers = Class E6, 6-in. diameter, 20° angle

Return idlers = Class C6, 6-in. diameter, 10-ft spacing

Analysis:

Using Table 6-8, drive factor $C_w = 0.30$ or 0.11, depending on use of lagged head pulley or dual drive.

$$W_m = \frac{33.3Q}{V} = \frac{33.3 \times 1,600}{500} = 106.6 \text{ lbs per ft}$$

From Figure 6.1, for 60°F, $K_t = 1.0$

Formula:

$$T_e = LK_t(K_x + K_yW_b + 0.015W_b) + W_m(LK_y + H) + T_{ac}$$

To find K_x and K_y , it is necessary to find

$$W_b + W_m = 15 + 106.6 = 121.6 \text{ lbs/ft}$$

Since 3½-ft idler spacing is given and the K_x value is calculated by using the formula:

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}$$

$$K_x = 0.00068(121.6) + 3.5 = 0.0826 + 0.800 = 0.8826$$

Examples of Belt Tension and Horsepower Calculations — Six Problems

K_y for $L = 2,000$ ft, the slope is $(75/2,000)(100\%) = 3.75\%$, and $W_b + W_m = 121.6$ lbs/ft. Table 6-2 gives $K_y = 0.018$.

Minimum tension for 3 percent sag, $T_0 = 4.2 S_i (W_b + W_m)$

$$T_0 = (4.2)(3.5)(121.6) = 1,788 \text{ lbs}$$

Determine Accessories:

In this case, the only accessories are nondriving pulley friction and skirtboards. Assume that skirtboards are 15 ft long and spaced apart two-thirds the width of the belt. Then the pull on the belt to overcome skirtboard friction is $T = C_s L_b h_s^2$. From the calculation of skirtboard friction, $h_s = (0.1)(48) = 4.8$ in. C_s is 0.1086, from Table 6-7, for phosphate rock at 80 pounds per cubic foot. Thus, to solve the equation:

$$T = C_s L_b h_s^2 = (0.1086)(15)(4.8)^2 = 38 \text{ lbs}$$

For the 30 ft of rubber edging on the skirtboards, the additional resistance is $(3)(30) = 90$ lbs. The total skirtboard resistance is $38 + 90 = 128$ lbs.

$LK_t K_x$	$= (2,000)(1)(0.8826)$	$= 1,765$
$LK_t K_y W_b$	$= (2,000)(1)(0.018)(15)$	$= 540$
$LK_t 0.015 W_b$	$= (2,000)(1)(0.015)(15)$	$= 450$
$K_y L W_m$	$= (0.018)(2,000)(106.6)$	$= 3,838$
$H W_m$	$= (75)(106.6)$	$= 7,995$
Nondriving pulley friction	$= (2)(200) + (2)(150) + (4)(100)$	$= 1,100$
Skirtboard resistance, T_{sb}		$= 128$
Effective tension, T_e		$= 15,816 \text{ lbs}$

Determine type of drive:

Analyze head pulley drive @ 240° wrap, $C_w = 0.30$

$$T_2 = C_w T_e = (0.30)(15,816) = 4,745 \text{ lbs}$$

Analyze dual drive @ 380° wrap, $C_w = 0.11$

$$T_2 = C_w T_e = (0.11)(15,816) = 1,740 \text{ lbs}$$

However, the minimum tension, $T_0 = 1,788$ lbs. This minimum should exist close to the loading point on the carrying run of the belt, or at T_t , to avoid more than 3 percent sag between the troughing idlers spaced at 3.5 ft intervals.

If $T_t = 1,788$ lbs, the weight of the return belt is $H W_b = 75 \times 15 = 1,125$ lbs, and the resistance of the return belt is $0.015 L W_b$, then $T_2 = 1,788 + 1,125 - 450 = 2,463$ lbs.

Using $T_2 = 2,463$ lbs, the saving in belt tension with the dual drive over a single-head pulley drive is $4,745 - 2,463 = 2,282$ lbs, or $2,282/48 = 48$ lbs/in. width of belt. This saving in belt cost may be enough to offset the cost of a drive.

$$\begin{aligned}
 T_2, \text{ by choice} &= 2,463 \\
 \text{Return run friction } (2,000)(0.015)(15) &= \underline{+ 450} \\
 &2,913 \\
 \text{Less weight of return belt } (75)(15) &= \underline{-1,125} \text{ lbs} \\
 \text{Tail tension, } T_t &= 1,788 \text{ lbs}
 \end{aligned}$$

Final Tensions:

$$T_e = 15,816 \text{ lbs}$$

$$T_2 = 2,463 \text{ lbs}$$

$$T_t = T_e + T_2 = 15,816 + 2,463 = 18,279 \text{ lbs}$$

$$T_t = 1,788 \text{ lbs}$$

Horsepower at Motor Shafts:

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(15,816)(500)}{33,000} = 239.64$$

$$\text{Drive pulley friction loss hp} = \frac{(2)(200)(500)}{33,000} = 6.06$$

$$\text{Add 5 percent for speed reduction loss} = 0.05 (239.64 + 6.06) = \underline{12.29}$$

$$\text{Total hp at motor shafts} = 257.99 \text{ hp}$$

$$\text{Belt tension} = \frac{18,279}{48} = 381 \text{ lbs per inch of belt width}$$

Problem 2

Declined Belt Conveyor

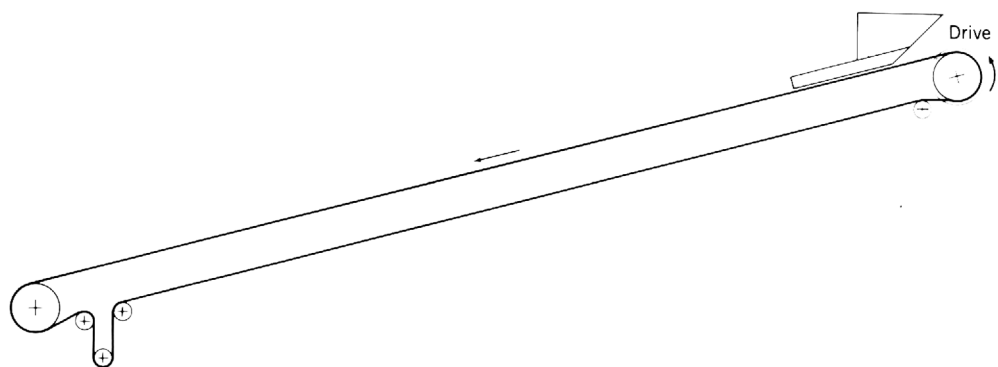


Figure 6.21 Declined belt conveyor.

Before attempting the solution of a declined conveyor, certain peculiar conditions must be considered.

A declined conveyor, which delivers material below the elevation at which it is received, will generate power if the net change in elevation is more than 2½ percent of the conveyor length. It may generate power at a lower slope, depending on conditions. An electric motor, acting as a generator, is used to retard the conveyor. A brake is used to stop the conveyor.

The motor size is determined by the maximum horsepower, either positive or negative, that it will be called on to produce, and it usually is set by the horsepower generated. The drive is usually located at the tail (feed) end of the conveyor, involving special design problems. One of these is that the motor must start the conveyor by driving through the gravity takeup without lifting the takeup pulley. Care must be taken to check the horsepower and belt tensions for an empty and partially loaded belt.

The brake must be large enough to absorb the torque generated and to decelerate the load. However, the retarding torque must be limited so that it does not overstress the belt. Frequently, on large conveyors, the limiting factor in brake selection will be its holding power within its ability to absorb and dissipate heat. Refer to “Break Heat Absorption Capacity,” pages 191-192.

When a conveyor runs downhill, friction forces increase the belt tension in the direction of motion, while gravity forces decrease the belt tension, by the weight per foot of belt and load, for every foot that the belt and load are lowered.

Reduced friction.

The belt, load, and idler friction absorb some of the power that the motor or brake would be compelled to absorb if these quantities did not exist. Therefore, it is important not to overestimate the friction forces or else the selected size of motor or brake might be too small. In order to avoid overestimating the friction forces, the effective tension, T_e is calculated as follows:

$$T_e = LK_t(K_x + C_1K_yW_b + C_10.015W_b) + C_1W_mLK_y - HW_m + C_1T_{ac}$$

in which factor C will vary from 0.5 to 0.7 and, for average conditions, will be 0.66.

For *declined conveyors only*, determine K_x by the formula $K_x = 0.00068 (W_b + W_m)$. The additive term A_i/S_i is omitted because the allowance for grease and seal friction, represented by the factor A_i , is no longer on the safe side. It may under some conditions approach zero, so the safe course in declined conveyors is to make $A_i = 0$.

Problem:

Determine effective tension (T_e), slack-side tension (T_2), maximum tension (T_1), tail tension (T_t), and belt and motor horsepower requirements.

In this problem, only two accessories are considered, pulley friction of nondriving pulleys and skirtboard friction. The belt speed is too low to involve any appreciable material acceleration and no cleaning devices are employed.

Conveyor Specifications:

$W_b = 10$ lbs/ft from Table 6-1

$L =$ length = 1,200 ft

$V =$ speed = 450 fpm

$H =$ drop = 200 ft

$Q =$ capacity = 1,000 tph

$S_i =$ spacing = 4 ft

Ambient temperature = 32°F, minimum

Belt width = 36 in.

Material = limestone at 85 lbs/cu ft, 4-in. maximum lumps

Drive = lagged grooved tail pulley, wrap = 220 degrees

Troughing idlers = Class C6, 6-in. diameter, 20-degree angle, $A_i = 1.5$

Return idlers = Class C6, 6-in. diameter, 10 ft spacing

Analysis:

From Table 6-8, drive factor, $C_w = 0.35$

From Figure 6.1, for 32°F, $K_t = 1.0$

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(1,000)}{450} = 74 \text{ lbs per foot}$$

Formula (not considering C_I factor):

$$T_e = LK_t(K_x + K_yW_b + 0.015W_b) + W_m(LK_y - H) + T_{ac}$$

To find K_x and K_y it is necessary to find

$$W_b + W_m = 10 + 74 = 84 \text{ lbs/ft}$$

K_x must be calculated for two cases. In the first calculation, K_x is taken at its normal value, so that the tension for the full friction can be determined. In the second calculation, K_x is taken at its reduced value, so that the tension for the reduced friction can be calculated.

$$\text{Normal } K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}$$

$$K_x = (0.00068)(84) + \frac{1.5}{4} = 0.05712 + 0.375 = 0.4321$$

$$\text{Reduced } K_x = 0.00068(84) = 0.05712$$

K_y also must be determined for two cases. In the first instance, K_y will have its normal value as selected from Tables 6-2 and 6-3. This value is then employed for the full friction calculation of the tension. In the second instance, K_y is modified by the reduced friction factor C .

The slope is $(200/1,200)(100\%) = 16.6$ percent. From Table 6-2, it can be seen that 4 ft is not a tabular spacing. For the 16.6 percent slope, $L = 1,200$ and $W_b +$

Examples of Belt Tension and Horsepower Calculations — Six Problems

$W_m = 84$, the correct value of K_y is 0.01743, by double interpolation. For simplification in calculations, use K_y value of .018.

The minimum tension for 3 percent sag is:

$$T_0 = 4.2S_i(W_b + W_m) = (4.2)(4)(84) = 1,411 \text{ lbs}$$

Determine Accessories:

The accessories are nondriving pulley friction and skirtboard friction. Assume the skirtboards to be 10 ft long and spaced apart, two-thirds the width of the belt. Then the pull to overcome skirtboard friction is:

$$T = C_s L_b h_s^2$$

$h_s =$ 10 percent of belt width or $(0.1)(36) = 3.6$ in. From Table 6-7, C_s for limestone is 0.128. Thus, to solve the equation:

$$T = (0.128)(10)(3.6)^2 = 17 \text{ lbs.}$$

For the 20 ft of rubber skirtboard edging, the additional resistance is $(3)(20) = 60$ lbs. Total skirtboard resistance, $T_{sb} = 17 + 60 = 77$ lbs

Full friction, T_e :

$LK_t K_x$	$=$	$(1,200)(1)(0.4321)$	$=$	518.5
$LK_t K_y W_b$	$=$	$(1,200)(1)(0.018)(10)$	$=$	216.0
$LK_t 0.015 W_b$	$=$	$(1,200)(1)(0.015)(10)$	$=$	180.0
$K_y L W_m$	$=$	$0.018(1,200)(74)$	$=$	1,598.4
$-H W_m$	$=$	$-(200)(74)$	$=$	-14,800.0
Nondriving pulley friction $= (2)(150) + (3)(100)$				$=$ 600.0
Skirtboard resistance, T_{sb}				$=$ 77.0
Full friction, T_e				$=$ -11,610.1 lbs

Reduced friction, T_e :

$$T_e = LK_t(K_x + C_1 K_y W_b + C_1 0.015 W_b) + C_1 K_y L W_m - H W_m + C_1 T_{ac}$$

and

$LK_t K_x$	$=$	$(1,200)(1)(0.05712)$	$=$	68.5
$LK_t C_1 K_y W_b$	$=$	$(1,200)(1)(0.66)(0.018)(10)$	$=$	142.6
$LK_t 0.015 C_1 W_b$	$=$	$(1,200)(1)(0.015)(0.66)(10)$	$=$	118.8
$C_1 K_y L W_m$	$=$	$(0.66)(0.018)(1,200)(74)$	$=$	1,054.9
$-H W_m$	$=$	$-(200)(74)$	$=$	-14,800.0
Nondriving pulley friction $= [(2)(150) + (3)(100)]0.66$				$=$ 396.0
Skirtboard resistance $= (77)(0.66)$				$=$ 50.8
Reduced friction, T_e				$=$ - 12,968.4 lbs

Full friction, T_2 :

$$T_2 = C_w T_e = (0.35)(11,610.1) = 4,064 \text{ lbs}$$

Reduced friction, T_2 :

$$T_2 = C_w T_e = (0.35)(12,968.4) = 4,539 \text{ lbs}$$

Full friction, T_t :

$$T_t = T_2 - 0.015LW_b - HW_b = 4,064 - 180 - 2,000 = 1,884 \text{ lbs}$$

Reduced friction, T_t :

$$T_t = T_2 - 0.015C_1LW_b - HW_b = 4,539 - 120 - 2,000 = 2,419 \text{ lbs}$$

Full friction, T_1 :

$$T_1 = T_e + T_2 = 11,610 + 4,064 = 15,674 \text{ lbs}$$

Reduced friction, T_1 :

$$T_1 = T_e + T_2 = 12,968 + 4,539 = 17,507 \text{ lbs}$$

Table 6-11. Final tensions, full and reduced friction.

Final Tensions	Full Friction, lbs	Reduced Friction, lbs
T_e	11,610	12,968
T_2	4,064	4,539
T_1	15,674	17,507
T_t	1,884	2,419

Horsepower at Motor Shaft:

The horsepower at the motor shaft should be based on the higher of the two values for T_e .

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(-12,968)(450)}{33,000} = -176.84$$

$$\text{Drive pulley friction hp} = \frac{(200)(450)}{33,000} = +2.73$$

$$\text{Additional 5 percent for speed reduction losses} = +8.71$$

$$\text{horsepower at motor shaft} = -165.4 \text{ hp}$$

$$\text{Belt tension} = \frac{17,507}{36} = 486 \text{ lbs/in of belt width.}$$

Problem 3

Horizontal Belt Conveyor

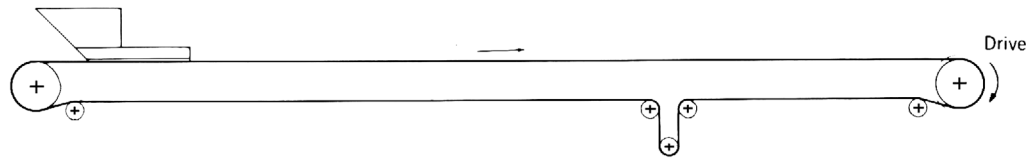


Figure 6.22 Horizontal belt conveyor.

Problem:

Determine effective tension (T_e), slack-side tension (T_2), maximum tension (T_1), tail tension (T_t), belt and motor horsepower requirements.

In this problem, only two accessories are considered, pulley friction from non-driving pulleys and skirtboard friction. Material acceleration force has been omitted in this example. The discharge is made freely over the head pulley. No belt-cleaning devices are employed.

Conveyor Specifications:

- W_b = 17 lbs/ft, from Table 6-1
- L = length = 2,400 ft
- V = speed = 500 fpm
- H = lift = 0
- Q = capacity = 3,400 tph
- S_i = spacing = 3 ft

Ambient temperature = 60°F

Belt width = 48 in.

Material = iron ore at 150 lbs/cu ft 10-in. maximum lumps from a gyratory crusher

Drive = lagged and grooved head pulley, 220-degree wrap

Troughing idlers = Class E6, 6-in. diameter, 20-degree angle

Return idlers = rubber-disc type, Class C6, 6-in. diameter, 10 ft spacing

Analysis:

From Table 6-8, drive factor, $C_w = 0.35$

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(3,400)}{500} = 226.4 \text{ lbs per ft}$$

From Figure 6.1 for 60°F, $K_t = 1.0$

Formula:

$$T_e = LK_t(K_x + K_yW_b + 0.015W_b) + W_m(LK_y + H) + T_{ac}$$

To find K_x and K_y it is necessary to find

$$W_b + W_m = 17 + 226.4 = 243.4 \text{ lbs/ft}$$

thus:

$$K_x = 1.099, \text{ for } 3.0 \text{ spacing } A_i = 2.8 \text{ and } W_b + W_m = 243.4 \text{ lbs from equation (3)}$$

$$K_y = 0.021, \text{ for } L = 2,400, \text{ slope } 0^\circ \text{ and } W_b + W_m = 243.4 \text{ lbs. Refer to Table 6-2.}$$

$$\begin{aligned} \text{Minimum tension, } T_0, \text{ for 3 percent sag} &= 4.2 S_i (W_b + W_m) = (4.2)(3)(243.4) \\ &= 3,067 \text{ lbs} \end{aligned}$$

Determine Accessories:

In this case the only accessories are pulley friction and a loading chute plus skirtboards. Assume that skirtboards are 10 ft long and spaced apart, two-thirds the width of the belt. The pull on the belt to overcome skirtboard friction is $T = C_s L_b h_s^2$. From the calculation of skirtboard friction, from page 101, 90, $h_s = (0.1)(48) = 4.8$ inches. C_s is (safely).276, from Table 6-7, for iron ore @ 150 pounds per cubic foot. Therefore, $T = (0.276)(10)(4.8)^2 = 64$ lbs. For the additional 20 ft of rubber edging on the skirtboards, additional resistance is $(3)(20) = 60$ lbs. Total skirtboard resistance, $T_{sb} = 64 + 60 = 124$ lbs.

$LK_t K_x$	$= (2,400)(1)(1.099)$	$=$	2,638
$LK_t K_y W_b$	$= (2,400)(1)(0.021)(17)$	$=$	857
$LK_t 0.015 W_b$	$= (2,400)(1)(0.015)(17)$	$=$	612
$K_y L W_m$	$= (0.021)(2,400)(226.4)$	$=$	11,411
$H W_m$	$= (0)(226.4)$	$=$	0
Nondriving pulley friction	$= (4)(100) + (2)(150) =$		700
Skirtboard resistance, T_{sb}			<u>124</u>
Effective tension, T_e		$=$	16,342 lbs
$T_2 = C_w T_e = (0.35)(16,342)$		$=$	<u>+ 5,720</u>
Maximum tension, $T_l = T_e + T_2$		$=$	22,062 lbs
Tail tension, $T_t = T_2 + 0.015 L K_t W_b + \text{pulley friction}$			$= 5,720 + 612 + 700 = 7,032 \text{ lbs}$

Final tensions:

$$\begin{aligned} T_e &= 16,342 \text{ lbs} \\ T_2 &= 5,720 \text{ lbs} \\ T_l &= 22,062 \text{ lbs} \\ T_t &= 7,032 \text{ lbs} \end{aligned}$$

Horsepower at Motor Shaft:

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(16,342)(500)}{33,000} = 247.61$$

$$\text{Drive pulley hp} = \frac{(200)(500)}{33,000} = 3.03$$

$$\text{Add 5\% for speed reduction loss} = 0.05(247.61 + 3.03) = 12.53$$

$$\text{Horsepower at motor shaft} = 263.17 \text{ hp}$$

$$\text{Belt tension} = \frac{T_I}{\text{width}} = \frac{22,062}{48} = 460 \text{ lbs per inch of belt width}$$

$$\text{Horsepower at motor shaft} = 263 \text{ (select 300 hp, 1,750 rpm, motor)}$$

Acceleration Calculations:

The following calculations are used to determine the acceleration forces and times:

WK^2 of drive (all values are taken at motor speed and should be obtained from the equipment manufacturer):

WK^2 of motor	=	101 lb-ft ²
Equivalent WK^2 of reducer	=	20 lb-ft ² (common practice is to take 1/5 of WK^2 of motor)
WK^2 of coupling	=	4 lb-ft ²
Equivalent WK^2 of drive pulley	=	5 lb-ft ²
Total WK^2 of drive	=	130 lb-ft ² , at motor speed

Converting this WK^2 value by using the equation for equivalent weight, page 139, 62,870 lbs is calculated as follows:

$$\text{Drive equivalent weight is } (130 \text{ lb-ft}^2) \left(\frac{1,750 \text{ rpm}}{500 \text{ fpm}} \right)^2 (2\pi)^2 = 62,870 \text{ lbs}$$

For purposes of calculating the equivalent weights, the pulley diameters must first be estimated. The diameters of the head and tail pulleys are assumed to be 42 in.; the rest of the pulleys are assumed to be 30 inches. The actual required pulley diameters are a function of the characteristics of the belt to be used and the belt tension at the pulley. The assumed pulley diameters should be reviewed after this information is known.

Conveyor equivalent weight is as follows:

Pulleys—From Table 8-1, one 42" diameter x 51" pulley with maximum bore of 5" weight = 1,275 lbs

From Table 8-1, five 30" diameter x 51" pulleys with maximum bore of 4", weight = 5 x 780 lbs = 3,900 lbs

6 (total) 1,275 + 3,900 = 5,175 lbs

Thus, 5,175 lbs is the approximate total weight for all of the nondriving pulleys on this conveyor. For more accurate calculations, the equipment manufacturer can supply actual weights. This weight is distributed among all of the elements that make up each pulley (rim, end and center discs, hubs, etc.). The belt must accelerate and decelerate these pulleys.

A generally accepted method for determining the equivalent weight of conveyor pulleys is to use $\frac{2}{3}$ of the actual total weight. Therefore,

$$\begin{aligned} \text{Belt, carrying run, from Table 6-1,} & \quad \frac{2}{3}(5,175 \text{ lbs}) &= 3,450 \text{ lbs} \\ & \quad 17 \text{ lbs/ft} \times 2,400 \text{ ft} &= 40,800 \text{ lbs} \\ \text{Belt, return run, 17 lbs/ft} \times (2,400 \text{ ft} + 30 \text{ ft}) & &= 41,310 \text{ lbs} \end{aligned}$$

Idlers, troughing, from Table 5-11, for 48" belt width and Class E6, the weight is 81.9 lbs

$$\text{thus, } 81.9 \text{ lbs} \times \frac{2,400 \text{ ft}}{3\text{-ft spacing}} = 65,520 \text{ lbs}$$

Idlers, return, from Table 5-12, for 48" belt width and Class C6, the weight is 48.4 lbs

$$\text{thus, } \frac{48.4 \text{ lbs} \times 2,400 \text{ ft}}{10\text{-ft spacing}} = 11,616 \text{ lbs}$$

Total conveyor equivalent weight = 162,696 lbs

Material load (226.4 lbs/ft)(2,400 ft) = 543,360 lbs

Total equivalent weight for system =

$$62,870 \text{ lbs} + 162,696 \text{ lbs} + 543,360 \text{ lbs} = 768,926 \text{ lbs}$$

Percent of total within the conveyor:

$$\frac{706,056}{768,926} \times 100\% = 91.8\%$$

Having selected a belt for $T_l = 22,062$ lbs, as explained in Chapter 7, at an allowable rating of 90 lbs/in./ply, 6 plies are required and the rated tension is 25,920 lbs.

If the starting tension is limited to 180 percent of the rated tension (see page 113), then the allowable extra belt tension for acceleration is

$$(1.80)(25,920) - (22,062) = 46,656 - 22,062 = 24,594 \text{ lbs}$$

The time for acceleration is found from the equation:

$$F_a t = M \frac{V_1 - V_0}{60}$$

where:

F_a = the allowable extra accelerating tension = 24,594 lbs

t = time, seconds

V_1 = final velocity = 500 fpm

V_0 = initial velocity = 0 fpm

M = mass of conveyor system = $\frac{706,056}{32.2} = 21,926$ slugs

Solving for t :

$$t = \frac{M}{F_a} \times \frac{(V_1 - V_0)}{60} = \frac{21,927}{24,594} \times \frac{(500 - 0)}{60} = 7.43 \text{ seconds}$$

This means that in order not to exceed the maximum permissible belt tension at 46,656 lbs, the time used for acceleration should not be less than 7.43 seconds.

Assume a 300 horsepower motor is used, with a maximum torque of 200 percent of the full-load torque. This corresponds to a force of 39,600 lbs acting at the belt line, if the friction losses of the drive are not considered and the belt speed is 500 fpm. This is not excessive when compared to the 46,656 lbs belt tension allowable at 180 percent of belt rating.

Another limiting factor may be the time that the motor needs to accelerate the system. The average torque – available during acceleration of the chosen motor – taken from its speed torque curve is 180 percent of full load torque. For a drive efficiency of 95 percent, it was found in Problem 3, that the horsepower at the motor shaft to operate the loaded conveyor is 263.17 hp.

$$\text{Horsepower} = \frac{(\text{pull in lbs})(\text{belt speed, fpm})}{33,000}$$

Therefore:

$$(\text{pull in lbs}) = \frac{(\text{hp})(33,000)}{(\text{belt speed, fpm})}$$

Also, the horsepower delivered by the motor is practically proportional to the torque, assuming no appreciable drop in speed from the full-load speed. Therefore, at 180 percent torque, the motor will deliver $(1.8)(300) = 540$ hp.

The force available for acceleration of the total equivalent mass of the loaded conveyor system, for a belt speed of 500 fpm, is:

$$\begin{aligned}
 F_a &= \left[\frac{(300)(1.8)(33,000)}{500} - \frac{(263.17)(33,000)}{500} \right] 0.95 \\
 &= 0.95 - [(300)(1.8) - 263.17] \frac{33,000}{500} \\
 &= \frac{(263)(33,000)}{500} = 17,358 \text{ lbs}
 \end{aligned}$$

The total equivalent mass = $768,926/32.2 = 23,880$ slugs

From the equation,

$$F_a = Ma$$

$$\text{the acceleration, } a = \frac{F_a}{M} = \frac{17,358}{23,880} = 0.727 \text{ ft per sec}^2$$

The time needed is:

$$t = \frac{V_t - V_o}{60a}$$

Therefore:

$$t = \frac{500 - 0}{(60)(0.727)} = 11.46 \text{ seconds.}$$

The time required by the motor to accelerate the loaded conveyor, 11.46 seconds, is greater than the minimum acceleration time to stay within the maximum allowable belt tension, 7.43 seconds. Therefore, the conveyor is safe to start, fully loaded, with the equipment selected.

Had the starting belt stress been limited to 120 percent of the normal belt rating instead of 180 percent, the allowable extra belt tension would have been $(1.2)(25,920) - (22,062) = 9,042$ lbs and the acceleration time, $t = (21,927)(500 - 0)/(9,042)(60) = 20.21$ sec minimum. This is more than the time calculated for the motor to accelerate the loaded system, 11.46 seconds. So, if such limitation had been placed on the starting belt stress, the system would not have been safe to start with the equipment selected. In fact, the belt stress during acceleration must be:

$$\begin{aligned}
 \text{extra belt tension} &= \frac{(21,927)(500 - 0)}{(11.46)(60)} = 15,752 \text{ lbs} \\
 \% \text{ of normal belt rating} &= \frac{(15,752) + 22,062}{25,920} (100\%) = 146\%
 \end{aligned}$$

The foregoing assumes that the mass between the slack side of the drive pulley and the takeup is negligible. If the takeup is far removed from the drive, this should be taken into account in the calculations.

In Chapter 13 it is indicated that the acceleration time for NEMA Type C motors, in general, be considered as 10 seconds or less. It, therefore, would be prudent to

check with the motor manufacturer to make sure that the calculated acceleration time of 11.46 seconds would not cause the motor to overheat during starting. In the case of *this particular problem*, the motor manufacturer was asked what maximum safe acceleration time would be for his 300 hp NEMA Type C motor. The manufacturer stated that any time up to 20 seconds would be permissible.

Therefore, the conveyor in this problem could be safely started with the equipment selected, provided the allowable belt tension during starting was 146 percent of normal belt rating, or greater. The use of the 300 hp NEMA Type C motor is justified, provided that operating conditions of this particular conveyor are such that abnormal starting conditions (which would require forces considerably in excess of those calculated) are unlikely to occur. If abnormal starting conditions are likely to occur, even infrequently, consideration should be given to the use of a different means of starting that would satisfy all the requirements described earlier in the section, "Acceleration and deceleration forces."

Deceleration Calculations:

In the preceding example on acceleration, it was found that the total equivalent mass of the conveyor system under normal conditions of operation is equal to 23,880 slugs (see page 158). As these calculations are based on the belt speed of 500 fpm or 8.33 fps, the kinetic energy of the system is:

$$\frac{MV^2}{2} = \frac{(23,880)(8.33)^2}{2} = 828,503 \text{ ft lbs}$$

Earlier in this problem (3), it was found that 263.17 hp is required to operate this conveyor at its rated speed of 500 fpm. Because the conveyor is horizontal, this represents the product of the friction forces and the distance traveled in unit time. This means that the frictional retarding force is:

$$\frac{(263.17)(33,000)}{500} = 17,369 \text{ lbs}$$

The average velocity of the conveyor during the deceleration period would be

$$\frac{500 + 0}{2} = 250 \text{ fpm}$$

Because the total work performed has to be equal to the kinetic energy of the total mass,

$$(t)(250 \text{ fpm})(17,369) = 828,503 \text{ ft lbs}$$

where:

t = time in minutes

Therefore:

$$t = \frac{828,503}{(250)(17,369)} = 0.191 \text{ minutes, or } 11.45 \text{ seconds}$$

and the belt will have moved $(0.191)(250 \text{ fpm}) = 47.75 \text{ ft}$ in this time. As the belt is fully loaded (by assumption), the belt will discharge the following amount of material:

$$\left(\frac{3,400 \text{ tph}}{60}\right)\left(\frac{47.75}{500}\right) = 5.41 \text{ tons}$$

If 5.41 tons of material discharged is objectionable, the use of a brake has to be considered. Such a step, however, can be justified only if the reduced deceleration time is still greater than, or at least equal to, the deceleration cycle of whatever piece of equipment delivers to the conveyor in this example.

Also, another difficulty arises. Suppose it is desirable or necessary to reduce the deceleration time from 11.45 seconds to 7 seconds. Since the total retarding force is inversely proportional to the deceleration time, the additional braking force required must be:

$$17,369 \times \left(\frac{11.45-7}{7}\right) = 11,042 \text{ lbs}$$

If the brake is connected to the drive pulley shaft, the drive pulley is required to transmit to the belt a braking force equal to

$$11,042 \left(\frac{768,926-(62,870)}{768,926}\right) = 10,139 \text{ lbs}$$

The difference between the 11,042 lbs and the 10,139 lbs is the braking force required to decelerate the drive and drive pulley and is not transmitted to the belt.

However, under coasting conditions, the belt tension is principally governed by the gravity takeup which, if located adjacent to the head pulley, would provide a maximum tension equal to T_2 , or 5,720 lbs. Obviously, it is impossible to secure a braking force of 10,139 lbs on the head pulley. Even a much smaller force than this would result in looseness of the belt around the head pulley.

The solution is to provide the braking action on the tail pulley, where it would increase rather than decrease the contact pressure between the belt and pulley. However, a further check on the tail pulley indicates that with 11,042 lbs braking tension, a plain bare tail pulley with 180-degree wrap angle could not produce a sufficient ratio of tight-side to slack-side tension.

Therefore, it would be necessary to do one or a combination of the following: increase the takeup tension weight, lag the tail pulley, or snub the tail pulley for a greater wrap angle. If the increasing takeup weight should result in a heavier and more costly belt carcass, the second and third remedies are preferable and more economical.

It should be noted that the above calculations are based on maximum friction losses and therefore will give a minimum coasting distance. Since most installations operate under variable conditions, braking and coasting problems should be investigated for a range of friction values. These lower friction values for K_x and K_y can be found by the methods outlined in Problem 2, and can result in a lower frictional retarding force approaching 60 percent of the original. This lower retarding force will show greater coasting distances or larger braking forces.

Problem 4

Complex Belt Line

In the previous examples, the application of the CEMA horsepower formula was limited to belt conveyors with a linear profile and an overall centers length not exceeding 3,000 ft. However, the CEMA horsepower formula can be applied to belt conveyors having more than one change in slope and a total centers length of more than 3,000 ft provided certain procedures are followed. This problem entails a belt conveyor that has two changes in slope and a total centers length of 4,000 ft.



Figure 6.23 Complex belt line.

The K_y factor is dependent upon the average tension in that portion of the belt in which the tension is being analyzed. Tables 6-2 and 6-3 were developed on the basis of the limitations and generalizations stated on page 91, and for normal average tensions in the belts within the limitations specified. For belt conveyors exceeding these limitations, it is necessary first to assume a tentative value for the average belt tension. The graphical method for conveyor horsepower determination, pages 141-145, may be of assistance in estimating this value. After estimating the average belt tension and idler spacing, reference to Table 6-4 will provide values for A and B for use in equation (4), page 96. By using this equation, an initial value for K_y can be calculated. The comparison of this calculated average belt tension with the tentative value will determine the need to select another assumed belt tension. The process should be repeated until there is reasonable agreement between the estimated and calculated average belt tensions.

The following example of a belt tension analysis of a 4000-ft belt conveyor with two changes of slope demonstrates the method of calculation and the use of the table. See Figure 6.23.

Belt conveyors with different profiles can be analyzed in a similar manner, but the various problems, which increase in importance with long and complex belt conveyors, must be carefully analyzed. It is suggested that the designer of such complex conveyors check calculations with a CEMA member company before establishing the final conveyor design.

Problem:

Determine the effective tension, T_e ; slack-side tension, T_2 ; maximum tension, T_1 ; tail tension, T_t ; concave curve tension at bottom of incline run, convex curve tension at top of incline run; belt horsepower at drive terminal; belt stress; resulting drive factor.

Accessories are omitted in this example in order to clarify the procedure; however, they should be included in actual practice.

Conveyor Specifications:

Q = capacity = 800 tph

Material = crushed limestone, 85 lbs/cu ft, 8-in maximum lumps

Ambient temperature, above freezing. Continuous operation

L = length = 4,000 ft

H = lift = 70 ft (see Figure 6.23)

S_i = idler spacing = 4 ft

b = belt width = 36 in.

V = speed = 400 fpm

Drive = dual, 380° wrap, both pulleys lagged

Troughing idlers = Class C6, 6 in diameter, 20° angle, $A_i = 1.5$

Return idlers = rubber-disc type, Class C6, 6 in diameter, 10 ft spacing

W_b = belt weight = 10 lbs/ft

Constants:

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(800)}{400} = 66.6 \text{ lbs per ft}$$

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i} = 0.00068(10 + 66.6) + \frac{1.5}{4} = 0.427$$

Analysis:

Since each portion of this conveyor is analyzed separately, and each is less than or equal to 3,000 ft in length, Table 6-2 can be used to obtain a tentative K_y factor in order to calculate the average belt tension. This K_y value is then checked by using Table 6-4, equation (4), page 96, and the average belt tension. The final tensions in each portion of the conveyor are then accurately determined.

The profile is divided into three portions: (1) initial horizontal portion, 3,000 ft long; (2) inclined portion, 800 ft long, 70-foot lift; (3) final horizontal portion, 200 ft long.

Initial horizontal section, 3000 ft long, 3 percent sag in belt,

where:

$$\begin{aligned} K_t &= 1.0 \\ K_x &= 0.427 \\ W_b &= 10 \text{ lbs/ft} \\ W_m &= 66.6 \text{ lbs/ft} \\ (W_b + W_m) &= 76.6 \text{ lbs/ft} \end{aligned}$$

K_y from Table 6-2 would be 0.023.

The *average tension* is:

$$\frac{T_t + K_t[K_x L + K_y L W_b] + K_y L W_m + T_t}{2}$$

Here, T_t is at least equal to T_0 . And T_0 , for 3 percent sag is

$$4.2 S_i (W_b + W_m) = (4.2)(4)(76.6) = 1,287 \text{ lbs}$$

Thus, the *average tension* is:

$$\frac{1,287 + (0.427)(3,000) + (0.023)(3,000)(76.6) + 1,287}{2}$$

or,

$$\frac{1,287 + 1,281 + 5,860 + 1,287}{2} = \frac{9,140}{2} = 4,570 \text{ lbs}$$

Equation (4) indicates $K_y = 0.0255$, for 4,570 lbs average tension, and $(W_b + W_m) = 76.6$ lbs. Re-estimate using $K_y = 0.0255$. *Average tension* is:

$$\frac{1,287 + 1,281 + (0.0255)(3,000)(76.6) + 1,287}{2}$$

or,

$$\frac{1,287 + 1,281 + 5,860 + 1,287}{2} = \frac{9,715}{2} = 4,858 \text{ lbs}$$

Equation (4) checks $K_y = 0.0255$, for an average tension of 4,858 lbs, and $(W_b + W_m) = 76.6$ lbs.

The formula for the actual tension because of friction in the initial horizontal portion (see page 117) is:

$$T_{fcx} = L_x[K_t(K_x + K_y W_b)] + L_x K_y W_m$$

where:

$$\begin{aligned} L_x &= 3,000 \text{ ft} \\ K_x &= 0.427 \\ K_t &= 1.0 \\ K_y &= 0.0255 \\ W_b &= 10 \text{ lbs/ft} \\ W_m &= 66.6 \text{ lbs/ft} \end{aligned}$$

Therefore:

$$\begin{aligned} T_{fcx} &= L_x[K_x + K_y(W_b + W_m)], \text{ since } K_t = 1.0 \\ T_{fcx} &= 3,000[0.427 + 0.0255(76.6)] \\ &= 3,000(2.38) \\ &= 7,141 \text{ lbs} \end{aligned}$$

The tension at the beginning of the vertical concave curve is calculated using the formula for belt tension at any point on the conveyor length, for point X on the carrying run (page 117), at the intersection of the initial horizontal run and the inclined run:

$$\begin{aligned} T_{cx} &= T_t + T_{wcx} + T_{fcx} \\ T_{wcx} &= H_x(W_b + W_m) = (0)(76.6) = 0 \end{aligned}$$

so

$$T_{cx} = 1,287 + 7,141 = 8,428 \text{ lbs}$$

The tension at the bottom of the incline, therefore, is 8,428 lbs. The estimated K_y is 0.024, for the first approximation of the calculation for the upper end of the incline from Table 6-2 for a value of $(W_b + W_m) = 76.6$, and a slope of $(70/800)(100\%) = 8.8\%$. Average tension is:

$$\frac{T_t + K_t(K_x L + K_y L W_b) + K_y L W_m + H(W_b + W_m) + T_t}{2}$$

in which T_t is the tension at the bottom of the incline, or 8,428 lbs, so,

$$\frac{8,428 + (0.427)(800) + (0.024)(800)(76.6) + (70)(76.6) + 8,428}{2}$$

or,

$$\frac{8,428 + 342 + 1,471 + 5,362 + 8,428}{2} = \frac{24,031}{2} = 12,016 \text{ lbs}$$

As stated on page 92, the minimum K_y value = .016 for 12,016 lbs tension and $W_b + W_m = 76.6$. Re-estimate using $K_y = .016$. Average tension then is:

$$\frac{8,428 + 342 + (0.016)(800)(76.6) + 5,362 + 8,428}{2}$$

or,

$$\frac{8,428 + 342 + 980 + 5,362 + 8,428}{2} = \frac{23,540}{2} = 11,770 \text{ lbs}$$

This checks $K_y = 0.016$ minimum value, for average tension of 11,770 lbs, and $W_b + W_m = 76.6$ lbs/ft.

From page 117, $T_{cx} = T_t + T_{wcx} + T_{fcx}$. Here, T_t is 8,428 lbs, the tension at the bottom of an incline.

$$\begin{aligned} T_{wcx} &= H_x(W_b + W_m) = (70)(76.6) = 5,362 \text{ lbs} \\ T_{fcx} &= L_x K_t [K_x + K_y(W_b + W_m)] \text{ where } L_x = 800 \text{ ft and } K_t = 1.0 \\ &= (800)(1.0)[0.427 + 0.016(76.6)] \\ &= 800(1.653) \\ &= 1,322 \text{ lbs} \\ T_{cx} &= 8,428 + 5,362 + 1,322 = 15,112 \text{ lbs} \end{aligned}$$

The tension at the top of the incline, then, is 15,112 lbs.

The final horizontal portion is 200 ft long:

$$\begin{aligned} K_t &= 1.0 \\ K_x &= 0.427 \\ W_b &= 10 \text{ lbs/ft} \\ W_m &= 66.6 \text{ lbs/ft} \\ W_b + W_m &= 76.6 \text{ lbs/ft} \end{aligned}$$

K_y will be at a minimum value because of the high tension that is obvious in this portion of the belt. From page 91, the minimum K_y of 0.016 is applicable at the indicated *average* tension (obviously more than 15,112 lbs) and with $W_b + W_m = 76.6$ lbs/ft.

From page 117, $T_{cx} = T_l + T_{wcx} + T_{fcx}$. Here, T_t is the tension at the beginning of this horizontal section, or 15,112 lbs and $T_{wcx} = 0$, since $H_x = 0$.

$$\begin{aligned} T_{fcx} &= L_x K_t (K_x + K_y W_b) + L_x K_y W_m \text{ where } L_x = 200 \text{ ft and } K_t = 1.0 \\ &= (200)(1)(0.427 + (0.016)(10)) + (200)(0.016)(66.6) \\ &= 117 + 213 = 330 \text{ lbs} \\ T_{cx} &= 15,112 + 330 = 15,442 \text{ lbs} \end{aligned}$$

In this case, $T_{cx} = T_l = 15,442$ lbs

The final tension at the head pulley is 15,442 lbs. $T_e = T_l - T_2$. To find T_2 , refer to Figure 6.7A, where $T_t = T_2 - T_b + T_{yr}$. The tail tension, T_t , was taken at $T_0 = 1,287$ lbs to avoid more than 3% belt sag between idlers.

Thus,

$$\begin{aligned}1,287 &= T_2 - T_b + T_{yr} \\ T_b &= HW_b = (70)(10) = 700 \text{ lbs} \\ T_{yr} &= 0.015LW_bK_t = (0.015)(3,000 + 800 + 200)(10)(1) = 600 \text{ lbs}\end{aligned}$$

Thus,

$$\begin{aligned}1,287 &= T_2 - 700 + 600, \text{ or } T_2 = 1,387 \text{ lbs} \\ T_e &= T_l - T_2 = 15,442 - 1,387 = 14,055 \text{ lbs}\end{aligned}$$

With the T_e and T_2 tensions now known, it is necessary to check the wrap factor, C_w . A 380-degree wrap, dual-pulley drive with lagged pulleys, requires a $C_w = 0.11$. (See Table 6-8.) From the known tensions:

$$C_w = \frac{T_2}{T_e} = \frac{1,387}{14,055} = 0.099$$

Since this is less than the required 0.11, the belt may slip on the drive pulleys. This situation can be corrected in one of two ways: (1) The wrap on the drive pulleys can be increased from 380 to 405 degrees, or (2) the takeup weight can be increased until $T_2/T_e = 0.11$. This requires an increase in all tensions of $(0.11 \times 14,055) - 1,387 = 160$ lbs.

Assume all tensions are increased by 160 lbs:

$$\begin{aligned}T_l &= 15,442 + 160 = 15,602 \\ T_2 &= 1,387 + 160 = 1,547 \\ T_t &= 1,287 + 160 = 1,447\end{aligned}$$

$$\text{Belt stress} = \frac{T_l}{\text{Belt width}} = \frac{15,602}{36} = 433 \text{ lbs per inch of width (PIW)}$$

Horsepower at belt line, excluding all accessories, is as follows:

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(14,055)(400)}{33,000} = 170.36$$

If drive efficiency = .94, horsepower at motor shaft = $170.36/.94 = 181.23$ hp. Acceleration and deceleration calculations for this example follow. For radii of concave and convex curves for this example, refer to Chapter 9.

Acceleration Calculations:

$$T_e = 14,055 \text{ lbs}$$

$$T_2 = 1,547 \text{ lbs}$$

$$T_1 = 15,602 \text{ lbs}$$

$$T_t = 1,447 \text{ lbs}$$

$$b = \text{Belt width} = 36 \text{ in.}$$

$$L = \text{Length} = 4,000 \text{ ft}$$

$$H = \text{Lift} = 70 \text{ ft}$$

$$Q = \text{Capacity} = 800 \text{ tph}$$

$$V = \text{speed} = 400 \text{ fpm}$$

$$\text{Material} = \text{crushed limestone at } 85 \text{ lbs/cu ft}$$

$$W_m = \text{Material weight} = 66.6 \text{ lbs/ft}$$

$$W_b = \text{Belt weight} = 10 \text{ lbs/ft}$$

$$\text{Troughing idlers Class C6, 6-in. diameter } 20^\circ \text{ angle, at 4 foot spacing}$$

$$\text{Rubber-disc return idlers, Class C6 6-in. diameter at 10 foot spacing}$$

$$\text{Horsepower at motor shaft} = 181.23 \text{ (select one 75 hp and one 125 hp motor, each to be 1,750 rpm)}$$

$$WK^2 \text{ of drive (all values are taken at motor speed and should be obtained from the equipment manufacturer)}$$

$$WK^2 \text{ of motor} = 58.0 \text{ lbs-ft}^2$$

$$\text{Equivalent } WK^2 \text{ of reducer} = 11.6 \text{ lbs-ft}^2$$

(It is common practice to estimate the WK^2 of the reducer to be 20 percent of the WK^2 of the motor)

$$WK^2 \text{ of coupling} = 2.0 \text{ lbs-ft}^2$$

$$\text{Equivalent } WK^2 \text{ of drive pulley} = 2.0 \text{ lbs-ft}^2$$

$$\text{Total } WK^2 = 73.6 \text{ lbs-ft}^2, \text{ at motor speed}$$

Converting this WK^2 value to the equivalent weight at the belt line,

$$\text{Drive equivalent weight (lbs)} = (WK^2) \left(\frac{2\pi \text{ rpm}}{V} \right)^2$$

$$\text{Drive equivalent weight is } (73.6 \text{ lb-ft}^2) \left(\frac{1,750 \text{ rpm}}{400 \text{ fpm}} \right)^2 (2\pi)^2 = 55,615 \text{ lbs}$$

Conveyor equivalent weight is as follows:

Pulleys — for reasons given on page 155, first assume the non-drive pulley diameter: two at 48 in. and four at 36 in.

From Table 8-1, two (48 in. diameter x 38 in.) pulleys with max. bore of 5 in.,
weight = $2 \times 1,270 = 2,540 \text{ lbs}$

From Table 8-1, four (36 in. diameter x 38 in.) pulleys with max. bore of 4 in.,
weight = $4 \times 715 = 2,860 \text{ lbs}$

$$\begin{array}{rcl} \text{From page 156, } \frac{2}{3}(2,540 \text{ lbs} + 2,860 \text{ lbs}) & = & 3,600 \text{ lbs} \\ \text{Belt, carrying run, from Table 6-1, } 10 \text{ lbs/ft} \times 4,000 \text{ ft} & = & \underline{40,000} \text{ lbs} \\ \text{Subtotal} & = & 43,600 \text{ lbs} \end{array}$$

$$\text{Belt, return run, } 10 \text{ lbs/ft} \times (4,000 \text{ ft} + 50 \text{ ft}) = 40,500 \text{ lbs}$$

Idlers, troughing, from Table 5-11, for 36-in. belt width and class C6, the weight is 43.6 lbs.

$$43.6 \left(\frac{4,000 \text{ ft}}{4\text{-ft idler spacing}} \right) = 43,600 \text{ lbs}$$

Idlers, return, from Table 5-12, for 36-in. belt width and Class C6, the weight is 37.6 lbs.

$$37.6 \left(\frac{4,000 \text{ ft}}{10\text{-ft idler spacing}} \right) = \underline{15,040} \text{ lbs}$$

$$\text{Total conveyor equivalent weight} = 142,740 \text{ lbs}$$

$$\text{Material load } (66.6 \text{ lbs/ft})(4,000 \text{ ft}) = 266,400 \text{ lbs}$$

$$\begin{array}{rcl} \text{Total equivalent weight of system} = & & \\ 55,615 \text{ lbs} + 142,740 \text{ lbs} + 266,400 \text{ lbs} & = & 464,755 \text{ lbs} \end{array}$$

Percent of total within the conveyor:

$$(142,740 + 266,400)/464,755 \times 100\% = 88\%$$

Having selected a belt for $T_l = 15,602$ lbs, as explained in Chapter 7, at an allowable rating of 70 lbs/in./ply, 7 plies are required and the rated tension is 17,640 lbs.

If the starting tension is limited to 180 percent of the rated tension (see page 113), then the allowable extra tension is:

$$(1.80)(17,640) - (15,602) = 31,752 - 15,602 = 16,150 \text{ lbs}$$

The time for acceleration is found from the equation:

$$F_a t = M \frac{V_1 - V_0}{60}$$

Where:

$$F_a = \text{the allowable extra accelerating tension} = 16,150 \text{ lbs}$$

$$t = \text{time, seconds}$$

$$V_1 = \text{final velocity} = 400 \text{ fpm}$$

$$V_0 = \text{initial velocity} = 0 \text{ fpm}$$

$$M = \text{mass of conveyor system} = \frac{409,140}{32.2} = 12,706 \text{ slugs}$$

Solving for t :

$$t = \frac{M(V_1 - V_0)}{F_a \cdot 60} = \frac{12,706}{16,150} = \frac{(400 - 0)}{60} = 5.24 \text{ seconds}$$

This means that in order not to exceed the maximum permissible belt tension at 31,752 lbs, the time used for acceleration should not be less than 5.24 seconds.

In determining the starting tension in the belt, the first step is to find the total horsepower available, in the form of tension. From this value, subtract the total tension to operate the loaded conveyor. The result will be the force available to accelerate the total system. The total horsepower, in the form of tension, available to accelerate the entire system comes from the 75 hp and 125 hp motors. The starting torque available from these NEMA Type C motors is a variable that should be confirmed by the motor manufacturer. For this example, the value is assumed to be 200 percent of the motor rating.

Then, the total tension available is:

$$\frac{2(75 + 125)(33,000)}{400} = 33,000 \text{ lbs}$$

From this value we subtract the tension required to operate the loaded conveyor:

$$33,000 - \frac{14,055}{0.94(\text{drive efficiency})} = 18,048 \text{ lbs}$$

The resulting tension available to accelerate the loaded conveyor is 18,048 lbs. The acceleration of the total system consists of the acceleration of the drive (12 percent of the total system) and of the conveyor (88 percent of the total system). However, in the process of acceleration, some amount of the available force (tension) is absorbed by the frictional losses (heat) in the drive machinery. Since this is a small amount compared to the total, a conservative approach is to ignore these losses because it is our aim to determine the effect of acceleration on the belt and its capacity to withstand tensile forces. Therefore, $0.88 \times 18,048 = 15,882$ lbs, which is the acceleration force (expressed in lbs of belt tension). The operating T_l is then added to this value in order to obtain the actual starting tension in the belt, which is $15,602 + 15,882 = 31,484$ lbs. This is not excessive when compared to the 31,752 lbs belt tension allowable at 180 percent of belt rating.

Another limiting factor may be the time that the motor needs to accelerate the system. The average torque available during acceleration of the chosen motor taken from its speed torque curve is 180 percent of full-load torque. For a drive efficiency of 94 percent, it was found in Problem 4, page 166, that the horsepower at the motor shaft to operate the loaded conveyor is 181.23 hp.

$$\text{Horsepower} = \frac{(\text{pull in lbs})(\text{belt speed, fpm})}{33,000}$$

Therefore:

$$(\text{pull in lbs}) = \frac{(\text{hp})(33,000)}{(\text{belt speed, fpm})}$$

The force available for acceleration of the total equivalent mass of the loaded conveyor system, for a belt speed of 400 fpm, is

$$\begin{aligned} F_a &= \left[\frac{(200)(1.80)(33,000)}{400} - \frac{(181.23)(33,000)}{400} \right] 0.94 \\ &= \frac{0.94 \times 33,000}{400} (200 \times 1.8 - 181.23) \\ &= 13,864 \text{ lbs} \end{aligned}$$

The total equivalent mass = $464,755/32.2 = 14,433$ slugs. From the equation,

$$F_a = Ma$$

$$\text{the acceleration, } a = \frac{F_a}{M} = \frac{13,864}{14,433} = 0.96 \text{ ft per sec}^2$$

The time needed is:

$$t = \frac{V_1 - V_0}{60a}$$

Therefore:

$$t = \frac{400 - 0}{(60)(0.96)} = 6.94 \text{ seconds}$$

The time required by the motor to accelerate the loaded conveyor, 6.94 seconds, is greater than the minimum acceleration time to stay within the maximum allowable belt tension, 5.24 seconds. Therefore, the conveyor is safe to start, fully loaded, with the equipment selected.

Had the starting belt stress been limited to 140 percent of the normal belt rating instead of 180 percent, the allowable extra belt tension would have been $(1.4)(17,640) - (15,602) = 9,094$ lbs and the acceleration time $t = (12,706)(400 - 0)/(9,094)(60) = 9.31$ sec minimum. This is more than the time calculated for the motor to accelerate the loaded system, 6.94 seconds. If such a limitation had been placed on the starting belt stress, the system would not have been safe to start with the equipment selected. In fact, the belt stress during acceleration must be:

$$\begin{aligned} \text{extra belt tension} &= \frac{(12,706)(400 - 0)}{(6.94)(60)} = 12,206 \text{ lbs} \\ \% \text{ of normal belt rating} &= \frac{12,206 + 15,602}{(17,640)} (100\%) = 158\% \end{aligned}$$

The foregoing assumes that the mass between the slack side of the drive pulley and the takeup is negligible. If the takeup is far removed from the drive, this should be taken into account in the calculations.

In Chapter 13, "Accelerating time," it is indicated, in general, that the acceleration time for NEMA Type C motors be considered as 10 seconds or less. It is always prudent to check with the motor manufacturer to make sure that the calculated acceleration time will not cause the motor to overheat during starting.

Deceleration Calculations:

In the foregoing example of acceleration calculations, it was found that the total equivalent mass of the conveyor system under normal conditions of operation is equal to 14,433 slugs. As these calculations are based on the belt speed of 400 fpm or 6.67 fps, the kinetic energy of the system is:

$$\frac{MV^2}{2} = \frac{14,433 \times 6.67^2}{2} = 320,733 \text{ ft-lbs}$$

On page 167 it was calculated that 181.23 hp is required at the motor shaft to operate this conveyor at its rated speed of 400 fpm. This represents the product of the friction plus gravity forces and the distance traveled in unit time. This means that the frictional plus gravitational retarding force is:

$$\frac{(181.23)(33,000)}{400} = 14,951 \text{ lbs}$$

The average velocity of the conveyor during the deceleration period would be:

$$\frac{400 + 0}{2} = 200 \text{ fpm}$$

Because the total work performed has to be equal to the kinetic energy of the total mass:

$$(t)(200 \text{ fpm})(14,951) = 320,733 \text{ ft lbs}$$

where:

t = time in minutes

Therefore:

$$t = \frac{320,733}{(200)(14,951)} = 0.1073 \text{ minutes, or 6.44 seconds}$$

and the belt will have moved $(0.1073)(200 \text{ fpm}) = 21.46 \text{ ft}$ in this time. As the belt is fully loaded (by assumption), it will discharge the following amount of material:

$$\left(\frac{800 \text{ tph}}{60}\right)\left(\frac{21.46}{400}\right) = 0.72 \text{ tons}$$

If 0.72 tons of material discharge is objectionable, the use of a brake has to be considered. Such a step, however, can be justified only if the reduced deceleration time is still greater than, or at least equal to, the deceleration cycle of whatever piece of equipment delivers to the conveyor in this example.

Also, another difficulty arises. Suppose it is desirable or necessary to reduce the deceleration time from 6.44 seconds to 5 seconds. Since the total retarding force is inversely proportional to the deceleration time, the additional braking force required must be:

$$14,951 \times \left(\frac{6.44 - 5}{5} \right) = 4,306 \text{ lbs}$$

If the brake is connected to the drive pulley shaft, the drive pulley is required to transmit to the belt a braking force equal to

$$4,306 \text{ lbs} \times .88 = 3,789 \text{ lbs}$$

The difference between the 4,306 lbs and the 3,789 lbs is the braking force required to decelerate the drive and drive pulley and is not transmitted to the belt.

However, under coasting conditions, the belt tension is principally governed by the gravity takeup which, if located adjacent to the head pulley, would provide a maximum tension equal to T_2 , or 1,547 lbs. Obviously, it is impossible to develop a braking force of 3,789 lbs on the head pulley. Even a much smaller force than this would result in looseness of the belt around the head pulley.

The solution is to provide the braking action on the tail pulley where it would increase rather than decrease the contact pressure between the belt and pulley. However, a further check on the tail pulley indicates that with 3,789 lbs braking tension, a plain, bare tail pulley with a 180-degree wrap angle could not produce a sufficient ratio of tight-side to slack-side tension.

Therefore, it would be necessary to do one or a combination of the following: increase the takeup tension weight, lag the tail pulley, or snub the tail pulley for a greater wrap angle. If the increased takeup weight should result in a heavier and more costly belt carcass, the second and third remedies are preferable and more economical.

It should be noted that the above calculations are based on maximum friction losses and therefore will give a minimum coasting distance. Since most installations operate under variable conditions, braking and coasting problems should be investigated for a range of friction values. These lower friction values for K_x and K_y can be found by the methods outlined in Problem 2, and can result in a lower frictional retarding force approaching 60 percent of the original. This lower retarding force will show greater coasting distances or larger braking forces.

Problems 5 and 6

Comparison of Tension and Horsepower Values on Two Similar Conveyors

The two belt conveyors compared here in Problems 5 and 6 have the same load capacity, carry the same bulk material, have the same length, the same operating speed, and the same lift. The only difference is that one conveyor has a concave vertical curve while the other has a convex vertical curve.

The CEMA formula for power to operate belt conveyors determines the effective tension, T_e . The previous examples show how to obtain T_2 , T_1 , and T_t . *Frequently, the belt conveyor designer will require belt tensions elsewhere*; for instance, in the determination of the radius of a concave vertical curve. For a discussion of the belt tensions at any point on a belt conveyor, see "Belt Tension at Any Point, X, on Conveyor Length," page 117. Formulas for the belt tensions in belt conveyors having concave and convex vertical curves are given in Figures 6.8 through 6.16, inclusive.

The comparison of the two belt conveyors in Problems 5 and 6 shows how the factor K_y changes with increasing belt tension.

In Problem 5, Figure 6.24, the K_y factor for the tail half, L_1 , of the conveyor is selected for a 300-foot horizontal conveyor. The K_y factor for the inclined drive half, L_2 , is selected for the total conveyor length of 600 ft with an average slope of lift/total length = $36/600 = 6$ percent, because the belt tension is higher than it would be for a 300-foot inclined conveyor, due to the belt pull at the end of the horizontal portion.

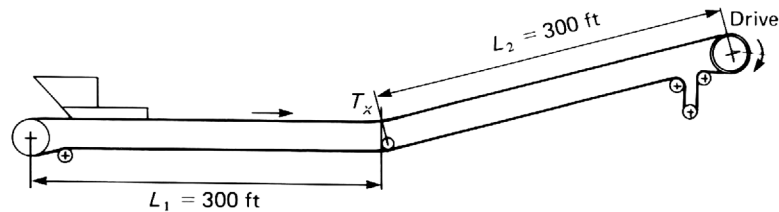


Figure 6.24 Belt conveyor with concave vertical curve.

In Problem 6, Figure 6.25, the K_y factor for the tail half, L_1 , of the conveyor is selected for a 300-foot conveyor inclined at a slope of lift/300 ft = $36/300 = 12$ percent. The K_y factor for the drive half, L_2 , of the conveyor is less than it would be for a 300-ft horizontal conveyor, because of the high belt tension existing at the top of the inclined portion. The criterion for determining the K_y value to use for the horizontal drive half, L_2 , of this conveyor is the K_y value of a 600-foot inclined conveyor at a 6 percent slope. The K_y value for the horizontal portion cannot be more, and probably is a little less than this K_y value.

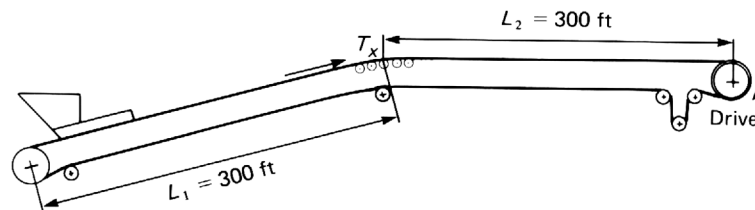


Figure 6.25 Belt conveyor with convex vertical curve.

The difference in the calculated effective tensions in Problems 5 and 6 is small. But larger and longer conveyors would entail more significant differences.

$$W_b = 10 \text{ lbs/ft from Table 6-1}$$

$$H = 36 \text{ ft}$$

$$L = 600 \text{ ft}$$

$$L_1 = 300 \text{ ft}$$

$$L_2 = 300 \text{ ft}$$

$$V = 500 \text{ fpm}$$

$$Q = 1,000 \text{ tph}$$

$$S_i = 4.5 \text{ ft}$$

$$\text{Ambient temperature} = 60^\circ\text{F}$$

$$\text{Belt width} = 36 \text{ in.}$$

$$\text{Material} = 100 \text{ lbs/cu ft}$$

$$\text{Drive} = \text{lagged head pulley, wrap} = 220^\circ$$

Troughing idlers = Class E6, 6-in. diameter 20° angle, $A_i = 2.8$

Return idlers = Class C6, 6-in. diameter, 10-foot spacing

To simplify the calculations, all accessories are omitted.

Analysis (Problem 5, Figure 6.24):

From Table 6-8, wrap factor, $C_w = 0.35$. From Figure 6.1, for 60°F, $K_t = 1.0$

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(1,000)}{500} = 66.6 \text{ lbs per ft}$$

$$W_b + W_m = 10 + 66.6 = 76.6 \text{ lbs/ft}$$

$$T_0, \text{ minimum tension for 3 percent sag} = 4.2 S_i (W_b + W_m) = (4.2)(4.5)(76.6) = 1,448 \text{ lbs}$$

$$T_1 \text{ is taken as } T_0 = 1,448 \text{ lbs}$$

$$T_2 = T_t - 0.015LW_b + HW_b = 1,448 - (0.015)(600)(10) + (36)(10) = 1,448 - 90 + 360 = 1,718 \text{ lbs}$$

$$K_x = (0.000568)(W_b + W_m) + \frac{A_i}{S_i} = (0.00068)(76.6) + \frac{2.8}{4.5} = 0.6743$$

The effective belt tension, T_e , is figured individually for each half of the conveyor.

Horizontal portion, 300 ft long, K_y from Table 6-2, for 0° slope, 300 ft and $(W_b + W_m) = 76.6$, is 0.0347. Corrected for 4.5-foot idler spacing, Table 6-3, gives $K_y = 0.0349$.

From “Belt Tension at Any Point, X, on Conveyor Length,” page 117, tension is:

$$T_{fcx} = T_t + T_{wcx} + T_{fcx}$$

$$\text{but, } T_{wcx} = H_x(W_b + W_m) = 0, \text{ for a horizontal belt}$$

$$\text{and } T_{fcx} = L_x[K_t(K_x + K_y W_b)] + L_x K_y W_m$$

$$\text{and } K_t = 1.0 \text{ for } 60^\circ\text{F}$$

Therefore:

$$T_{fcx} = L_x K_x + L_x K_y W_b + L_x K_y W_m = L_x K_x + L_x K_y (W_b + W_m)$$

Thus,

$$T_{cx} = T_t + 0 + L_x K_x + L_x K_y (W_b + W_m)$$

Calling L_x equal to L_1 for the first (horizontal) half of the conveyor:

$$\begin{aligned} T_{cx} &= T_t + L_1 K_x + L_1 K_y (W_b + W_m) = 1,448 + (300)(0.6743) \\ &\quad + (300)(0.0349)(76.6) = 1,448 + 202 + 802 \\ &= 2,452 \text{ lbs} \end{aligned}$$

For the inclined portion, drive half L_2 .

$$K_x = 0.6743$$

K_y is 0.028, for a slope of $(36/600)(100\%) = 6\%$, and $W_b + W_m = 76.6$, and a length of 600 ft, from Table 6-2 for the tabular idler spacing. The corrected value is 0.0298, from Table 6-3, for a 4½-foot spacing.

$$T_{cx} = T_t + T_{wcx} + T_{fcx}. \text{ However, } T_t \text{ is the tension existing at the bottom of the incline, so:}$$

$$\begin{aligned} T_{cx} &= 2,452 + T_{wcx} + T_{fcx} = 2,452 + H_x(W_b + W_m) \\ &\quad + L_x K_x + L_x K_y (W_b + W_m) \end{aligned}$$

substituting L_2 for L_x , and 36 for H_x

$$\begin{aligned} T_{cx} &= 2,452 + 36(W_b + W_m) + L_2 K_x + L_2 K_y (W_b + W_m) \\ &= 2,452 + (36)(76.6) + (300)(.6743) + (300)(.0298)(76.6) \\ &= 2,452 + 2,757.6 + 202.3 + 684.8 \\ &= 6,097 \text{ lbs} \end{aligned}$$

Adding to T_{cx} the nondriving pulley friction, $(2)(150) + (4)(100) = 700$ lbs, the tension in the belt at the head pulley $= T_{cx} + 700 = T_1 = 6,097 + 700 = 6,797$ lbs

$$T_e = T_1 - T_2 = 6,797 - 1,718 = 5,079 \text{ lbs}$$

$$\text{Belt horsepower} = \frac{T_e V}{33,000} = \frac{(5,079)(500)}{33,000} = 77 \text{ hp}$$

Analysis (Problem 6, Figure 6.25):

T_0 has been calculated in Problem 5 as 1,448 lbs.

$$\begin{aligned} \text{Take } T_t &= T_0 = 1,448 \text{ lbs} \\ T_t &= T_2 + L(0.015W_b) - H_{wb}, \\ T_2 &= T_t - L(0.015W_b) + HW_b = 1,448 - (600)(0.015)(10) + (36)(10) \\ &\quad + (36)(10) = 1,448 - 90 + 360 = 1,718 \text{ lbs} \end{aligned}$$

Inclined portion, 300 ft long. The slope of the incline is $(36/300)(100\%) = 12\%$. For this slope, a length of 300 ft and $W_b + W_m = 76.6$, the value of K_y , from Table 6-2,

is 0.0293. Corrected for 4½-foot idler spacing, from Table 6-3, $K_y = 0.0312$. K_x already has been calculated in Problem 5 as 0.6743.

From “Belt Tension at Any Point, X, on Conveyor Length,” page 117, the tension at point X in the carrying run is:

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

Since $K_t = 1.0$ for 60°F, the equation can be written:

$$T_{cx} = T_t + H_x(W_b + W_m) + L_x K_x + L_x K_y(W_b + W_m)$$

So, for the inclined tail half ($L_1 = L_x$) of the conveyor,

$$\begin{aligned} T_{cx} &= 1,448 + (36)(76.6) + (300)(0.6743) + (300)(0.0312)(76.6) \\ &= 1,448 + 2,757.6 + 202.3 + 716.9 = 5,125 \text{ lbs} \end{aligned}$$

$T_{cx} = 5,125$ lbs. This, then, is the tension in the belt at the top of the incline and at the beginning of the horizontal portion.

Horizontal portion = 300 ft long

$$K_x = 0.6743$$

K_y is dependent on the average belt tension, which, as is seen from the preceding calculation, will be very high. Used as a criterion of K_y , the value of K_y is 0.028, calculated from a 600-foot long inclined conveyor, at an average slope of 6%, from Table 6-2. Corrected for 4½-foot idler spacing, Table 6-3 gives $K_y = 0.0298$.

From “Belt Tension at Any Point, X, on Conveyor Length,” page 117, the tension at point X in the carrying run (at the head pulley, in this case) is $T_{cx} = T_t + T_{wcx} + T_{fcx}$.

However, T_t is the tension at the start of the horizontal run = 5,125 lbs. $T_{wcx} = H_x(W_b + W_m)$. And since $H_x = 0$, then $T_{wcx} = 0$. Also, $L_x = L_2 = 300$ ft.

$$\begin{aligned} T_{cx} &= 5,125 + T_{fcx} = 5,125 + L_2 K_x + L_2 K_y(W_b + W_m) \\ &= 5,125 + (300)(0.6743) + (300)(0.0298)(76.6) \\ &= 5,125 + 202 + 685 = 6,012 \text{ lbs} \end{aligned}$$

Add to T_{cx} the nondriving pulley friction $(2)(150) + (4)(100) = 700$ lbs

$$T_{cx} + 700 = T_1 = 6,012 + 700 = 6,712 \text{ lbs}$$

$$T_e = T_1 - T_2 = 6,712 - 1,718 = 4,994 \text{ lbs}$$

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(4,994)(500)}{33,000} = 75.7 \text{ hp}$$

The comparison of these two conveyors, each having the same length, lift, size, speed, and capacity, and handling the same material, shows that the conveyor with the concave curve will have a higher belt tension at the head pulley, will require a higher effective tension, and will require more horsepower than the conveyor with the convex curve.

Belt Conveyor Drive Equipment

The engineering of practically all belt conveyor installations involves a comprehensive knowledge of the proper application of conveyor drive equipment, including speed reduction mechanisms, electric motors and controls, and safety devices.

Belt Conveyor Drive Location

The best place for the drive of a belt conveyor is at the location that results in the lowest maximum belt tension. For horizontal or inclined conveyors, the drive usually is at the discharge end, while for declined conveyors the drive is usually at the loading end. For special conditions and requirements, it may be advisable to locate the drive elsewhere. See “Drive Arrangements” and “Analysis of Belt Tensions.”

Economics, accessibility, or maintenance requirements may make it preferable to locate the drive internally on the conveyor. Frequently, for the larger conveyors, a saving in supporting structures can be realized by doing so. Inclined boom conveyors sometimes are driven at the loading end for this reason.

Belt Conveyor Drive Arrangement

Belt conveyor drive equipment normally consists of a motor, speed reduction equipment, and drive shaft, together with the necessary machinery to transmit power from one unit to the next. The simplest drive, using the minimum number of units, usually is the best. However, economic reasons may dictate the inclusion of special-purpose units in the drive. These special-purpose units may be required to modify starting or stopping characteristics, to provide hold-back devices, or perhaps to vary the belt speed.

Speed-Reduction Mechanisms

The illustrations in Figures 6.26 through 6.33 show most of the belt conveyor drive equipment assemblies currently in common use.

The following comments apply to these figures:

- Figure 6.26, Gear motor directly connected by flexible coupling to drive shaft, is a simple, reliable and economical drive.

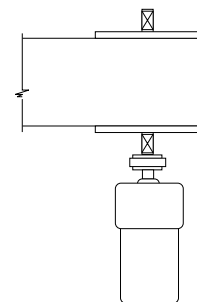
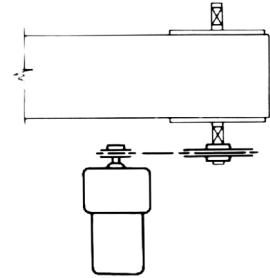


Figure 6.26 Gear motor is directly connected, by a flexible coupling, to the motor's drive shaft.

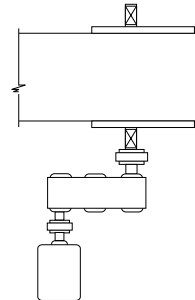
- Figure 6.27 — Gear motor combined with chain drive to drive shaft — is one of the lowest cost flexible arrangements and is substantially reliable.

Figure 6.27 Gearmotor combined with chain drive or synchronous belt drive to drive shaft — is one of the lowest cost flexible arrangements, provides additional reduction, and is substantially reliable.



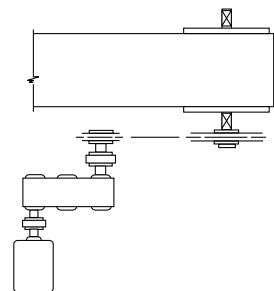
- Figure 6.28 — Parallel-shaft speed reducer directly coupled to the motor and to drive shaft — is versatile, reliable, and generally heavier in construction and easy to maintain.

Figure 6.28 Parallel-shaft speed reducer directly coupled to the motor and to drive shaft — is particularly well suited to large conveyors, is versatile, reliable, and generally heavier in construction and easy to maintain.



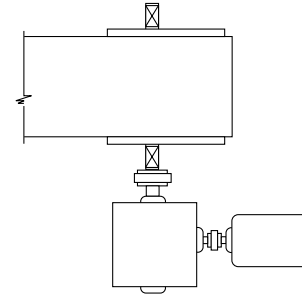
- Figure 6.29 — Parallel-shaft speed reducer coupled to motor, and with chain drive, to drive shaft — provides flexibility of location and also is suitable for the higher horsepower requirements.

Figure 6.29 Parallel-shaft speed reducer coupled to motor, and with chain drive, to drive shaft — provides flexibility of location and also is suitable for low speed high torque requirements.



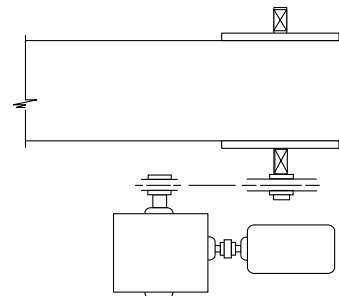
- Figure 6.30 — Spiral-bevel helical speed reducer, or worm-gear speed reducer, directly coupled to motor and to drive shaft — is often desirable for space-saving reasons and simplicity of supports. The spiral-bevel speed reducer costs substantially more than the worm-gear speed reducer but is considerably more efficient.

Figure 6.30 Spiral-bevel helical speed reducer, helical-worm speed reducer, or worm-gear speed reducer, directly coupled to motor and to drive shaft — is often desirable for space-saving reasons and simplicity of supports. The spiral-bevel speed reducer costs somewhat more than the helical-worm speed reducer and considerably more than the worm-gear speed reducer but is more efficient than the helical-worm and considerably more efficient than the worm gear.



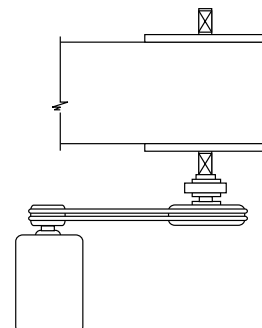
- Figure 6.31 — Spiral-bevel helical speed reducer, or worm-gear speed reducer, coupled to motor and, with chain drive, to drive shaft — is a desirable selection for high reduction ratios in the lower horsepower requirements. This drive is slightly less efficient, but has lower initial costs and is most flexible in terms of location.

Figure 6.31 Spiral-bevel helical speed reducer, helical-worm speed reducer, or worm-gear speed reducer, coupled to motor and, with chain drive to drive shaft — is a desirable selection for high reduction ratios in the lower horsepower requirements. This drive is slightly less efficient, but has lower initial costs and is most flexible in terms of location.



- Figure 6.32 — Drive-shaft-mounted speed reducer with V-belt reduction from motor — provides low initial cost, flexibility of location, and the possibility of some speed variation and space savings where large speed reduction ratios are not required and where horsepower requirements are not too large.

Figure 6.32 Drive-shaft-mounted speed reducer with direct drive of V-belt reduction from reducer mounted motor — provides low initial cost, flexibility of location, and the possibility of some speed variation and space savings where large speed reduction ratios are not required and where horsepower requirements are not too large.



- Figure 6.33 — Dual-pulley drive, shown in Figure 6.33, is used where power requirements are very large, and use of heavy drive equipment may be economical by reducing belt tensions.

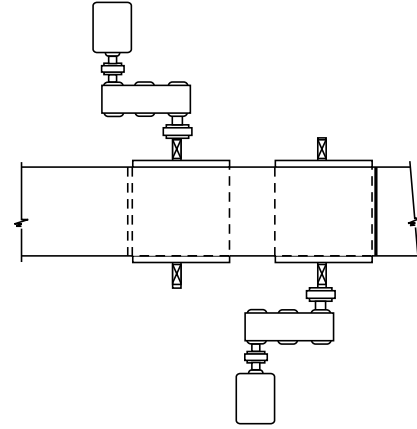


Figure 6.33 Two motors (dual-pulley drive) coupled to helical or herringbone gear speed reducers, directly coupled to drive shafts.

Selection of the type of speed-reduction mechanism can be determined by preference, cost, power limitations, limitations of the speed-reduction mechanism, limitations of available space, or desirability of drive location. The use of speed reducers in the drives for belt conveyors is almost universal today. However, space-saving considerations and low initial cost sometimes may dictate the use of countershaft drives with guarded gear or chain speed reductions.

All of the drives shown can be assembled in either left- or right-hand arrangement.

Drive Efficiencies

To determine the minimum horsepower at the motor, it is necessary to divide the horsepower at the drive shaft by the overall efficiency of the speed reduction machinery.

To determine the overall efficiency, the efficiencies of each unit of the drive train are multiplied together. The final product is the overall efficiency.

The efficiencies of various speed-reduction mechanisms are listed in Table 6-11. These efficiencies represent conservative figures for the various types of drive equipment as they apply to belt conveyor usage. They do not necessarily represent the specific efficiencies of the drive units by themselves. Rather, they take into account the possible unforeseen adverse field conditions involving misalignment, uncertain maintenance, and the effects of temperature changes. While there are some variations in efficiency among different manufacturers' products, the data in Table 6-11 generally cover the efficiencies of the various speed-reduction mechanisms.

As an example of the application of the overall drive efficiency — the result of combining equipment unit efficiencies — consider a belt conveyor drive consisting of a double-helical-gear speed reducer and an open-guarded roller chain on cut sprockets. The approximate overall efficiency, according to Table 6-11, is $(0.94)(0.93) = 0.874$. If the calculated minimum horsepower at the drive shaft is 13.92 hp, then the required motor horsepower is $13.92/0.874 = 15.9$ hp. Therefore, it is necessary to use at least a 20-hp motor.

Mechanical Variable Speed Devices

The most common mechanical methods of obtaining variable speeds of belt conveyors are: V-belt drives on variable-pitch diameter sheaves or pulleys, variable-speed transmissions, and variable-speed hydraulic couplings.

The choice of these devices depends upon the power and torque to be transmitted, the speed range and accuracy of control required, how well the chosen control works into the system, and the relative initial and maintenance costs.

Creeper Drives

In a climate with low temperatures that cause ice to form on the conveyor belt, with resulting loss in conveyor effectiveness, it is good practice to consider the installation of a creeper drive in connection with the drive equipment. The creeper drive can also be used to provide an effective means for inspecting a conveyor. The creeper drive consists of an auxiliary small motor and drive machinery, which, through a clutch arrangement, takes over the driving of the empty conveyor at a very slow speed. This creeper drive is arranged to be operative at all times when the conveyor is not handling any load, thus preventing the formation of harmful ice deposits on the conveyor belt. Creeper drives are normally run at about 10 percent of normal belt speed.

Table 6-12. Mechanical efficiencies of speed reduction mechanisms.

Type of Speed Reduction Mechanism	Approximate Mechanical Efficiency
V-belts and sheaves	0.94
Roller chain and cut sprockets, open guard	0.93
Roller chain and cut sprockets, oil-tight enclosure	0.95
Single reduction helical or herringbone gear speed reducer or gearmotor	0.98
Double reduction helical or herringbone gear speed reducer or gearmotor	0.97
Triple reduction helical or herringbone gear speed reducer or gearmotor	0.95
Double reduction helical gear, shaft-mounted speed reducers	0.97
Spiral bevel-helical speed reducer single, double, or triple reduction	*See note below
Low-ratio (up to 20:1 range) helical-worm speed reducers	0.90
Medium-ratio(20:1 to 60:1 range) helical-worm speed reducers	0.85
High-ratio (60:1 to 100:1 range) helical-worm speed reducers	0.78
Low-ratio (up to 20:1 range) worm-gear speed reducers	0.90
Medium-ratio(20:1 to 60:1 range) worm-gear speed reducers	0.70
High-ratio (60:1 to 100:1 range) worm-gear speed reducers	0.50
Cut gear spurs	0.90
Cast gear spurs	0.85

*Efficiencies for bevel-helical speed reducers are slightly less than the efficiencies shown for helical-gear speed reducers.

Backstops

A loaded inclined belt conveyor of sufficient slope tends to move backwards, when forward motion is stopped by a cessation or interruption of power or a mechanical failure in the driving machinery. Should the loaded belt move backward, the material would pile up at the tail end of the conveyor. This could seriously damage the belt, impose a safety hazard, and result in the need to clean up and dispose of the spilled material. To prevent this reversal of motion, a backstop is used.

A backstop is a mechanical device that allows the conveyor to operate only in the desired direction. It permits free rotation of the drive pulley in the forward direction but automatically prevents rotation of the drive pulley in the opposite direction.

There are three general backstop designs: ratchet and pawl, differential band brake, and over-running clutch. See Figures 6.34, 6.35, and 6.36.

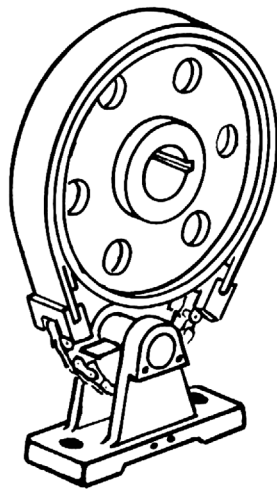


Figure 6.34 Typical differential band brake backstop.

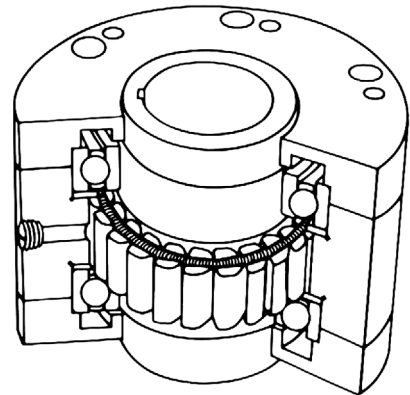


Figure 6.35 Sprag type holdback.

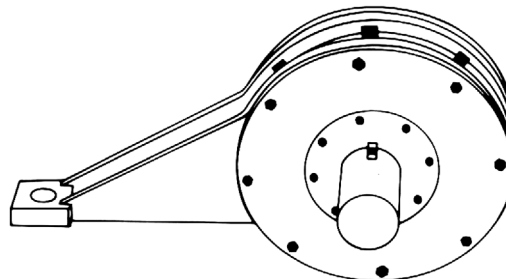


Figure 6.36 Over-running clutch backstop.

Determine Need and Capacity of Backstop, Inclined Conveyors

When the force required to lift the load vertically is greater than one-half the force required to move the belt and load horizontally, a backstop is required. That is, when:

$$HW_m > \frac{L(K_x + K_y W_b + 0.015 W_b) + W_m L K_y}{2}$$

See the formula for T_e , page 87, omitting T_p , T_{am} , and T_{ac} .

Because a backstop is a safety device, it is important that the friction forces that retard the reverse motion of the conveyor are not overestimated. The above formula reduces these friction forces by 50 percent and *eliminates both the temperature correction factor and the friction introduced by the conveyor accessories.*

Backstops are rated on the basis of the pound-ft (lb-ft) of torque they can safely develop. To determine the approximate amount of torque required of a backstop mounted on the drive pulley shaft, the following analysis applies:

r = radius of drive pulley, ft

rpm = revolutions per minute of drive pulley shaft

Torque required of backstop:

$$\text{Torque} = r \left[HW_m - \frac{L(K_x + K_y W_b + 0.015 W_b) + W_m L K_y}{2} \right]$$

$$\text{Torque} = \frac{\text{hp} \times 5,250}{\text{rpm}} \quad \text{and} \quad \text{hp} = \frac{\text{rpm} \times \text{Torque}}{5,250}$$

Horsepower of brake:

$$\text{hp}_b = \left(\frac{\text{rpm} \times r}{5,250} \right) \left[HW_m - \frac{L(K_x + K_y W_b + 0.015 W_b) + W_m L K_y}{2} \right]$$

This analysis applies to straight inclined conveyors. For conveyors with irregular profiles, a special analysis must be made.

Most manufacturers of backstops recommend size selection based on maximum breakdown or stalled torque of motor. Refer to specific manufacturer for his selection procedure.

Brakes

A loaded declined regenerative conveyor, when operating, is restrained from running away by the power source. Any interruption of power or mechanical failure of the drive will permit the belt and load to run out of control. To prevent this, a properly located brake is required.

A horizontal conveyor, or a declined conveyor that is not regenerative, may coast to a degree that is not tolerable. In such cases, a brake is used to regulate the stopping time and distance.

A brake is a friction device for bringing a conveyor belt to a controlled stop. While brakes are used to bring a conveyor to rest in the event of power failure or mechanical drive failure, they also are used to control the coasting distance of a conveyor as it is being decelerated, in order to limit the amount of material that will discharge over the

head pulley during the stopping interval. Brakes are used instead of backstops on inclined reversible conveyors, because backstops are unidirectional.

The brakes used in belt conveyor control operate on the principle that the braking surfaces are engaged by springs and disengaged either by a magnet or by hydraulic pressure induced by an electric motor-hydraulic pump combination. These two types generally are classified by the method of disengaging the braking surfaces. Eddy-current brakes are also used for deceleration.

Practically all conveyors involving lift or lowering need, in addition to the braking force, a holding action after the conveyor has come to a standstill, if for no other reason than safety. In the case of an inclined conveyor, this holding action could be provided by a backstop. However, for any declined conveyor, there is an obvious need for some device that permits application of a controlled torque to decelerate the load at a reasonable reduction in the rate of speed, yet allows sufficient holding power to keep the conveyor belt securely at a standstill when fully loaded but not in operation.

Any conveyor which, under some condition of loading, is regenerative must, for purposes of deceleration analysis and holding power of the brake, be considered as a declined conveyor.

Mechanical Friction Brakes

The mechanical friction brakes are commonly operated electrically. For safety reasons (power failure) such brakes should be spring-set, and power-released.

These mechanical friction brakes provide both the necessary decelerating torque and final holding action. They are interconnected electrically with the motor such that when the power to the motor is off, the holding coil, on the brake also is de-energized, thus allowing a spring to set the brake. For this reason, these brakes are “fail-safe.”

The designer should bear in mind, though, that a friction brake is not a precision device, because of the inherently disadvantageous properties of brake linings. The coefficient of friction of brake lining, and with it the actual braking torque, is affected by temperature, humidity, and the degree to which the lining has become worn.

Eddy-Current Brakes

Eddy-current brakes produce a dynamic braking torque by means of a smooth drum that rotates in a magnetic field produced by a stationary field coil. Eddy currents are generated in the surface of the drum as it rotates. A magnetic attraction between these eddy currents and the poles of the field assembly produces a braking torque on the drum. This torque varies directly with the field current and the speed of the drum. It can be adjusted in a stepless manner by a control system.

For holding action, because the eddy-current brake is not effective in case of power failure, it should be combined with an auxiliary mechanical friction brake. As an eddy-current brake drum slows down, the torque that it is capable of exerting diminishes and is zero when the drum ceases to rotate. Thus, an eddy-current brake cannot be expected to hold a conveyor belt in a standstill position. The auxiliary friction brake also serves to decelerate the conveyor in case of power failure.

Deceleration can also be achieved within the drive motor and its control. There are three basic ways of achieving this braking action, none of which provides holding power after the conveyor belt has come to rest. For this reason, some type of auxiliary external brake is always needed to hold the conveyor belt at a standstill.

Plugging the Motor

Here, the current is reversed and counter torque is developed. This force attempts to rotate the motor in a direction opposite to the existing motion. The energy is dissipated as heat. The motor must be de-energized when zero speed is reached, otherwise the motor will attempt to accelerate in the reverse direction. Among others, squirrel-cage motors are most suitable for this application. There is no holding effect at zero speed and the electrical power losses during plugging are high.

Dynamic Braking

Dynamic braking is a system of electric braking in which motors are used as generators and the kinetic energy of the load is employed as the actuating means of exerting a retarding force. To dynamic brake a.c. motors, it is necessary to provide a source of d.c. excitation during the braking period. The control is so arranged that when the stop button is depressed and the a.c. line contactor is opened, another contactor closes to connect the d.c. excitation to one phase of the motor primary. The motor now acts as a generator and is loaded by the induced current flowing through its squirrel-cage winding. The braking torque, which varies in proportion to the exciting current, rapidly increases as the motor slows down but then decreases at near zero speed. The braking torque disappears near zero speed and there is no holding effect at zero speed.

Regenerative Braking

Squirrel-cage motors operating above synchronous speed have inherent retarding torque characteristics. This retarding condition, known as regenerative braking, is applicable above the synchronous speed of the motor (or for multi-speed motors above their synchronous speeds). The energy generated by the motor flows back into the electric power line. Care must be taken to insure that the electric power system is capable of absorbing the power generated by the motor.

This fundamental type of braking is found to be especially useful for declined conveyors operating at a speed that drives the motor at its synchronous speed, plus slip.

Brakes and Backstops in Combination

Often a brake is used to control the stopping interval on an inclined conveyor. If the conveyor is a large and important one, which may reverse and run backward in the event of a mechanical failure, prudence dictates the use of a mechanical backstop as a safety precaution, in addition to the electrically operated brake.

Friction surfaces on brakes, and brakes used as backstops, do not develop the design friction factors until the braking surfaces have worn in to effect full contact. Therefore, friction brakes used as such or as backstops must be adjusted to compensate for this "wearing in" process.

Restraint of Declined Conveyors

Declined conveyors of the regenerative type, are restrained in normal operation by the drive motor which acts as a generator, when the belt and its load force the motor to rotate faster than its synchronous speed. The motor may fail to restrain the belt and load when it is forced to a speed where its current output is excessive and the overload protective device breaks the circuit. Proper selection of the motor and controls will avoid this contingency. Nevertheless, a brake must be supplied, one which will set when the power circuit is broken.

A centrifugal switch is often used on declined belt conveyors to open the electrical control circuit at a predetermined overspeed, and thus to set the brake. This will act as a safety against mechanical failure in the drive machinery.

A brake is usually located at the tail end of a declined conveyor.

Table 6-13. Backstop and brake recommendations.

Type of Conveyor	Backstop	Brake	Forces to be Controlled
Level or horizontal conveyor	Not required	Required when coasting of belt and load is not allowable or needs to be controlled	Decelerating force minus resisting friction forces
Inclined conveyor	Required if hp of lift equals or exceeds hp of friction	Not usually required unless preferred over backstop	Incline load tension minus resisting friction forces
Declined conveyor	Not required	Required	Decelerating force plus incline load tension minus resisting friction

Backstop and Brake Recommendations

Table 6-13 lists recommendations for the use of backstops and brakes on horizontal, inclined, and declined conveyors.

Deceleration by Brakes

Brakes are a necessity on declined conveyors so that the loaded belt may be stopped without excessive or runaway coasting. Brakes are also applied to horizontal and inclined belt conveyors for the same reason. Excessive coasting may discharge far more material than the succeeding conveyor or other units can handle. Mathematical calculation and the careful selection of a properly sized brake will eliminate such difficulties.

Devices for Acceleration, Deceleration, and Torque Control

Starting the Conveyor

Smooth starting of a conveyor belt is important. It can be accomplished by the use of torque-control equipment, either mechanical or electrical, or a combination of the two. The belt conveyor designer should investigate acceleration stresses of conveyor components to insure that the overall stresses remain within safe limits.

Smooth starting can be an important consideration, where excess horsepower may have been installed to provide for future increased capacity or for future extensions of the conveyor. In cases of conveyors having vertical curves or trippers, too rapid a start may cause excessive lifting of the belt from the idlers. This would necessitate a provision for gradual acceleration of the conveyor belt.

Controlled Acceleration

Acceleration can be controlled by several types of electrical devices.

Wound-Rotor Motors with Step Starting

By the addition of external resistance in the secondary winding, electrically accessible through slip rings, starting torque can be controlled by planned steps. This allows a program designed to suit the particular conveyor, and overcome the problems of excessive belt tension, shape of the vertical curves, and other problems that are solved by starting time control.

This type of electrical control device has been widely used for many years on large belt conveyor systems.

Squirrel-Cage Induction Motor with Autotransformer

Another method of controlling the torque, and with it the acceleration time, is the use of an induction motor (normal or high-torque) with autotransformer starting. Its use must be checked because the low-starting torque caused by the reduced voltage may not be enough to overcome the breakaway static friction in level or inclined conveyors.

Eddy-Current Couplings

These are electromagnetic devices composed of three basic parts: a rotor made up of multiple pole pieces (and secured to one shaft), a hollow iron cylinder or drum that surrounds the rotor (and is secured to the other shaft), and a stationary electromagnetic coil that surrounds both rotor and drum and provides the magnetic field in which they operate.

The electromagnetic coil is energized by a low-power, direct current supply. When either the rotor or the drum is rotated, eddy currents are induced. These eddy currents set up a secondary field and thus create a torque between the rotor and drum. The driven or output member never attains the same speed as the driving or input member. This inherent difference in speed is called "slip." The slip loss appears as heat, which must be dissipated by air or water cooling.

In a conveyor drive, the eddy-current coupling is placed between the squirrel cage motor and speed reducer, on the motor shaft, and on the speed reducer input shaft. Because the degree of excitation of the coil determines the slip between the driving and driven members, it is obvious that eddy current couplings provide an ideal means of controlled acceleration. Excitation of the coil can be increased over a definite time period, or it can be made responsive to tachometer feedback speed-regulating control. Sophisticated electronic control can be employed to regulate the coil excitation to produce virtually any desired result.

There are several advantages of eddy-current couplings. (1) They require low-power coil excitation. (2) They permit smooth, controlled starting. (3) The motor can be started and accelerated without connecting the load. On frequent start and stop applications the motor can run continuously. (4) Variable speed can be obtained. However, in variable speed applications the additional slip creates more heat that must, of course, be dissipated. (5) They make possible the use of squirrel-cage motors and across-the-line starters. (6) A modified eddy-current coupling can be used as a decelerating brake (not as a "holding brake," however).

The disadvantages of eddy-current couplings are: (1) They require additional drive space. (2) Water cooling must be provided for the larger sizes. (3) Generally, they are more expensive than a wound-rotor motor and reduced-voltage starting.

Fluid Couplings

These are mechanical two-piece mechanisms consisting of an impeller and a runner, both within a housing filled with oil. The impeller is connected to the driving shaft; the runner is connected to the driven shaft. In conveyor drives, the fluid coupling usually is placed between the motor and speed reducer. When the impeller is revolved, the oil is accelerated to the periphery and passes to the blades of the runner, producing a torque on the runner proportional to the weight and rate of fluid flow. The fluid coupling basically is a slip clutch. And, as with an eddy current coupling, the slip loss appears as heat. Unlike its electrical counterpart, a fluid coupling is not used as a variable-speed device.

When properly applied, a fluid coupling can produce reasonably smooth acceleration of high-inertia loads. The motor speed rises rapidly to a point near the maximum torque condition before the load is engaged. This makes the standard squirrel-cage motor an ideal driver, as its peak torque is about 200 percent of full-load torque. The fluid coupling permits the use of squirrel-cage motors with across-the-line starters.

Fluid couplings also provide an excellent solution to voltage drop problems by allowing the motor to achieve full speed before it experiences load. In addition, fluid couplings limit belt forces imparted due to higher voltage at start-up, thus preventing belt slippage.

Variable-Speed Hydraulic Couplings

These devices have been used very successfully worldwide. The variable-speed hydraulic coupling consists of a fluid coupling with input and output shafts, heat exchanger, charging oil pump, and associated control. The amount of oil in the coupling is variable by the position of an adjustable scoop tube. The control can be manual or completely automatic. Speed variations over a 4:1 range are possible. The device allows the AC motor to start under no load. The torque buildup starts at zero and the buildup can be controlled over an extended time. Smooth acceleration times up to 200 seconds are achievable. The device can also act as a clutch without AC motor shutdown.

Dry Fluid Couplings

These are similar to oil-filled fluid couplings, except that they consist of a housing that is keyed to the motor shaft and a rotor that is connected to the load. The housing contains a charge of steel shot instead of a fluid. When the motor is started, centrifugal force throws the charge of steel shot to the inner periphery of the housing, where it packs around the rotor. Some slippage takes place before the housing and rotor are finally locked together. Power thus is transmitted from the motor to the load.

The amount of the charge of steel shot determines the torque during acceleration. It can also determine the torque-limiting feature of this coupling.

Miscellaneous Fluid Couplings

These are similar to dry fluid couplings except, in lieu of a charge of steel shot, these fluid couplings employ silicone fluid or other environmentally suitable liquids. Characteristics of these couplings are such that manufacturers should be consulted for specific performance details before an application is made.

Centrifugal Clutch Couplings

These consist of a driving hub, a driven sleeve or drum, and a series of shoes connected to or driven by the hub. The periphery of each shoe is provided with a brake-lining material. The hub is carried by the driving shaft, the drum by the driven shaft.

Brake Requirement Determination (Deceleration Calculations)

When the hub rotates, centrifugal force impels the shoes outward against the inside of the drum, to transmit power to the load. Slippage occurs, which produces the effect of smooth starting.

Flywheel

Mechanical control of starting and stopping can be accomplished by means of a flywheel, which adds to the WK2 of the prime mover, thus increasing the starting time and limiting the torque input to the belt conveyor system, as well as increasing stopping time and distance.

Mechanical Clutches

This device can effectively control starting torque; it allows adjustment of the amount of torque, as well as the rate at which it is applied. The mechanical clutch can be preset for both rate of application and maximum limit.

Brake Requirement Determination (Deceleration Calculations)

In order to determine whether any braking action, other than the friction forces inherent in the system, is required, several different circumstances under which the conveyor might be stopped have to be considered. For instance, is the stopping intentional, or is it the result of power failure? Also, if another conveyor feeds onto it, or if the conveyor in question delivers its load to an additional belt, it is necessary to consider their respective motions and deceleration cycles.

It is obvious that the drives of a conveyor system that consists of more than one belt, and in which at least one belt feeds into another, have to be interconnected electrically in such a way that if one conveyor is stopped for any reason, the one feeding onto it is also stopped. This precaution alone does not suffice, however, if the physical properties of the first conveyor are such that it would coast longer than the second one. If this were to occur, it would result in a pile-up of material on the second belt and could cause a hazardous situation.

Generally speaking, in any system with more than one conveyor, the length of the deceleration cycle of any successive conveyor should be equal to or more than that of the preceding one.

If the inherent properties of the various units do not result in deceleration cycles that agree with this basic rule, two remedies are possible. (1) A brake can be applied to those conveyors that coast too long. This is a straightforward solution, and relatively easy to accomplish. (2) The stored energy of those conveyors that come to a stop too quickly can be augmented, for instance, by a flywheel. Although a flywheel will lengthen the stopping distance of a conveyor, it will also increase its acceleration time. This must be taken into consideration by the belt conveyor designer.

However, in most cases, the application of a brake will be found more convenient, unless its use overstresses any member of the unit to which it is applied.

Material Discharged During Braking Interval

To determine the amount of material discharged during the braking interval, it must be assumed that the conveyor decelerates at a constant rate. Therefore, the distance travelled, while stopping from full speed, is the average velocity multiplied by the time of braking interval.

$$\text{Distance, ft, conveyor travels} = \left(\frac{V + 0}{2} \right) \left(\frac{t_d}{60} \right) = \frac{V t_d}{120}$$

where :

V = belt speed, fpm
 t_d = actual stopping time, seconds

If the amount of material that can be safely discharged to the succeeding conveyor (or other) unit is known, the maximum length of time of the braking or decelerating interval can be determined as follows:

$$W_d = \frac{V t_m}{120} (W_m)$$

Therefore:

$$t_m = \frac{120 W_d}{W_m V}$$

where :

t_m = maximum permissible stopping time, seconds (braking or deceleration interval)
 W_d = weight, lbs, which can be discharged
 W_m = weight of material, lbs/ft of belt

Forces Acting During Braking or Deceleration

The forces that act on the conveyor during a braked stop (deceleration) include inertia; frictional resistance; gravity material load force; inclines or declines; and braking force.

The frictional resistance forces and the gravity material load forces, if any, are equal to T_e . The braking force is equal to the algebraic sum of the other forces.

Therefore, for horizontal, inclined, and non-regenerative declined belt conveyors, the braking force = inertial forces + T_e or:

$$F_d = \frac{M_e V}{60 t_m} - T_e = \frac{W_e V}{60 g t_m} - T_e$$

For regenerative declined belt conveyors, braking force = inertial forces + T_e , or:

$$F_d = \frac{M_e V}{60 t_m} + T_e = \frac{W_e V}{60 g t_m} + T_e$$

where

F_d = braking force, lbs, at belt line
 M_e = equivalent moving mass, slugs
 g = acceleration of gravity, 32.2 ft/sec²
 W_e = equivalent weight of moving parts of the conveyor and its load, lbs. See Problems 3 and 4, pages 153-161 and 161-172, respectively.
 V = speed of belt, fpm
 t_m = maximum permissible stopping time, seconds (braking or deceleration interval)
 T_e = effective or driving horsepower tension, lbs

Brake Location

An analysis of the belt tension diagram during deceleration should be made to determine the appropriate pulley on which to apply the brake. The braking force will be additional to the friction and positive lift forces.

If the brake is installed on a head-end drive pulley, the automatic takeup force must be sufficient to transmit the braking force through the takeup. The wrap factor at the braking pulley must be checked for adequacy during braking. Also, the minimum belt tension in the carrying run of the conveyor must be maintained during braking. The maximum permissible belt tension must not be exceeded during deceleration.

For inclined or short horizontal conveyors, it may be possible to brake through the head or drive pulley, providing the takeup has sufficient force to absorb the braking force and still maintain a slack-side tension to meet the wrap factor requirements. If this is not practical, as in the case of a long horizontal or declined conveyor, then the braking force must be applied to the tail pulley.

The maximum belt tension during deceleration should be calculated to insure that it does not exceed the recommended allowable starting (or braking) tension. (See page 113, and Chapter 13, "Controlled Deceleration.") If it is found that the belt tension does exceed the allowable amount, a heavier belt may be required. Or the belt conveyor can be reanalyzed to provide for a smaller braking force acting over a longer time period. If the conveyor is subjected to frequent stops, the pulleys and shafts must be selected for the higher tensions introduced during deceleration.

Braking Torque

The braking force (lbs) determined above and acting at the belt, multiplied by the radius (ft) of the braked conveyor pulley, gives the required torque rating of the brake (lb-ft), provided the brake is installed on the same shaft that carries the braking pulley.

$$\text{Torque} = F_d r$$

where:

F_d = braking force at the belt

r = radius of conveyor pulley, ft, on the same shaft as the brake

If the brake is to be installed on some shaft other than the pulley shaft, the torque requirement is converted by multiplying the above torque by the revolutions/minute of the shaft for which the torque was determined. This product is then divided by the revolutions/minute of the shaft on which the brake will be mounted. Select the brake with the next higher torque rating.

Brake Heat Absorption Capacity

The discussions above relate to the selection of a brake on the basis of torque only. Stopping a moving mass involves the absorption of the kinetic energy of the belt, the load, and the moving machinery. This energy only can be dissipated in the form of heat at the brake. The resulting temperature rise of the brake elements must not damage the brake. For this reason, a discussion of brake design and brake heat absorption follows.

Industrial brake linings usually are made of either woven or molded material, plus various fillers and adhesives. The coefficient of friction of these linings against a brake wheel varies considerably with different ambient conditions. Because of the

nature of these variations, definite values of the friction coefficient cannot be given. Nevertheless, some variations that can be expected are approximated below.

Coefficients, and consequently torque values, may vary widely for new linings and/or new wheels, until both lining and wheel surfaces are worn in. This requires approximately 4,000 to 6,000 full-torque brake-setting operations. During this period static torque may drop as much as 30% below the initial setting, and dynamic torque as much as 50%. For this reason, the discussion will relate only to well-worn-in linings and wheels.

Both static and dynamic torque vary with wheel surface temperatures. At 50°C to 75°C, the static torque may be as much as 30 percent to 35 percent high. But it then drops off rather rapidly with increase in wheel temperature. At 115°C to 135°C, the static torque is about normal. At 150°C, it may be 5% to 7% below normal. The dynamic torque may be 10 percent to 15 percent high at 40°C to 60°C, and then rise rapidly, until at 115°C to 150°C it may be as high as 140 percent. It then drops off rather rapidly with further temperature rise.

Because of these variations, brake wheels are rated at 120°C rise for normal energy dissipations. The ratings, which are expressed in “hp seconds,” are based on a maximum temperature rise of 120°C at the brake wheel, when the brake is applied at the listed time intervals. The brakes have lower ratings for more frequent stops because they will not cool sufficiently between stops to absorb the heat of rapidly repeated stopping.

Humidity will also have an adverse effect on braking torque because industrial brake linings absorb moisture. If a brake is allowed to stand inoperative for some time in high ambient humidity, the braking torque may be reduced as much as 30% when the brake is first set. This condition is self-correcting, because the heat generated in braking rapidly drives off the moisture. Usually the torque will be restored to almost normal at the end of the first braking cycle. In this case, the only effect is a longer time than usual to make the first stop.

Variations in any given lining material, and in surface conditions of the lining and wheel, may result in a 10 percent plus or minus variation in torque during successive stops.

From the above factors, it is evident that industrial brakes are not precision devices. The normal method of setting brake torque by measuring either spring length or adjusting-bolt length is at best an approximation. Where braking effects are important to a conveyor operation, the brake should be readjusted for optimum braking by actually stopping and holding the load after the brake first is installed. For critical conveyor applications, it may be necessary to readjust the brake more than one time during the break-in period for new linings.

Brake Calculations

To check the brake wheel heat absorption for a single stop of a loaded conveyor, first determine the actual stopping time for the brake selected.

$$t_d = \frac{W_e V}{(32.2)(60)} \div \left(\frac{Z_b}{r} \right) \left(\frac{\text{rpm}_b}{\text{rpm}_p} \right) + T_e$$

where:

- t_d = actual stopping time, seconds
- W_e = equivalent weight of the moving mass, lbs
- V = velocity or speed of belt, fpm
- Z_b = torque rating or setting of brake, lb-ft
- rpm_b = revolutions/minute of brake shaft
- rpm_p = revolutions/minute of drive pulley shaft

NOTE: For regenerative belt conveyors, T_e may be negative.

The energy that must be absorbed by the brake when making a single stop of a loaded conveyor is expressed as follows:

$$\text{Energy, in horsepower seconds} = \frac{(Z_b)(\text{rpm}_b)(t_d)}{10,500}$$

The symbols are the same as above, but with the brake on the drive pulley shaft, $\text{rpm}_p = \text{rpm}_b$.

The heat absorption should be approved by the brake manufacturer for the anticipated duty cycle.

If the brake selected does not have the heat-absorption capacity required, either a modified or larger brake with the necessary heat-absorption capacity should be used. The spring should be adjusted to the torque desired.

Example

As a numerical example of brake selection, the belt conveyor specifications for Problem 3, page 153, will be used. Since the WK^2 and total equivalent weight for the conveyor have been calculated, only the essential portions of these specifications are repeated here.

Conveyor Specifications:

V	=	belt speed = 500 fpm	
W_m	=	weight of material/ft of belt	= 226 lbs
T_e	=	effective tension	= 16,405 lbs
T_2	=	slack-side tension	= 5,720 lbs
T_0	=	minimum tension	= 3,067 lbs
T_t	=	tail tension	= 7,054 lbs
Equivalent weight of conveyor moving parts		=	162,696 lbs
Weight of material load		=	543,360 lbs
Total equivalent weight for belt tension determination		=	706,056 lbs
Drive equivalent at belt		=	<u>62,870</u> lbs
W_e = total equivalent weight for brake determination		=	768,926 lbs

Assuming that the conveyor discharges into a hopper that holds 9,000 lbs of material, the maximum permissible stopping time is as follows:

$$t_m = \frac{120 W_d}{W_m V} = \frac{(120)(9,000)}{(226)(500)} = 9.54 \text{ seconds}$$

Brake force at belt line is:

$$F_d = \frac{W_e V}{60 g t_m} - T_e = \frac{(768,926)(500)}{(60)(32.2)(9.54)} - 16,342 = 4,517 \text{ lbs}$$

Analysis:

If the total equivalent retarding force of 4,517 lbs is applied to the head-end drive shaft, a proportion equal to:

$$\frac{(62,870)(500)}{(60)(32.2)(9.54)} = 1,706 \text{ lbs}$$

of the equivalent force would be absorbed in retarding the drive components. The remainder, $4,517 - 1,706 = 2,811$ lbs of the equivalent force, would be transmitted to the belt by the pulley to retard the conveyor moving parts and the load. This force is T_{eb} .

During braking, the highest tension in the belt will be T_{1b} , on the return run just past the drive pulley. If the automatic takeup is to be on the verge of yielding to the braking force, T_{1b} can be assumed to be equal to T_2 , the slack-side tension during normal operation of the belt. Because:

$$T_{1b} - T_{2b} = T_{eb}$$

and substituting T_2 for T_{1b} :

$$T_2 - T_{2b} = T_{eb} = 5,720 - T_{2b} = 2,811$$

Therefore:

$$T_{2b} = 5,720 - 2,811 = 2,909 \text{ lbs}$$

This is the tension in the carrying run at the head-end drive pulley during braking. It is insufficient, for the minimum tension, $T_0 = 3,067$ lbs. Also, $T_{2b} = (C_{wb})(T_{eb})$, or:

$$C_{wb} = \frac{T_{2b}}{T_{eb}} = \frac{2,909}{2,811} = 1.03 \text{ wrap factor during braking}$$

This is sufficient, as the wrap factor for the drive is 0.35 to prevent slip between the pulley and belt.

When it is shown that braking at the head-end drive produces too low a tension in the carrying run, or too small a wrap factor, it is necessary to increase the belt tensions by increasing the automatic takeup force.

The alternative to braking at the head-end drive pulley is to apply the braking force to the tail pulley. In this case, the entire braking force of 4,517 lbs must be transmitted to the belt.

When the power of the drive is cut off, and just as the brake takes effect, the tension in the return run at the tail pulley will be:

$$T_{2b} = T_2 + \text{pulley friction} + \text{return-belt idler friction} \\ - \text{inertia of moving parts of the return run}$$

$$\text{Pulley friction} = (4)(100) + (1)(150) = 550 \text{ lbs}$$

$$\begin{aligned} \text{Return idler friction} &= L(0.015 W_b) \\ &= (2,400)(0.015)(17) = 612 \text{ lbs} \end{aligned}$$

Equivalent weights of the moving parts of the return run are:

$$\text{Return belt, } LW_b = (2,400)(17) = 40,800 \text{ lbs}$$

Return idler weight of rotating parts is 48.4 lbs, from Table 5-14, for a 48-in.-wide belt and Class C6 idlers.

$$\text{Total return idler rotating weight} = \left(\frac{2,400}{10} \right) (48.4) = 11,616 \text{ lbs}$$

Rotating weight of pulleys, from Problem 3, page 156, is 3,450 lbs

$$\text{Total equivalent moving parts of return run} = 40,800 + 11,616 + 3,450 = 55,866 \text{ lbs}$$

Equivalent force at the belt line

$$\begin{aligned} &= \frac{W_e V}{60 g t_m} \\ &= \frac{(55,866)(500)}{(60)(32.2)(9.54)} \\ &= 1,516 \text{ lbs} \end{aligned}$$

Therefore:

$$T_{2b} = 5,720 + 550 + 612 - 1,516 = 5,366 \text{ lbs}$$

And because $T_{eb} = 4,517$, and $C_{wb} = T_{2b}/T_{eb}$, then

$$C_{wb} = \frac{5,366}{4,517} = 1.19$$

This is very satisfactory, since a 180-degree wrap, bare pulley requires only that the wrap factor, C_{wb} , be 0.84 or larger. See Table 6-8. The maximum belt tension when braking = $4,517 + 5,366 = 9,883$ lbs. This is well within the maximum of $1.80(25,920) - 9,883 = 36,773$ lbs (see page 157). It therefore is appropriate to place the brake on the tail pulley shaft of this conveyor.

Assuming the tail pulley radius is 1.5 ft and the pulley is revolving at 53 rpm, the torque at the tail pulley shaft is:

$$F_d r = (4,517)(1.5) = 6,776 \text{ lb-ft}$$

As the brake will be mounted directly on the tail pulley shaft, the required brake torque will be 6,776 lb-ft.

For this problem, assuming alternating current electric power is available, the brake selected from the brake manufacturers' catalog is an a.c. magnetic brake with a 10,000 lb-ft rating. This is the next larger size than that calculated at 6,776 lb-ft.

Actual stopping time, using this 10,000 lb-ft brake is:

$$\begin{aligned} t_d &= \frac{\frac{W_e V}{(32.2)(60)}}{\left(\frac{Z_b}{r}\right)\left(\frac{rpm_b}{rpm_p}\right) + T_e} \\ &= \frac{\frac{(768,926)(500)}{(32.2)(60)}}{\left(\frac{10,000}{1.5}\right)\left(\frac{53}{53}\right) + 16,342} \\ &= 8.65 \text{ seconds} \end{aligned}$$

This is less than the maximum of 9.54 seconds that is permissible.

Energy absorbed is:

$$p = \frac{(Z_b)(rpm_p)(T_d)}{10,500} = \frac{(10,500)(53)(8.65)}{10,500} = 432.62 \text{ hp seconds}$$

The brake selected is capable of heat absorption of 3,400 hp-seconds every 15 minutes. This indicates that a loaded stop can be made safely without overheating the brake.

Factors in the Composition of Conveyor Belting
Conveyor Belt Cover Characteristics, Composition, and Design
Loading Considerations
Belt and System Considerations
Conveyor Belt Selection

The purpose of this chapter is to familiarize the user with the general requirements and method of proper selection of conveyor belting. Because of the multitude of conveyor belt constructions presently available and the impossibility of treating them all in this chapter, only those basic types and grades of conveyor belting that apply to a majority of conveyor applications will be covered.

Anyone using the data in this chapter should recognize that a belt selection determined by the data will be conservative. While the selected belt will meet the specified conditions, it may not always be the most economical construction available. This is particularly true because of the continuing developments in the fields of elastomers and synthetic fibers for use in conveyor belts.

For major conveyor belt applications, the complete duty and operating requirements of the conveyor must be analyzed to develop final specifications.

A complete analysis of a conveyor system for determining the conveyor belt specifications requires consideration of the design details which follow.

1. Material conveyed: general description; density (pounds per cubic foot, lbs/ft³); maximum lump size; presence of oils or chemicals, if any; maximum temperature of load, if hot; requirements for fire resistance.
2. Maximum loading rate or required maximum capacity, short tons (2000 lbs) per hour (tph).
3. Belt width, inches.
4. Belt speed, feet per minute (fpm).
5. Profile of conveyor: distance along conveyor path, tail to head, feet; lift or drop, $\pm$ feet, *or* elevations of top and bottom of any inclines or declines; angles of slope of all inclines or declines; locations and radii of all vertical curves.
6. Drive: single-pulley or two-pulley; if dual drive, distribution of total motor horsepower at primary and secondary drive pulleys; angle of belt wrap on drive pulley(s); location of drive; pulley surface, bare or lagged; type of lagging; type of starting to be employed.
7. Pulley diameters: These should be checked against the actual belt specification.
8. Takeup: type, location, and amount of travel.
9. Idlers: type, roll diameter, angle of trough; spacing, including transition distance at head and tail.
10. Type of loading arrangement: numbers of chutes; lumps-to-belt freefall distance; skirtboard length; impact idlers or bed, if any; angle of loading to belt.
11. Lowest cold weather operating temperature anticipated, if applicable.

12. Type of belt splice to be used: vulcanized or mechanical.
13. Type(s) of belt cleaners to be used: vulcanized or mechanical.

Factors in the Composition of Conveyor Belting

While a belt conveyor system is composed of many important parts, none is more economically important than the conveyor belt itself, which, in most cases, will represent a substantial part of the initial cost. Therefore, the selection of the conveyor belt must be made with great care.

In general, a conveyor belt consists of three elements: top cover, carcass, and bottom cover. Figure 7.1 illustrates a cross section of a typical belt. The primary purpose of the covers is to protect the belt carcass against damage and any special deteriorating factors that may be present in the operating environment. The belt carcass carries the tension forces necessary in starting and moving the loaded belt, absorbs the impact energy of material loading, and provides the necessary stability for proper alignment and load support over idlers under all conditions of loading.

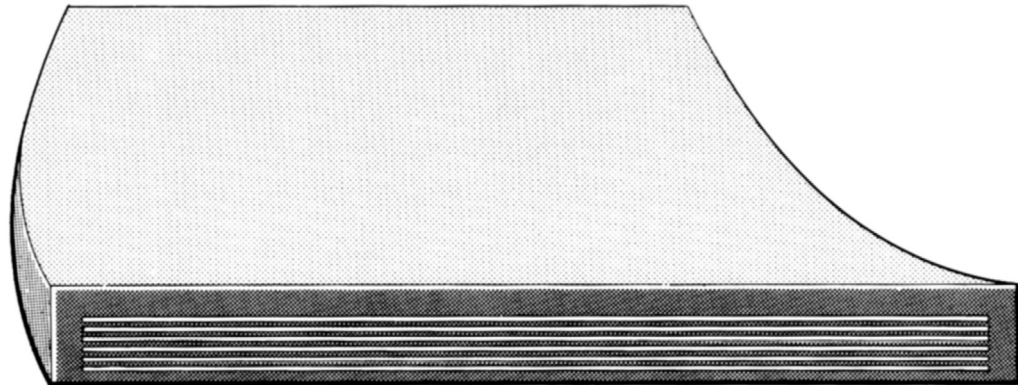


Figure 7.1 Cross section of a fabric-reinforced belt (Cut/slit edge).

Although covers and carcass are treated as separate components, successful operation depends upon their working together as a single integral unit to provide the necessary belt characteristics.

Conveyor Belt Covers: Characteristics, Composition, and Design

Conveyor belting today is available in a wide range of types and constructions. Varieties of conveyor belting are suitable to fit an extraordinary range of conveyed material, operating conditions and design criteria.

The development of a broad range of synthetic rubbers, polymers, elastomers, and fibers over the past forty years has increased the service life and extended the operational envelope of conveyor belting. Previously, natural rubber covers and cotton fiber reinforcement (carcasses) were the only options.

Today, conveyor belt covers will consist of compounds comprised of natural rubbers, styrene-butadiene rubber (SBR) blends of natural and other synthetics, nitriles, butyl, ethylene propylene-based polymer (EPDM), polychloroprene (neoprene), polybutadiene, polyvinyl chloride (PVC), urethanes and silicones, etc., and the list goes on and continues to grow. Each of these elastomers has specific usefulness for various ranges of properties and operating conditions from which manufacturers and end-users can choose.

Conveyor belting and its corresponding cover composition(s) can be designated as either (1) general purpose belting, or (2) special purpose belting. Each of these two broadly classified groups should be further defined depending upon the specific end use.

General Purpose Belting

General purpose covers and belting serve a broad range of industrial applications including mining, ore processing, lumber, paper/pulp, and agriculture, to name a few. By and large, these belts will have covers of either natural rubber, SBR, polybutadiene, and acrylonitrile or blends thereof. These cover compounds are further defined by the Rubber Manufacturers Association (RMA) and belting industry as either Grade I or Grade II.

General Purpose Rubber Covers

RMA Grade I — Will consist of natural or synthetic rubber or blends which will be characterized by high cut, gouge, and tear resistance and very good to excellent abrasion resistance. These covers are recommended for service involving sharp and abrasive materials, and for severe impact loading conditions.

RMA Grade II — The elastomeric composition will be similar to that of Grade I with good to excellent abrasion resistance in applications involving the conveyance of abrasive materials, but may not provide the degree of cut and gouge resistance of Grade I covers.

When covers are tested in accordance with ASTM D412, the tensile strength and elongation at break shall comply with the requirements of Table 7-1, for the grade of cover, as appropriate.

Table 7-1. Properties of covers.

Grade	Minimum Tensile Strength (p.s.i.)	Minimum Tensile Strength (MPa)	Minimum Elongation @ Break (%)
I	2500	17	400
II	2000	14	400

The tensile strength and elongation at break values are not always sufficient in themselves to determine the suitability of the belt cover for a particular service. The values in the above table should only be specified for conveyors or materials with a known history of performance and where it is known that compliance with the value will not adversely affect other in-service properties.

Cover and ply adhesion

When belting is tested in accordance with ASTM D378, the adhesion for covers and between adjacent plies should not be less than the values given in Table 7-2.

Table 7-2. General purpose rubber cover and ply adhesion.

Adhesion between Adjacent Plies		Adhesion between Cover and Ply			
30 lbs/in.	4 KN/m	Cover Thickness 1/32" (0.8 mm) through 1/16" (1.6 mm)		Covers Greater than 1/16" (1.6 mm)	
		20 lbs/in.	3 KN/m	30 lbs/in.	4.4 KN/m

Special Purpose Belting

Special purpose belting and its components (covers) are just that: those that require special characteristics and properties. Conveyor applications and systems that operate outside the normal parameters covered under general purposes will include high temperatures (above 175° F/80° C), low temperatures, (below 40°F/5°C), fire/flame resistance, oil exposure, food ("FDA") processing, and chemical resistance. Conveyor belt manufacturers provide products to meet these and other demands with a wide variety of elastomers and carcass constructions. The following list of conveyor cover compound types is not all inclusive, but is a general guide for special applications:

Hot Materials Handling

Cover compounds consisting of butyl (and bromo/chloro butyl) or EPDM can resist the degrading effects of high temperatures up to approximately 400°F/200°C. Some specially formulated SBR-based compounds will perform in high temperature environments but not generally to the same range or degree as EPDM or butyl-based covers. Neoprene (polychloroprene) and Hypalon (chloro-sulfonated polyethylene) based compounds also exhibit good heat aging properties. Belting with silicone or Viton (fluorocarbon polymers) covers will withstand very high temperatures best, with extended operating range up to approximately 700°F.

NOTE: Polyester and nylon fibers/textiles will melt at temperatures above 500°F/260°C. Loss of dimensional stability and softening will occur well before this temperature is reached. Glass fiber carcasses are often recommended where operating temperatures exceed 400°F/200°C.

Oil Resistant Belting

Belt covers designed to resist swelling and degradation in oily environments will often incorporate nitrite based polymer, polyvinyl chloride (PVC), or urethane. The type of oil encountered as well as the temperatures in which the belt must operate is of prime importance. Highly aromatic and asphaltene-based materials, as well as exposure to diesel fuel, are best handled with a nitrite- or urethane-based compound. PVC belting will resist light oil (e. g., mineral and naphthenic oils) degradation at lower temperatures. Neoprene/polychloroprene compounds will also resist low aromatic oils and fuels satisfactorily.

Food Processing

Food processing entails belt exposure to both vegetable oil and animal fats. In such environments, PVC and nitrite-based belt constructions predominate. Both have good resistance to swelling and degradation under these conditions.

Belt Selection

Fire/Flame Resistance Belting requiring flame resistance is engineered to meet underground mining regulations and specifications. Currently, belt and belt compounds using SBR, nitrile, polychloroprene (neoprene), and PVC are routinely utilized.

Low Temperature Environments Generally, most general purpose (Grades I and II) belting and compounds will resist stiffening down to -40°F/°C. For most general purpose belting, when there are prolonged periods of downtime during which the belt is exposed to -40° for several days or weeks, hard starts may be difficult or deleterious to the belt because of cold-set. When these conditions are expected, belts can be obtained which have suitable low temperature plasticizers and low glass-transition polymers or blends incorporated to permit maximum flexibility and operation.

Chemical Exposure Conveyor belting manufacturers should be consulted when systems are being operated in specific chemical environments. The condition in which the conveyor belt is operating should be clearly defined. Consideration of the chemical concentration and temperature, as well as the possible presence of incidental processing chemicals or oils should also be taken into account.

Table 7-3. Conveyor belt cover quality selection.

Cover Grade	Major Advantages			General Applications
	Cut & Tear Resistance	Abrasion Resistance	Oil Resistance	
General Service				
Grade I	Excellent	Excellent	Not recommended	Large size ore, sharp cutting materials. For extremely rugged service.
Grade II	Good	Good to Excellent	Not recommended	Sized materials with limited cutting action--primarily abrasion. For heavy duty service.
Oil and Chemical Service				
Chloroprene (Neoprene) oil resistant	Good	Very Good	Very good for petroleum oils. Fair for vegetable and animal oils.	Heavily oil sprayed coal (petroleum oil up to 20% aromatics, No. 2 diesel fuel). Any material treated with or containing large amounts of petroleum oil.
Buna N Nitrile oil resistant	Good	Good	Very good for petroleum, vegetable, and animal oils.	Oily grain or seed service (soybeans, crushed corn, etc.) Food handling. Greasy, oil-sprayed coal (petroleum oil up to 40% aromatics. No. 2 heating oil.)
Medium oil resistant	Good	Good	Limited for petroleum, vegetable, and animal oils.	Lightly sprayed coal, mildly oily grains and feeds, wood chips, phosphates.

Cover Considerations The covers should be of sufficient thickness and quality to protect the carcass. Covers for general service applications are listed in Tables 7-4 and 7-5, which list suggested minimum thickness for carrying and pulley side covers, respectively.

The cover gauge required for a specific belt is a function of the material conveyed and the handling methods used. Increased cover thickness is required as the following conditions become more severe: material abrasiveness, maximum material lump size, material weight, height of material drop onto the belt, loading angle, belt speed, and frequency of loading as determined by the frequency factor.

Table 7-4. Suggested minimum carry thickness for normal conditions: RMA—Grade II belting.

Class of Material	Examples	Thickness (inches)*
Package handling	Cartons, food products	Friction to 1/32
Light or fine, nonabrasive	Wood chips, pulp, grain, bituminous coal, potash ore	1/16 to 1/8
Fine and abrasive	Sharp sand, clinker	1/8 to 3/16
Heavy, crushed to 3" (76 mm)	Sand and gravel, crushed stone	1/8 to 3/16
Heavy, crushed to 8" (203 mm)	R.O.M. coal, rock, ores	3/16 to 1/4
Heavy, large lumps	Hard ores, slag	1/4 to 5/16

*Note: Cover thicknesses are nominal values subject to manufacturers' tolerances.

Table 7-5. Suggested minimum pulley cover thickness: RMA—Grade II belting.

Operating Conditions	Thickness (inches)*
Non-abrasive materials	1/32
Abrasive materials	1/16
Impact loading**	3/32

*Note: Cover thicknesses are nominal values subject to manufacturers' tolerances.

**While an increased cover gauge helps protect the carcass, if impact is severe, a correct system design that includes carcass design, top cover thickness, and impact-absorbing belt support in the conveyor loading zone is the preferred method of handling.

Deteriorating Conditions

Table 7-6 establishes the basis for determining cover quality for conditions which attack or cause deterioration in the belt. The actual cover thickness generally should follow the guidelines for a Grade II cover in Table 7-4. For all special materials not listed, or where extreme concentrations of chemical solutions are likely to be encountered, a belt manufacturer should be consulted to determine appropriate cover quality and thickness.

Table 7-6. Deteriorating conditions for conveyor belt covers.

Typical Materials Handled Without Cover Deterioration	
Chemicals	Materials wetted with or containing the following chemicals and not over 150° F may be handled satisfactorily on conveyor belts with covers of Grades I and II: Black sulfate liquor Ethyl Alcohol Sulfur, elemental, dry Sulfuric acid (dilute)
Heat	Hot fine material up to 150° F/65°C Hot lump material up to 150° F/65°C
Fertilizers	Super phosphate Triple super phosphate Phosphate rock or pebbles, acid treated, to produce super or triple super phosphate

Note: With loading conditions resulting in maximum cover wear, the top cover thickness may have to be increased by 1/16" to 3/16" above the values listed in Table 7-4 in order to obtain a reasonable life.

*Materials Handled
Resulting in
Deterioration of
Covers*

Chemicals not listed *may* have a deteriorating effect on the rubber covers of conveyor belting, but, because of considerations of concentration and temperature, do not lend themselves readily to classification. Therefore, when handling chemicals not listed in Table 7-6, consult the belt manufacturer for cover quality recommendations.

Molded Covers

For special applications and/or unusual operating conditions, covers with special molded surfaces may be used to advantage. One type has a rough top, or various patterns of molded surface designed primarily for conveying packages up inclines, but is also occasionally used for conveying light-weight bulk materials on steep inclines. The second type is a ribbed or cleated cover used in bulk conveying to allow the conveyor incline to be increased without backsliding the load. Also, special designs for handling wet materials or slurries permit drainage or retention of fluids as required.

Frequency factor

The frequency factor indicates the number of minutes for the belt to make one complete turn or revolution. It can be determined using the following formula:

$$F_f = \frac{2L}{V}$$

Where:

L = Center-to-center length of the belt conveyor in feet

V = Belt speed, fpm

F_f = Frequency factor in minutes

For a frequency factor of 4.0 or over, minimum top cover thicknesses can be considered based on the loading conditions. For a frequency factor of 0.2, the appropriate top cover thickness should be increased up to twice this minimum amount. For frequency factors between 0.2 and 4.0, increase the top cover thickness accordingly.

Loading Considerations

- Loading Conditions Resulting in Normal Cover Wear*
- Material feed is in the same direction as belt travel. See Chapter 12, “Loading the Belt.”
 - Equivalent free fall of material onto the conveyor belt is not over 4 feet. See Tables 7-10 and 7-11.
 - Loading area of the belt conveyor is horizontal or has a slope of not more than 8 degrees.
 - Properly designed chutes and skirtboards to form, center, and settle the load on the belt. See Chapter 12, “Loading Chutes and Skirtboards.”
 - Material temperature is in the range of 30°F to 150°F (-1°C to 65°C).
 - Material handled contains nothing that will deteriorate the cover or carcass of the belt. See Table 7-6.

- Loading Conditions Resulting in Minimum Cover Wear*
- All the conditions above (resulting in normal cover wear) plus the following:
- In the process of loading, the material is traveling at approximately the same speed as the belt. See speed-up conveyor, Chapter 12, “Direction of Loading.”
 - Special attention has been paid to the equipment design in the loading area to reduce impact on the conveyor belt to a minimum. A minimum is an equivalent free fall of less than 3 ft. The use of a carefully designed loading chute and the use of adequate, impact-absorbing belt support are recommended.
 - Provision has been made in loading to place the fines on the conveyor belt first, to provide a bed for large lumps. See Chapter 12, “Loading Chutes,” and Figure 12.3.
 - With loading conditions resulting in minimum cover wear as herein defined, top covers of a conveyor belt sometimes can be reduced 1/32 inch to 1/16 inch from the values listed in Table 7-4.

- Loading Conditions Resulting in Maximum Cover Wear*
- Any of the following conditions constitute loading conditions resulting in maximum cover wear:
- Material is loaded 90 degrees transversely (right angle) to the direction of the belt. See Chapter 12, “Direction of Loading.”
 - Material is loaded at more than 90 degrees transversely to the direction of the belt. See Chapter 12, “Direction of Loading.”
 - Loading area has a slope in excess of 8 degrees to the horizontal.
 - Equivalent free fall of the loaded material is greater than 4 feet. See Tables 7-10 and 7-11.
 - Material loaded has no velocity in the direction of belt travel, or has a negative velocity in the direction of belt travel.

Breakers

Breakers are woven fabrics primarily of nylon and/or polyester. The “Leno” weave is most often utilized because of its open nature. Cord or transcord (woven cord) may also be incorporated. Placement of the breakers is generally above the carcass (or below if applied to the pulley cover). Because of the open nature of their weave, these fabrics tend to dissipate impact energy and help to prevent puncture of the belt carcass from sharp materials (rip-rap and hard ores) through the cover. Cover adhesion may also be increased by adding breakers, but this is not the primary reason for their incorporation today.

Molded Edge Belting

In the past, all belts were manufactured, totally encapsulated in rubber compound (molded edge). This was to prevent moisture and chemical degradation of what was then cotton and rayon carcasses. Today, with the almost exclusive use of synthetic fibers, the need to extend the covers around the edge of the belt is minimal. Molded edge belts are now made for applications where “stringing” (warp yarns from the carcass snagging on the system supports and idlers) is anticipated, or where process chemicals and leaching agents (as in metal ore processing) are concentrated enough to have adverse effect on the polyester and nylon carcasses. All steel cord belting is encapsulated or molded edge.

Cut/Slit Edge Belting

Cut or slit edge belting is the predominant heavy-duty belt type offered today. The polyester and nylon fabrics are resistant to moisture and mildew attack as well as the effects of most chemicals.

Steel Cord Belt Covers

Cover Carcass Dimensions:

To protect the steel cords from impact, abrasion, and water during the entire service life of the belt, a minimum thickness of rubber must encapsulate the cords. Table 7-7 indicates the minimum thickness “F” above and below the cords required for this protection.

Additional rubber B and C indicate the thickness used for cover wear. See Figure 7.2.

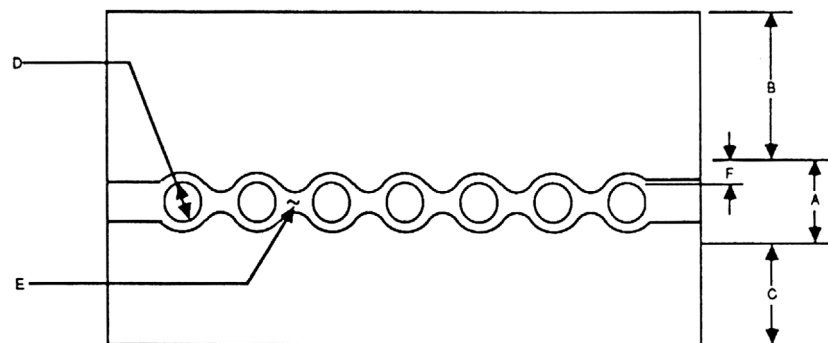


Figure 7.2 Protecting steel cords with rubber.

A = Protective covering for cords during the entire belt life

$$(A = 2F + D)$$

B = Amount of top cover used for the service life of the belt

C = Amount of bottom cover used for the service life of the belt

D = Diameter of the cord

E = Rubber encapsulating the steel cords and especially compounded for compatibility with the cover rubber and bonding to the steel cords

F = Thickness of rubber to protect the cords during service. This protective rubber is not part of the top or bottom wear covers used to estimate belt tonnage.

Table 7-7. Guide for minimum protective rubber “F” .

Cord Diameter		Minimum Thickness “F” (above and below cords)	
inches	mm	inches	mm
.161	4.1	3.5	0.137
.220	5.6	3.9	0.157
.327	8.3	5.8	0.228
.374	9.5	5.8*	0.228

Minimum thickness of protective rubber “F” should not be less than 3.5 mm or 0.7 times the cord diameter, whichever is greater.

*This value has been lowered from the calculated 6.6 mm as a result of favorable field experience. For thickness of covers “B” and “C,” consult belt manufacturer.

The Belt Carcass

The covers of a belt provide protection for the tension member or carcass of the belt. The carcass carries the load and is the primary reinforcement for resistance to tear and impact and for mechanical fastener retention.

Most conveyor belt carcasses are made of one or more plies of woven fabric/textile. High tension carcasses of single-layered, cabled-steel cord are also employed.

Carcass Types

Depending on the type and/or duty expected of a belt, various carcass styles are possible. The advent of high tenacity (strength) synthetic fibers (polyester/nylon/aramid) has changed conveyor belt and system design tremendously since their introduction. Although some multi-ply cotton carcass belts are still in use and manufactured, the thrust of heavy-duty conveyor belt technology has been directed at reduced multi-ply polyester and nylon belting with standard fabric offerings in pounds per inch width (PIW) tension ratings in multiples of 75, 110, 150, and 200 for standard “rubber” belting.

- Straight warp fabric/carcass design have also been available since the early 1980s which carry the concept of reduced-ply and lighter belting further.

- PVC belting often utilizes a solid or interwoven carcass for higher tension, heavy-duty conveyor systems.
- Ultra-high tension conveyors will require the use of steel cord/cable belts and, more recently, Aramid-based carcass design.
- Lightweight belting of either rubber, PVC, or urethane will generally have a woven, interwoven carcass of spun or filament polyester as well as standard cotton fabrics and blends.

Detailed descriptions of these fabrics and carcass designs are listed below as textile reinforcements.

Table 7-8. Some materials used in belting reinforcement (belt carcass).

Common Name	Composition	General Comments
Cotton	Natural Cellulose	Only natural fiber used to any great extent for belting. High moisture absorption. Susceptible to mildew attack and loss of strength.
Glass	Glass	High strength. Very low elongation. Used in high temperature applications.
Kevlar*	Aramid	Very low elongation and very high strength. Does not melt, but does decompose at high temperature.
Nomex*	Aramid	Very high strength and high elongation. Excellent high-temperature properties.
Nylon	Polyamide	High strength and high elongation, with good resistance to abrasion, fatigue, and impact. Moderate moisture absorption. High resistance to mildew.
Polyester	Polyester	High strength, low elongation. Good abrasion and fatigue resistance. Low moisture absorption. Excellent resistance to mildew.
Steel Cord	Steel	Very high strength, very low elongation. Superior troughing characteristics. Excellent heat resistance. Good fatigue and abrasion resistance.

*Trade names.

Textile Reinforcements

Textile fabrics are the most commonly used materials for reinforcing plies in conveyor and elevator belting. Textile fabrics are also used for conveyor belt “breakers” plies. Fabric properties are governed by the yarn material and size and by the fabric construction and weave. Standard heavy duty multi-ply belt fabrics are dip-treated with an Rescorchiol-Formaldehyde-Latex (RFL) coating to provide adequate adhesion with rubber compounds.

These fabrics are woven (usually at right angles) of warp yarns, which run lengthwise, and filling (weft) yarns, which run crosswise.

Non-woven fabric

A mat of fibers bonded together chemically and/or needle-punched to provide strength and flexibility.

Woven Fabrics

The most common, and least complicated, fabric pattern used for flat belts is the *plain weave* shown in Figure 7.3. In this construction the warp and filling yarns cross each other alternately. A belt with two or more of these layers of fabric is known as a multi-ply belt. Other common constructions used to a lesser degree include *broken twill* (Figure 7.4); *basket /Oxford weave* (Figure 7.5); and *Leno weave* (Figure 7.6), which has an open mesh and is usually used as a breaker fabric.

Woven Cord (Figure 7.7) is composed of strong warp yarns with very fine filling yarns used to hold the warp yarns in position.

Solid woven (Figure 7.8) consists of interwoven multiple layers of warp and filling yarns.

Straight warp fabric (Figure 7.9) contains basic tension-bearing warp yarns which are essentially straight, that is, without crimp. Also, binder warp yarns are interwoven with the filling yarns to provide mechanical fastener holding strength.

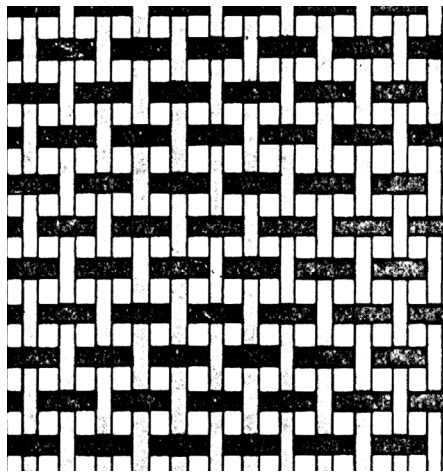


Figure 7.3 Plain weave.

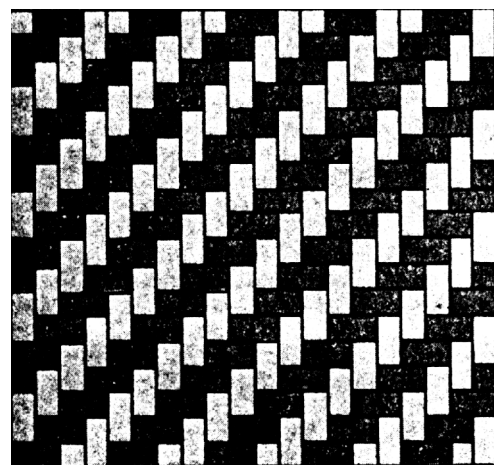


Figure 7.4 Broken twill/crowfoot.

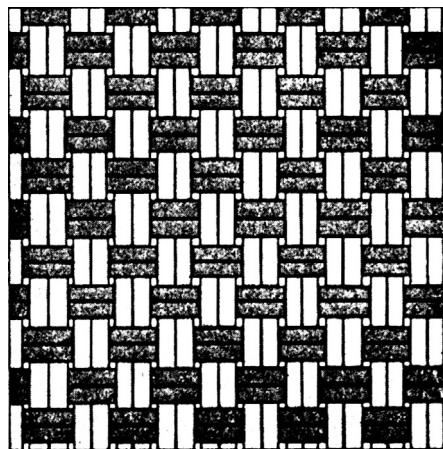


Figure 7.5 Basket or oxford weave.

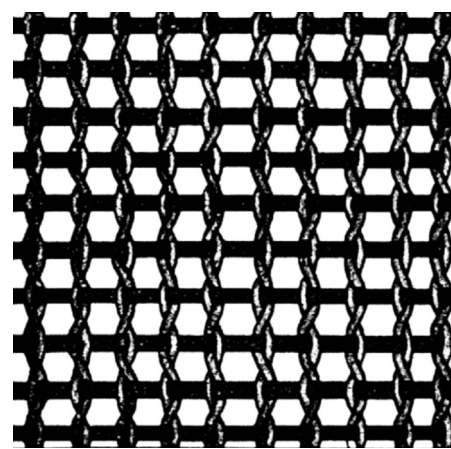


Figure 7.6 Leno weave.

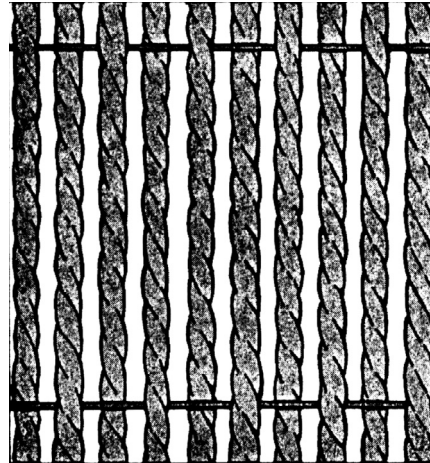


Figure 7.7 Woven cord.

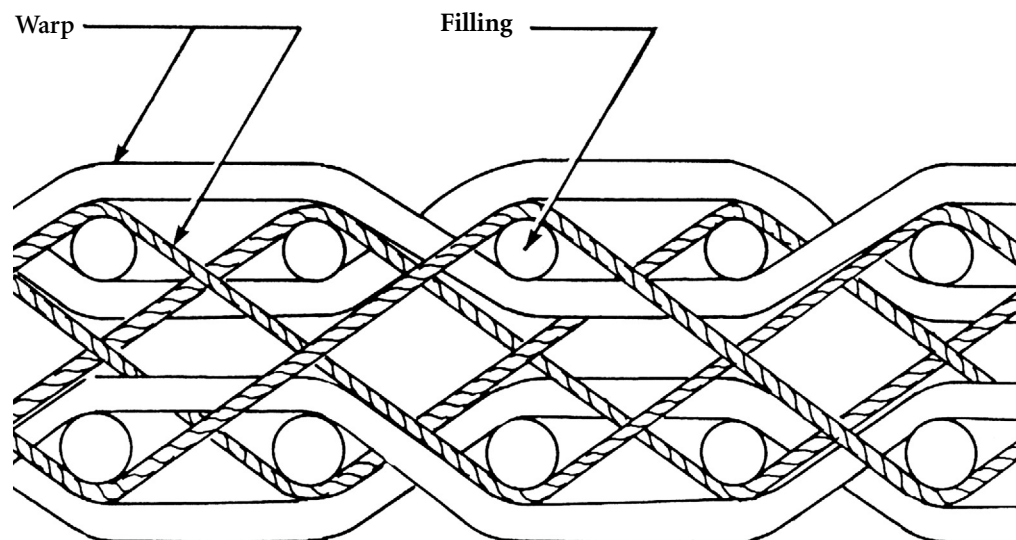


Figure 7.8 Solid woven fabric.

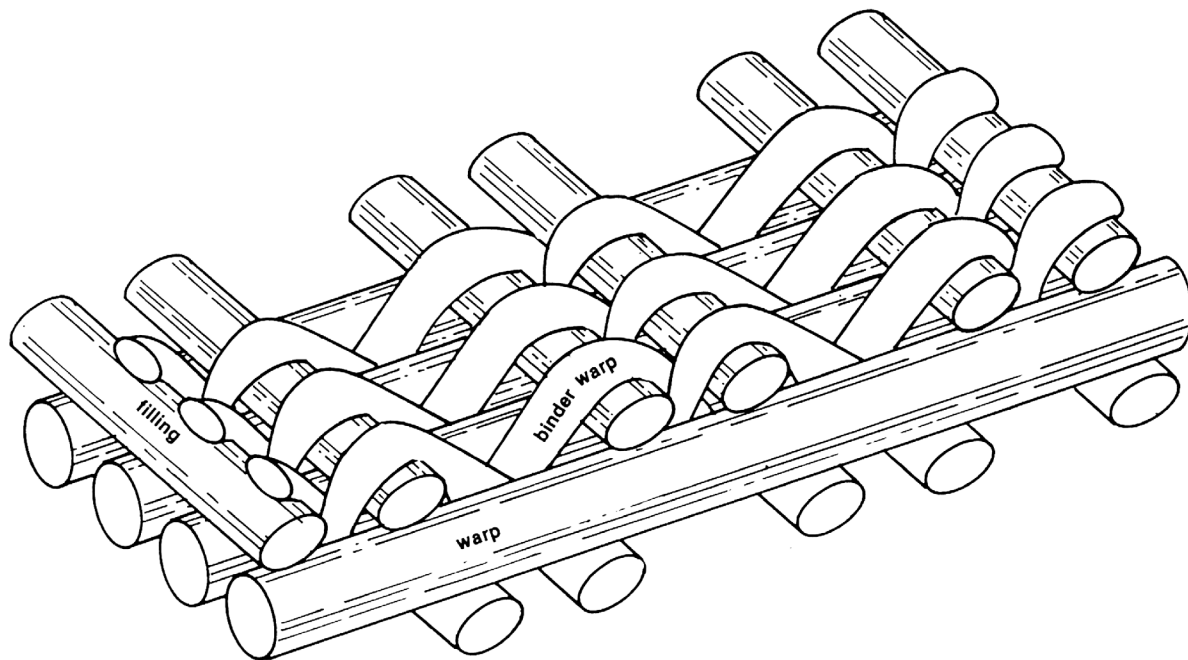


Figure 7.9 Straight warp fabric.

*Belting Fabrics by
Fiber Content*

Some of the most commonly used belting fabrics known by their major fiber content are:

Cotton — Fabric with cotton in both the warp and filling yarns.

Cotton-Synthetic — Fabric with cotton warp yarns and synthetic filling yarns. The synthetics most commonly used are nylon and polyester.

Polyester — Fabric with polyester fiber warp yarns and filling yarns.

Nylon — Fabric with nylon fiber warp and filling yarns.

Polyester-Nylon — Fabric with polyester warp and nylon filling yarns.

Solid woven fabrics are composed of spun and/or filament yarns. The spun yarns commonly used may be either cotton or synthetic or combinations thereof. The filament yarns are usually nylon or polyester.

**Steel
Reinforcements**

Steel Cord

Steel cord is used as a reinforcement in belting where it is better able to satisfy the requirements of the service conditions. Steel cord is used to obtain high strength,

excellent length stability, low bending stresses and, in some cases, to provide superior troughing characteristics. The wires or filaments used in conveyor belt steel cords are usually made of high carbon steel and have a surface finish which facilitates adhesion to the surrounding rubber and reduces corrosion during use.

Steel Cord Carcass

Steel cord conveyor belts are made with a single layer of parallel, uniformly tensioned steel cords serving as the load bearing member. The cords are completely embedded in elastomeric compound. As shown in Figures 7.10 and 7.11, steel cord belting is made in two construction types.

1. All gum compound construction with steel cords and elastomeric compound only.

2. Fabric reinforced construction with one or more plies of rubberized fabric above and/or below the steel cords and separated from the cords by the elastomeric compound. This construction is only used in special high impact service applications.

Both types have molded edges.

Steel cord belts have covers of elastomeric compounds selected for the expected service conditions.

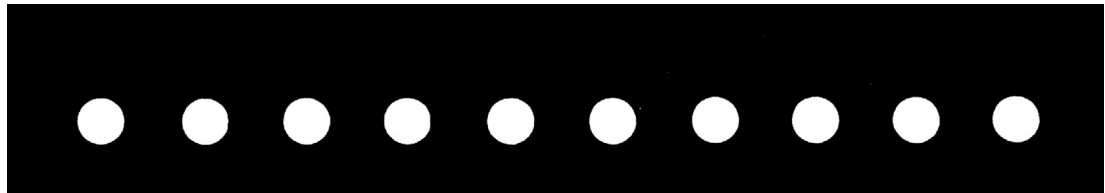


Figure 7.10 All-gum steel cord belt.



Figure 7.11 Fabric-reinforced steel cord belt.

Other Wire Components

Several other forms of wire are used in belting for special purposes such as rip resistance and transverse stiffness. A variety of wire structures is used, some of which include (1) pierce tape, (2) flat wire braid, (3) tire tread wire, and (4) wire tire cord.

Belt Splices

Conveyor belting is made endless, usually at the job site, by the use of either mechanical fasteners or vulcanized splices. Figure 7.12 illustrates a vulcanized fabric belt splice, and Figure 7.13 illustrates a steel-cord belt splice. The vulcanized-splice method provides a stronger connection and longer service life. However, in many

cases a mechanical fastener splice is acceptable, and in certain cases it actually may be preferred. Some of the advantages and disadvantages of vulcanized versus mechanically-fastened splices are described below.



Figure 7.12 Vulcanized-fabric belt splice.

Vulcanized Splice

Advantages:

- (1) Strength. It has the highest practical strength.
- (2) Long service life. Correctly applied on the appropriate conveyor equipment and properly cared for, a vulcanized splice can last for years. However, a vulcanized splice normally will not last for the life of the belt.
- (3) Cleanliness. A vulcanized splice is smooth and continuous. Thus, conveyed material cannot seep through it. Also, a vulcanized splice does not damage or interfere with belt cleaners, as can be the case with mechanically-fastened splices.

Disadvantages:

- (1) Greater initial expense. The initial cost of a vulcanized splice is many times greater than that of a mechanically fastened splice.
- (2) To insure sufficient takeup travel for accommodating both elastic and permanent variation in belt length, longer takeup travel must be provided.
- (3) Replacing or renewing a vulcanized splice can be time consuming and costly, especially in emergency repair situations.

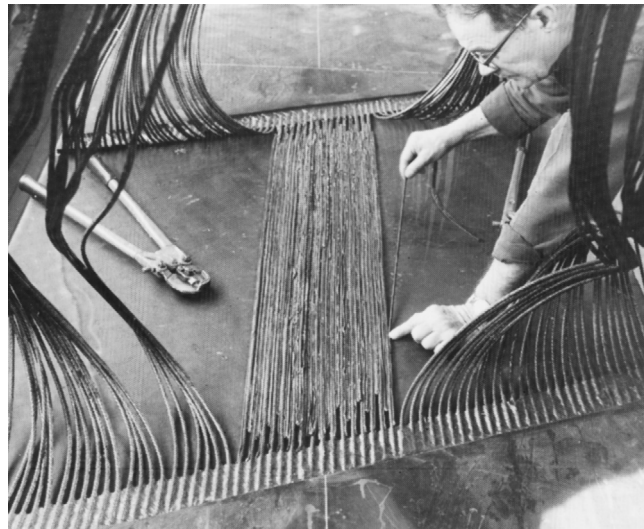


Figure 7.13 Steel-cable belt splice.

Mechanically Fastened Splice

Advantages:

- (1) Quick to make. A mechanically fastened splice can be installed by experienced personnel in a very short time, whereas it takes hours to complete a vulcanized splice.
- (2) Low initial expense. The cost of labor and fasteners for a mechanically fastened splice will be a fraction of the cost of a vulcanized splice. Usually, only hand tools are required.
- (3) Takeup travel problems are minimized. If belt length variations exceed the amount which the takeup is capable of accommodating, the belt can be shortened and respliced quickly at relatively small cost.

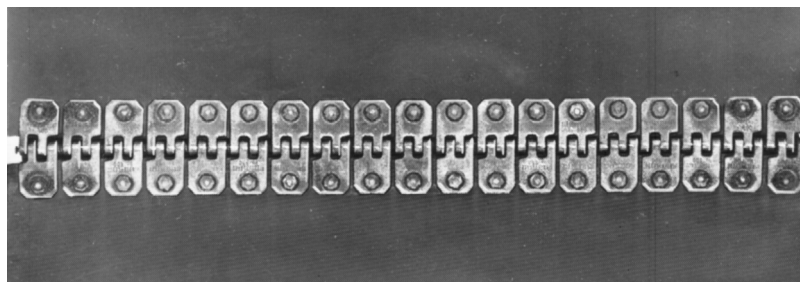


Figure 7.14 Hinged-plate type of mechanical splice.

Disadvantages:

- (1) Exposure of cut belt ends to the effects of moisture and materials may have a deleterious effect on the belt carcass fabric.

(2) Rough surface. Mechanical fasteners cannot be applied to produce as smooth a surface at the splice as that of a vulcanized splice. Belt cleaners and belt system supports sometimes catch on the fasteners with resultant damage to the splice, cleaner, or belt.

(3) It is very difficult to produce a mechanically fastened splice that can be considered leak-proof in conveying fine materials.

(4) In hot service, fasteners retain heat and transmit it directly into the belt carcass. This may cause local carcass degradation and early splice failure.

Belt and System Considerations

Elongation

The longitudinal (warpage) elongation of multi- or reduced-ply belts at the normal tension for the whole belt is usually less than 4 percent. The longitudinal elongation of the whole belt at its breaking strength is usually greater than 10 percent. This statement does not apply to certain high modulus, i. e., low stretch, belts.

Table 7-9. Recommended take-up travel in percent of center distance.*

	Fastened Splices		Vulcanized Splices	
	100% Rated Tension	75% or Less Rated Tension	100% Rated Tension	75% or Less Rated Tension
Screw Take-up	2%	1½%	4%**	3%**
Automatic Take-up	1½%	1%	2½% + 2 ft	

*For belts installed per the manufacturers' recommendations.

**Only short endless feeder belts and the like should be vulcanized on a conveyor with a screw take-up.

Troughability and Load Support

In order to achieve the desired carrying capacities without spillage over the edges, most bulk material-carrying conveyor belts are operated in a troughed configuration where the trough is usually formed by a 3-roll idler system as indicated in Figure 7.15 below. The angle of the troughing rolls will usually range from 20 degrees to 45 degrees.

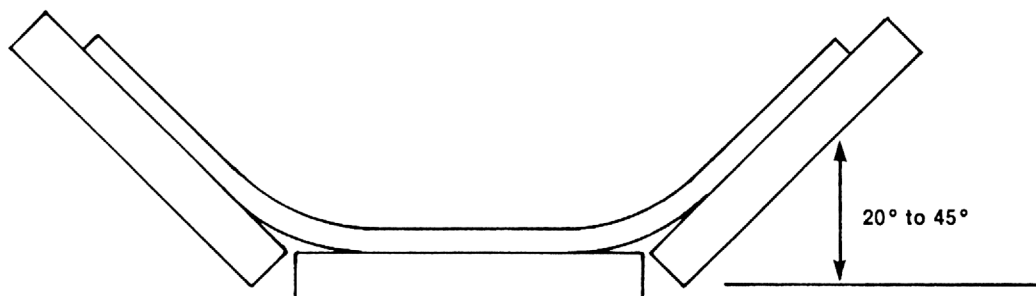


Figure 7.15 Belt troughing in-line idler.

When the belt is running empty, it must have sufficient lateral flexibility to retain contact with the center roll. Failure to do so will usually cause the belt to wander from side to side, and considerable edge damage may result.

Conversely, when the belt is running fully loaded, it must have sufficient lateral stiffness to support the load and bridge the gap between the center and troughing rolls. If the belt is too flexible in this regard, it may tend to crease into the idler gap and fail prematurely at that point. CEMA does not have a standard covering maximum roll gap for belt conveyor idlers. The roll gap varies with roll diameter and angle of trough, i.e.:

6-inch diameter rolls have smaller gap than 5-inch diameter rolls
35-degree trough has a smaller gap than 20-degree trough.

The potential problem can be reduced through selection and verification of application with the belt manufacturer.

From the foregoing, it is apparent that there are two extremes of lateral belt flexibility to be considered in making a belt selection, and these are generally referred to as minimum and maximum ply design. Reference to manufacturers' published tables is recommended, especially when the belt selection will be at or near either the minimum or maximum ply extreme, because of belt design variations and the fact that there are often two or more fabrics available with differing trough characteristics.

Several belt constructions made from one or more plies of synthetic fabrics are widely used and are generally referred to as multi- or reduced-ply constructions. Because of the wide variety of fabric strengths, constructions, and other factors offered in these types of belt, it is necessary to consult the various manufacturers for specific data. Tables showing typical belt selection data are on pages 221-224.

Impact Resistance

Loading bulk material on a conveyor belt creates some impacting force on the belt. This occurs since the material is dropped from some height above the belt surface and the forward speed of the belt may be different than the speed of the material when it contacts the belt.

Fine materials, regardless of weight per unit volume, do not present a problem on impacting the belt because the force is spread over a relatively large surface area. Cover damage due to gouging is minimal and carcass bruising is normally very low in operations involving fine materials.

Lumpy materials can cause appreciable impact on the belt. The heavier the lump, the greater height of fall, or the greater its angular velocity when it contacts the belt, the greater will be the energy tending to rupture the belt. When the material strikes the belt directly over a support such as an idler, damage to the carcass can result from the crushing action of the lump against the idler-supported belt.

Lumpy material having sharp corners and edges can cause cover nicks, cuts, and gouges. The heavier the lump, the greater height of fall, and the greater its angular velocity at the time of contacting the belt, the more extensive will be the damage to the cover. Sharp, pointed lumps can even penetrate the cover into the carcass and in rare instances completely penetrate through the belt.

To minimize impact damage, every effort should be made to provide good loading conditions for the material handled.

The selection of a cover grade and thickness, the type of textile fiber, fabric design, and number of plies can be varied depending upon the severity of the impact conditions. Given full information regarding the material conveyed and the loading conditions, the belt manufacturer can provide a belt that will embody the necessary elements to resist the damaging effects of impact.

The maximum fabric ratings shown in Table 7-10 are based on the use of impact-absorbing belt support idlers or cradles and good design of loading and transfer areas. The impact energy equals the lump weight factor (Table 7-11) times the equivalent free fall.

Table 7-10. Estimated maximum impact rating multi- or reduced-ply belts in foot-pounds (Joules).

Number of Plies	Fabric Rating per Ply (Pounds per Inch Width)				
	75	110	150	200	250
2	300 (407)	450 (610)	600 (814)	700 (949)	750 (1017)
3	400 (542)	600 (814)	700 (949)	800 (1085)	850 (1153)
4	500 (678)	700 (949)	800 (1085)	900 (122)	950 (1288)
5	--	800 (1085)	900 (1220)	1000 (1356)	1100 (1492)
6	--	900 (122)	950 (1228)	1100 (1492)	1200 (1627)

Notes: Where the actual impact energy is greater than that shown in the above table, consult the conveyor belting manufacturer.

The above impact rating is based on 10 percent lumps and 90 percent fines. If lumps exceed 10 percent, add one more ply to the belt than indicated in the table above.

For material 4 inches (100 mm) or less in size, use the values in the table; for material greater than 4 inches (100 mm) in size, add one more ply.

Equivalent Free Fall Calculation

Equivalent free fall is:

$$H_f + H_r (\sin^2 \Delta)$$

Where:

H_f = total free fall, ft (meters)

H_r = vertical height on loading chute slope, ft (meters)

Δ = angle in degrees that chute slope makes with the horizontal

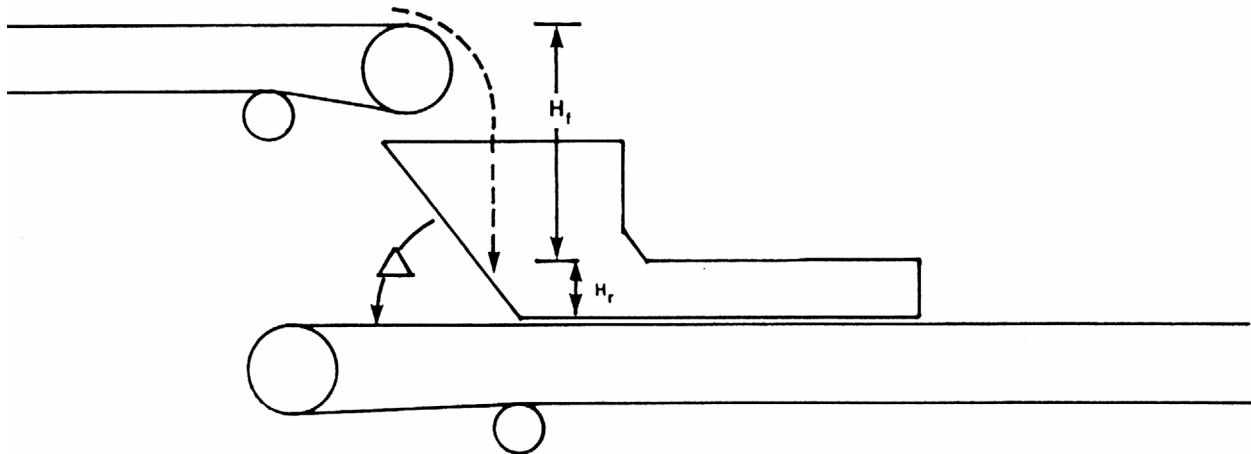


Figure 7.16 Equivalent free fall and location of values H_f and H_r .

Lump Weight Factor

The following tables are a close approximation of the weight of a lump based on cubic lump and slab breakage characteristics:

Table 7-11. Lump weight factor in pounds.

Density lbs/ft ³	Lump Size in Inches												
	2	3	4	5	6	7	8	9	10	12	14	16	18
50	0.4	1.3	3.0	5.8	10	14	21	30	40	70	100	148	211
75	0.6	1.9	4.5	8.6	15	21	31	44	61	105	149	222	316
100	0.7	2.6	5.9	12.0	20	28	41	59	81	140	199	296	421
125	0.9	3.2	7.4	14.0	25	35	52	74	101	175	248	371	527
150	1.1	3.8	9.0	17.0	30	42	62	89	121	210	298	444	632
175	1.3	4.5	10.4	20.2	35	49	73	104	142	245	348	518	737

Conveyor Belt Selection

Tension Ratings

The tension rating for a belt is the recommended maximum safe working stress that should be applied to the belt.

Belt tension is commonly referred to as the force applied to the belt per unit of belt width, such as pounds per inch width, or kilonewtons per meter width. Textile fabrics are frequently rated for their maximum safe working stress, which is expressed as the force applied per ply of fabric per unit width of the belt.

Information follows on multi- or reduced-ply, single-, and double-ply straight warp rubber belting tension ratings and also on PVC solid woven belting. This information is for illustrative purposes only.

There is no similarity among manufacturers about the following information that relates the number of plies of fabric, belt carcass thickness, minimum pulley diameter, troughability, etc. to the belt maximum safe working stress. These tables are presented to indicate the general availability of products.

Dissimilarities exist among manufacturers in:

1. The fiber, polyester and/or nylon, used for the fabric.
2. Recommended safe working strength for the fabric used.
3. Ratio of belt breaking strength to belt maximum safe working stress.

These factors, of course, affect the belt carcass thickness, belt weight, minimum pulley diameter, troughability, load support with different angle idlers, transition distance, impact resistance, etc. Thus, it is essential to confer with the belt manufacturer about the belt proposed for each application.

Because there is a wide variety of textile fibers, fabrics, and steel cords from which to choose, the belt manufacturer should be consulted for a recommendation of the type of belt carcass and tension rating of the belt that will best meet the user's needs.

Pulley Face

A crowned face is effective in centering a belt on the pulley if the approach to the pulley is an unsupported span that is unaffected by the steering action of idlers. Consequently, for troughed belts the crown on the head pulley of a conveyor is of little value in training the belt. Some benefit may be realized in terms of centering the belt through a tripper by having a crowned face on the tripper discharge pulley. Crowned face on low-tension bend pulleys in the return run and on a tail pulley where there is a relatively long unsupported span between return idler and the pulley may be slightly beneficial in centering the belt, but the contribution to overall training is minor.

Crowned-face pulleys should never be used for any pulleys on conveyors using steel-cable belt.

Multiple-ply belts should not be used on a crowned-face pulley where the tension will exceed 76 pounds per inch per ply. However, for all belts with textile carcasses, the best recommendation is that crowned pulleys be limited to locations where the belt will only be subjected to less than 40 percent of its rated tension.

Only straight-face pulleys should be used for all two-pulley drives and for drive snub pulleys.

The recommendations for the face width of belt conveyor pulleys are shown in Table 7-12. Return belt clearances are also indicated.

Table 7-12. Recommended pulley face width and belt clearances.

Conveyor belt widths b (inches)	Pulley face width P_f (inches)	Distance between discharge chute plates (inches)	Return belt clearance* minimum each side (inches)
42 and under	$b + 2$	$p_f + 3$	$2\frac{1}{2}$
over 42	$b + 3$	$p_f + 4$	3

*It may be desirable on conveyors with 500-foot centers and longer, for greater belt edge protection, that the next wider standard pulley face be used over that shown in the above table. For these conveyors, the stringers to carry the idlers should be spaced wider, allowing a return belt clearance of 5 or 6 inches or more, each side. This will result in increased cost of stringers, idler frames, and chutes, but often is considered worthwhile.

**Service
Conditions**

The data in the following tables apply if the following service conditions occur:

Mechanical Fastener Splice

1. Pulley diameters recommended by the belt manufacturer and fastener manufacturer are used.
2. No abnormal conditions, such as heat or chemicals, are applied to the belt that will reduce the belt fabric strength or change the flexibility of the belt fabric.
3. Recommended fasteners are properly applied.
4. Across the line, starting tension is limited to no more than 150 percent of the splice rating. Step phase or soft starting is preferred.

Vulcanized Splice

1. Pulley diameters recommended by the belt manufacturer are used.
2. Automatic take-up with adequate take-up travel is used.
3. Splices are made strictly in accordance with the belt manufacturer's specifications.

Where an adverse environmental condition or some special belt application exists, it is critical that the belt fabric ply tension rating be reduced by some factor recommended by the belt manufacturer. Some of the special conditions are:

1. Continuous excessive ambient temperature.
2. Exposure to deleterious chemicals.
3. Holes punched in the belt.

Elevator Belt Tension Recommendations

Elevator tension ratings may require modification under certain adverse environmental conditions. In such cases the rating in the following tables should be multiplied by an environmental factor of 0.75.

Adverse environmental factors for elevator belts include:

1. Elevated temperatures in the belt reinforcing fabric due either to high ambient temperatures or to conveying hot materials.
2. Abrasion of surface plies which are not protected by an elastomeric cover, such as friction surface belting in abrasive service.
3. Chemical service detrimental to the carcass fiber.

Table 7-13. Typical ratings for multi- or reduced-ply conveyor elevator rubber belting.

CONVEYOR															
Working Strength (PIW)	150	220	300	400	500	240	345	475	625	775	480	660	880	1000	1200
Number of Plies	2	2	2	2	3	3	3	3	3	3	4	4	4	4	6
Approximate Carcass Thickness (in.)	.12	.14	.14	.20	.210	.14	.19	.22	.25	.27	.265	.28	.335	.38	.50
Approximate Carcass Weight (lb/in. width)	.05	.075	.10	.104	.110	.08	.10	.108	.12	.135	.125	.141	.165	.188	.25
Minimum Pulley Diameter (in.)															
81-100%	14	16	16	18	24	18	24	24	30	30	30	30	36	42	--
61-80%	12	14	14	16	20	16	16	20	20	24	20	24	24	30	36
41-60%	10	12	12	14	18	12	14	18	18	20	16	20	20	24	30
To 40%	10	10	10	12	16	12	12	16	16	18	14	18	18	20	24
TROUGHABILITY															
Idler Troughing MINIMUM BELT WIDTH (INCHES) FOR EMPTY TROUGHING															
Angle 20°	14	14	18	24	24	20	24	24	30	36	36	42	48	--	--
35°	16	18	24	30	30	24	24	30	30	36	30	36	42	42	48
45°	NR	NR	24	30	30	30	30	30	36	42	36	42	48	48	54
MAXIMUM BELT WIDTH (INCHES) FOR LOAD SUPPORT															
Material Weight (lbs/cu ft)															
20° 0-40	42	54	72	66	72	60	72	84	84	84	72	84	84	84	116
41-80	36	42	60	60	66	60	60	72	72	72	72	72	84	84	108
81-120	36	42	54	54	60	42	54	72	72	72	72	72	84	84	108
Over 120	NR	36	48	48	54	42	48	48	60	60	60	60	72	84	96
35° 0-40	36	48	60	66	72	54	66	66	72	84	72	72	84	84	108
41-80	30	36	54	60	66	48	60	66	60	72	60	66	72	84	96
81-120	30	36	48	54	60	42	54	54	60	72	60	60	72	84	96
Over 120	NR	30	42	48	54	36	48	48	54	60	54	54	60	72	84
45° 0-40	36	48	60	60	66	48	60	60	72	72	72	72	84	84	108
41-80	30	42	48	54	60	60	42	54	60	60	54	72	72	84	96
81-120	24	36	48	48	54	36	48	48	54	60	54	72	72	84	84
Over 120	NR	30	36	42	48	30	42	42	54	54	48	54	54	72	84
ELEVATOR															
Minimum Pulley Diameter															
81-100% tension	16	16	18	22	26	20	18	24	30	36	22	30	36	42	48
61-80%	14	14	16	20	24	18	16	20	24	30	20	24	30	36	42
Up to 60%	12	18	14	22	14	14	18	20	24	18	20	24	30	36	--
Maximum Bucket Projection															
Spaced Industrial (100 lbs/cu.ft)	6	7	7	9	10	6	7	10	10	11	9	10	11	12	12
Spaced Continuous	NR	6	6	9	11	5	7	10	12	14	9	12	14	12	20

Belt Selection

Table 7-14. Typical ratings for straight warp conveyor or elevator rubber belting.

CONVEYOR													
	SINGLE PLY STRAIGHT WARP FABRIC						DOUBLE PLY STRAIGHT WARP FABRIC						
Working Strength (PIW)	190	220	225	330	385	440	440	550	660	800	1000	1250	1500
Approximate Carcass Gauge (in.)	.078	.103	.125	.131	.157	.165	.195	.234	.250	.281	.320	.328	.359
Approximate Carcass Weight (lbs/PIW)	.038	.049	.056	.064	.067	.072	.105	.114	.120	.134	.148	.165	.172
MINIMUM PULLEY DIAMETER (INCHES) DEPENDING ON FASTENER SPLICE SELECTED													
Working Tension													
81-100%	16	16	18	20	24	24	24	30	30	36	42	42	42
61-80%	14	14	16	18	20	20	20	24	24	30	36	36	36
Up to 60%	12	12	14	16	18	18	18	20	20	24	30	30	30
TROUGHABILITY													
Troughing Angle													
MINIMUM BELT WIDTH (INCHES) FOR EMPTY TROUGHING													
20°	12	14	14	18	18	18	24	24	24	30	30	30	30
35°	14	20	20	24	24	24	30	30	30	36	36	36	36
45°	16	24	24	24	24	24	36	36	36	42	42	42	42
MAXIMUM BELT WIDTH (INCHES) FOR LOAD SUPPORT													
Load Weight (lbs/cu.ft)													
20° 41-80	36	42	54	60	60	66	84	84	84	84	84	84	84
81-120	30	36	42	48	48	54	84	84	84	84	84	84	84
Over 120	30	36	42	48	48	54	60	66	84	84	84	84	84
35° 41-80	30	36	42	54	54	60	84	84	84	84	84	84	84
81-120	24	30	36	42	48	48	84	84	84	84	84	84	84
Over 120	24	30	36	42	42	48	54	60	66	84	84	84	84
45° 41-80	30	36	42	48	48	54	60	66	84	84	84	84	84
81-120	24	30	36	42	42	48	54	60	84	84	84	84	84
Over 120	18	24	30	36	36	42	48	54	60	66	84	84	84
ELEVATOR													
MINIMUM PULLEY DIAMETER													
Working Tension													
81-100%	16	16	18	20	24	24	30	30	30	36	42	42	42
61-80%	14	14	16	18	20	20	24	24	24	30	36	36	36
Up to 60%	12	12	14	16	18	18	20	20	20	24	30	30	30
MAXIMUM BUCKET PROJECTION													
Spaced Industrial	6	7	8	9	9	9	11	11	13	13	14	15	16
Continuous Industrial	6	7	8	9	9	9	11	11	13	13	14	15	16

Table 7-15A. Typical ratings for PVC solid woven conveyor and elevator belting (Table A).

Working Strength (PIW)	75	90	100	120	125	150	170	200
Total Thickness (64th in.)								
FS X FS	4	6	6	7	9	9	11	13
C X FS	5	7	7	9	11	11	13	14
C X C	6	8	8	9	11	11	14	15
Weight/Sq. Ft. (lbs)								
FS X FS	0.37	0.47	0.48	0.53	0.68	0.70	0.92	1.05
C X FS	0.49	0.70	0.72	0.86	1.00	1.05	1.15	1.40
C X C	0.61	0.75	0.80	0.95	1.09	1.15	1.40	1.45
Minimum Pulley Diameter (in.) Conveyor — (depends upon fastener selected)								
FS X FS	1.5	1.5	2.0	2.0	2.5	3.0	3.0	6.0
C X FS	1.5	1.5	2.5	2.5	3.0	3.0	3.5	6.0
C X C	2.0	2.0	2.5	2.5	3.0	3.0	3.5	6.0
Minimum Pulley Diameter (in.) Elevator — (depends upon fastener selected)								
C X C	6	6	6	8	8	10	12	12
Maximum Bucket Projection (in.)								
0-50 lbs/cu.ft (grain, wood chips)	—	—	—	—	3	4	4	6
51-100 lbs/cu.ft (coal, sand, potash)	—	—	—	—	3	4	4	6
Troughability--Minimum Belt Width (in.) for Satisfactory Troughing While Empty								
Idler Troughing Angle								
20°	—	—	—	—	12	12	12	12
35°	—	—	—	—	18	18	18	18
Load Support--Maximum Belt Width (in.)								
0-35 lbs/cu.ft								
20°	—	—	—	—	24*	24*	30*	36
35°	—	—	—	—	18*	18*	24*	30
36-120 lbs/cu. ft								
20°	—	—	—	—	—	—	—	30
35°	—	—	—	—	—	—	—	24
*For materials up to 25 lbs/ft.								

Belt Selection

Table 7-15B. Typical ratings for PVC solid woven conveyor and elevator belting (Table B).

Working Strength (PIW)	250	300	350	400	450	500	600	750	1000	1200
Total Thickness (64th in.)										
FS X FS	—	—	—	—	—	—	—	—	—	—
C X FS	—	—	—	—	—	—	—	—	—	—
C X C	16	17	18	20	21	22	23	25	28	29
Weight/Sq. Ft. (lbs)										
FS X FS	—	—	—	—	—	—	—	—	—	—
C X FS	—	—	—	—	—	—	—	—	—	—
C X C	1.50	1.70	1.90	2.00	2.10	2.15	2.33	2.55	3.00	3.12
Minimum Pulley Diameter (in.) Conveyor (depends upon fastener selected)										
FS X FS	—	—	—	—	—	—	—	—	—	—
C X FS	—	—	—	—	—	—	—	—	—	—
C X C	6	6	8	8	10	10	12	14	16	18
Minimum Pulley Diameter (in.) Elevator (depends upon fastener selected)										
C X C	14	16	18	20	24	—	—	—	—	—
Maximum Bucket Projection (in.)										
0-50 lbs/cu.ft (grain, wood chips)	7	7	8	8	9	9	9	10	12	12
51-100 lbs/cu.ft (coal, sand, potash)	6	7	7	7	9	9	9	10	10	10
Troughability--Minimum Belt Width (in.) for Satisfactory Troughing While Empty										
Idler Troughing Angle										
20°	12	18	18	18	24	24	24	30	36	36
35°	18	24	24	24	30	30	30	36	42	42
Load Support--Maximum Belt Width (in.)										
0-35 lbs/cu.ft										
20°	36	42	42	42	54	54	72	72	72	72
35°	30	36	42	42	48	48	60	66	72	72
36-120 lbs/cu. ft										
20°	30	36	42	42	48	48	60	72	72	72
35°	24	30	36	36	42	42	54	66	72	72

NOTE: If you have a need not covered by Tables 7.15 A or B, consult the belt manufacturer. It is virtually impossible to determine every conveyor design in a simple table. Often wider belts can be used for a given trough angle; however, full application details should be provided to the belt manufacturer for recommendations.

Conveyor Pulley Assemblies

Pulley Types

Standard Drum Pulleys

Standard Steel Wing Pulleys

Advantages of Using ANSI Standards

Mine Duty Pulleys

Engineered Pulleys

Pulley Overloads

Pulley Diameters

Pulley Face Widths

Pulley Crowns

Pulley Weights

Pulley Lagging

Thickness and Attachment

Lagging Hardness

Lagging Grooving

High Tension Applications

Shafting

Shaft Materials

Resultant Radial Load

Shaft Sizing by Stress Limit

Shaft Sizing for Deflection

It is accepted engineering practice to consider pulleys and shafts together because they form a composite structure whose operating characteristics are mutually related. Therefore, they are discussed as one topic of belt conveyor design and construction in this chapter.

Conveyor Pulley Assemblies

Conveyor pulley construction has progressed from fabricated wood, through cast iron, to present welded steel fabrication. Increased use of belt conveyors has led industry away from custom-made pulleys to the development of standard steel pulleys with universally accepted size range, construction similarities, and substantially uniform load-carrying capacity for use with belts having a carcass composed of plies or layers of fabric. “Standard” drum and wing pulleys are suitable for these applications. The present trend, however, is to use higher tonnage conveyor systems with wider, stronger belts that incorporate a carcass of either steel cables or high-strength tensile members. In these applications, where high tensions are encountered, the use of custom-made “engineered” welded steel pulleys is dictated. See Chapter 7 for a description of the various types of conveyor belts.

Pulley Types

The most commonly used conveyor pulley is the standard steel pulley as shown in Figure 8.1. They are manufactured in a wide range of sizes and consist of a continuous rim and two end discs fitted with compression type hubs. In most wide-faced conveyor pulleys, intermediate stiffening discs are welded inside the rim. Other pulleys available are self-cleaning wing types, which are used at the tail, takeup or snub locations where material tends to build up on the pulley face, and magnetic types, which are used to remove tramp iron from the material being conveyed. Figures 8.1 through 8.8 illustrate the more common types of conveyor pulleys now in use.

Standard Steel Drum Pulleys

Standard welded steel drum pulleys are defined by American National Standards Institute (ANSI) standard number B105.1. The standard establishes load ratings, allowable variation from nominal dimensions, permissible crown dimensions, and overall dimensions normally necessary to establish clearances for location of adjacent parts. The standard applies to a series of straight and crown-faced welded steel conveyor pulleys which have a continuous rim and two end discs, each with a compression type hub to provide a clamp fit on the shaft.



Figure 8.1 Typical welded steel pulley.



Figure 8.2 Fabricated curve crown pulley.



Figure 8.3 Spun-end curve crown pulley.

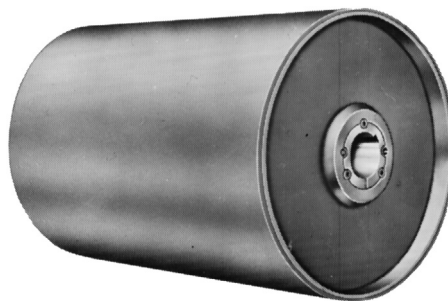


Figure 8.4 Lagged welded steel pulley.



Figure 8.5 Welded steel pulley with grooved lagging.

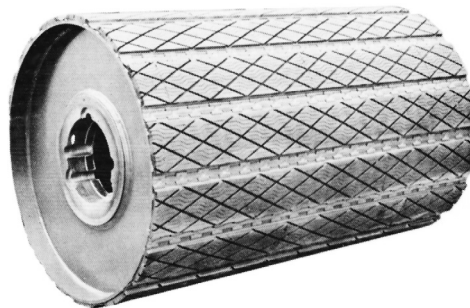


Figure 8.6 Slide-lagged pulley.

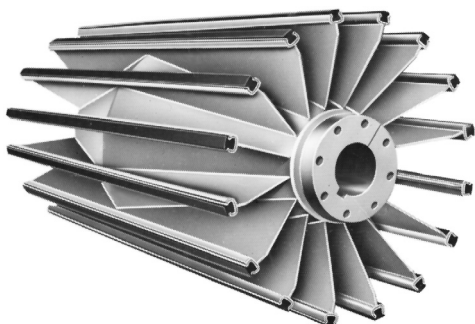


Figure 8.7 Lagged wing pulley.

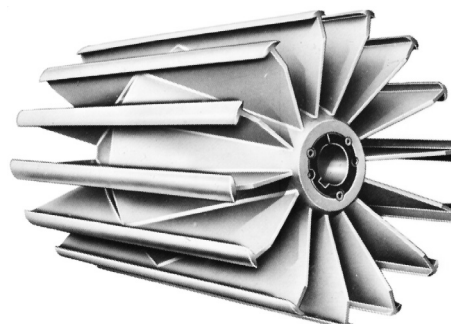


Figure 8.8 Fabricated wing type pulley.

The tabulated ratings for drum pulleys and shaft combinations are based on using non-journalled shafting with pulleys centrally located between two bearings. Ratings are based on either a maximum shaft bending stress of 8000 psi or a maximum free shaft deflection slope at the pulley hub of 0.0023 in./in., whichever governs. High strength shafting may be required with drive pulleys to withstand the added shaft stresses from torsional loads, overhung loads or turndowns for bearings. Belt tension limits are also provided and must be checked, especially for pulleys with low arc of contacts such as snubs or bends.

ANSI B105.1 is not applicable to single disc pulleys, wing pulleys, cast pulleys, or pulleys not utilizing compression hubs. The standard is not intended to specify construction details other than those listed above. The standard covers pulleys up to 60" diameter for shaft diameters up to 10 inches and face widths up to 66 inches and therefore encompasses the majority of combinations of welded steel pulleys with compression type hubs that are normally used in current belt conveyor and elevator practice.

Welded steel conveyor pulleys covered by ANSI B105.1 should not be used with steel cable or other high modulus conveyor belts because such belts create tension concentrations and demand manufacturing tolerances beyond the capacities of these pulleys. Such conveyor belts require engineered conveyor pulleys.

Standard Steel Wing Pulleys

Standard welded steel wing pulleys are defined by American National Standards Institute standard number 501.1. Like the drum pulley standard, this standard establishes load ratings, allowable variations from nominal dimensions, permissible crown dimensions, and overall dimensions normally necessary to establish clearances for location of adjacent parts. The standard covers pulleys up to 36 inches diameter for shaft diameters up to 8 inches and face widths up to 66 inches and therefore encompasses the majority of combinations of welded steel wing pulleys with compression type hubs that are normally used in current belt conveyor and elevator practice.

The standard applies to a series of straight and crown-faced welded steel wing pulleys which have a number of steel wing plates that extend radially from the longitudinal axis of two compression hub assemblies and are equally spaced about the pulley circumference. The purpose of the compression hubs is to provide a clamp fit on the shaft. The wings are supported or joined by welded steel plates so arranged as to form the shape of two frustums of cones or regular pyramids joined at their bases. A contact bar with generous radii is attached to the outer longitudinal edge of each wing to provide contact area with the belt.

The tabulated ratings for wing pulleys and shaft combinations are based on using non-journalled shafting through the pulley hubs, with pulleys centrally located between two bearings. Ratings are based on either a maximum shaft bending stress of 6000 psi or a maximum free shaft deflection slope at the hub of 0.0023 in/in, whichever governs. Belt tension limits are also provided.

Welded steel wing pulleys covered by ANSI 501.1 should not be used with steel cable or other high modulus conveyor belts because of the eccentricity inherent in the construction of a wing pulley. It is not recommended to operate standard wing pulleys above a belt speed of 450 feet per minute. For higher speeds, manufacturers should be consulted.

Advantages of Using ANSI Standards

The standard drum and wing pulleys discussed above can be used in most conveyor applications. The designer of belt conveyors will find the ANSI standards listed above invaluable in determining specifications for pulleys and shafting and in finding information that will permit the framework and supporting bearings to be detailed into the design. Suitable pulleys conforming to these standards can be obtained readily from the principal pulley manufacturers. Many of the sizes are considered stock sizes and are in their inventory of finished goods.

Mine Duty Pulleys

Standard size drum and wing pulleys are also available in mine duty construction. A mine duty pulley is one whose material thicknesses have been increased for a very rigid, conservative design. Mine duty pulleys were originally specified and used for underground mining operations where the abusive environment and high cost of installation demanded a more conservative design.

No ANSI standard governs the load ratings or material thickness of mine duty pulleys. Each manufacturer has established its own specifications and should be contacted for complete information regarding this class of pulley. A mine duty pulley will have much lower stress and deflection on the various components than a corresponding standard pulley. The conservative, rigid construction of mine duty pulleys makes them particularly useful for spares or replacements in critical or troublesome locations. Mine duty pulleys can be appropriate for conveyors with frequent starting and stopping, overloads exceeding 150% of running tensions, or where maximum reliability is necessary.

Engineered Pulleys

An engineered pulley is one which has been specifically designed to meet load conditions of a particular conveyor pulley. Specific information is required for proper and economical design, since the pulley designer must allow for sufficient strength in the rim, disc, shaft, and mounting system to carry the belt loads and to assure proper pulley to shaft connection.

Modern conveyor belting is often manufactured with steel cables or other high strength, high modulus tensile members for applications requiring high production or where large volumes of material need to be conveyed. Typical applications include overland conveyors, primary conveyors in open pit mines, large coal fired power plants, and ore processing facilities.

Special considerations in pulley design are required because of the increased loads imposed by these belts. Experience in design and application has helped engineers meet the need for pulleys that will accommodate the higher belt tensions generated under these conditions.

High tension belts, because of their high modulus, low stretch characteristics, require conveyor pulleys which conform to considerably higher standards than those of the pulleys used with normal fabric ply belts. Startup, braking, and other dynamic loads are more directly transmitted to pulleys. Concentricity of the pulley and proper alignment of the pulley and conveyor are of prime importance when high tensions are involved. Special emphasis should be placed on the importance of adequate, properly aligned structural supports for the pulley assembly to prevent load concentrations and overloads on the pulley resulting from misalignment.

Construction details vary widely, ranging from pulleys with flexible end discs and compression type hubs for plain shafting, to very heavy, rigid end discs fitted to specially machined shafts. Pulleys designed for this class of service are subject to more

Pulleys and Shafts

rigid manufacturing specifications, usually requiring accurately machined surfaces and close control of welded joints.

Table 8-1. Welded steel drum pulley weights, pounds. (These are representative weights for estimating purposes. Since actual weights may vary, final shaft and pulley designs should be checked with the manufacturer.)

Pulley		Belt width (inches)																		
Dia (in.)	Max Bore (in.)	18		24		30		36	42		48		54		60		72	84	96	
		Pulley face width (inches)																		
		20	22	24	26	30	32	36	38	44	46	51	54	57	60	63	66	78	90	102
6	2 ½	30	35	35	40	40	45	45	50	60	60	65	65	70	75	80				
8	2 ½	45	50	50	55	60	65	70	75	80	85	95	100	105	110	115				
10	2 ½	55	60	60	65	70	75	85	90	100	105	110	120	125	130	135	140			
	3	60	65	70	75	80	85	95	100	110	115	120	125	130	140	145	150			
	3½	75	80	80	85	90	95	105	110	120	125	130	135	140	145	150	155			
12	2½	70	75	75	85	90	100	105	115	120	130	140	150	155	160	165	175			
	3	80	85	90	100	105	110	115	125	130	135	145	155	160	170	175	180			
	3 ½	95	100	105	110	115	120	125	135	145	150	160	165	170	175	180	190			
14	2 ½	85	90	95	100	110	120	135	145	165	180	195	205	215	225	235	245			
	3	95	100	105	115	120	130	145	155	180	195	210	220	230	240	250	260			
	3 ½	105	110	115	130	140	150	160	175	190	205	220	230	240	245	255	265			
	4	130	135	140	150	160	170	185	200	210	220	235	245	255	260	275	280			
16	2 ½	105	110	115	130	140	150	160	175	200	220	235	240	250	260	270	280			
	3	115	125	130	140	150	165	170	185	220	240	250	260	270	280	290	300			
	3 ½	130	135	140	155	160	170	180	200	225	240	260	265	275	285	295	310			
	4	160	165	175	185	195	205	215	225	245	260	280	285	290	300	310	325			
18	2 ½	125	130	140	160	170	185	200	215	240	265	280	300	310	320	330	350			
	3	135	150	160	180	190	205	215	230	250	275	290	305	320	335	345	360			
	3 ½	145	155	160	180	200	215	225	240	265	275	295	310	330	345	355	375			
	4	175	185	190	215	225	235	250	270	295	310	330	355	375	400	410	430			
	4 ½	200	205	210	235	250	260	275	290	315	325	345	370	395	410	425	450			
20	2 ½	145	160	170	190	200	215	230	260	300	310	330	350	370	390	400	420			
	3	160	170	180	200	220	240	250	275	300	320	350	365	380	390	400	420			
	3 ½	170	180	195	210	225	240	270	285	300	330	360	375	390	400	415	435	500	550	
	4	190	200	215	240	250	270	285	305	330	360	400	410	425	435	450	470	550	600	
	4 ½	210	225	235	265	280	290	305	325	350	375	400	425	440	455	475	500	580	630	
	5																	600	650	
	6																	750	825	
24	3	200	210	230	250	265	285	300	325	400	425	450	475	500	510	525	550			
	3 ½	215	225	245	265	280	300	315	350	400	435	460	485	500	515	535	560	600	675	
	4	235	245	265	285	300	320	345	380	410	450	500	520	540	560	580	600	700	750	
	4 ½	255	265	280	300	315	335	370	400	460	480	500	550	570	590	610	635	725	800	
	5	280	290	300	335	365	400	420	440	500	530	560	615	635	655	680	710	900	950	
	6																	1100	1200	
	7																	1100	1200	
	8																	1200	1300	
30	3	315	330	350	380	400	430	450	485	525	550	670	700	750	800	850	900			
	3 ½	325	340	360	390	410	440	475	500	550	600	690	700	750	800	850	900	925	1000	
	4	350	375	400	425	450	475	500	525	575	635	700	750	800	825	860	900	1100	1200	
	4 ½	370	385	400	435	460	490	520	550	600	650	700	780	815	860	900	950	1100	1200	
	5	400	420	440	470	500	525	550	580	630	690	750	840	870	900	950	1000	1200	1300	
	6	525	550	575	600	625	650	680	740	775	825	875	900	925	950	1000	1050	1500	1600	
	7	550	560	575	600	650	675	725	750	800	850	900	950	975	1000	1050	1100	1500	1700	
	8																	1600	1800	
	9																	2200	2300	
	10																			

Conveyor Pulley Assemblies

Table 8-1. Welded steel drum pulley weights, pounds. (These are representative weights for estimating purposes. Since actual weights may vary, final shaft and pulley designs should be checked with the manufacturer.)

Pulley		Belt width (inches)																					
Dia (in.)	Max Bore (in.)	18		24		30		36		42		48		54		60		72		84		96	
		Pulley face width (inches)																					
		20	22	24	26	30	32	36	38	44	46	51	54	57	60	63	66	78	90	102			

36	3 ½	450	475	500	550	590	625	650	675	800	840	880	925	950	1000	1075	1150	1400	1500	1700			
	4	475	500	540	575	600	650	675	700	800	850	900	950	975	1000	1100	1200	1400	1500	1700			
	4 ½	490	500	550	600	625	650	700	750	800	850	900	950	1000	1050	1100	1200	1500	1600	1800			
	5	530	550	575	630	675	700	735	775	850	925	960	1000	1050	1100	1200	1300	1700	1900	2100			
	6	635	660	690	735	765	800	835	875	1000	1030	1100	1200	1250	1275	1300	1400	1900	2000	2200			
	7	725	750	800	850	900	950	1000	1050	1150	1175	1250	1350	1375	1400	1475	1550	2000	2100	2300			
	8	—	—	850	900	950	1000	1050	1100	1200	1250	1300	1400	1450	1500	1550	1600	2100	2300	2500			
	10																	2700	2900	3200			
42	4	625	650	700	750	800	850	900	950	1000	1075	1150	1200	1250	1300	1350	1400	1900	2100	2300			
	4 ½	640	675	700	775	850	875	900	950	1050	1100	1175	1225	1275	1300	1350	1500	2000	2200	2400			
	5	675	700	740	800	850	900	950	1000	1075	1150	1200	1250	1300	1350	1400	1500	2100	2200	2400			
	6	800	825	875	925	960	1000	1050	1125	1200	1250	1325	1400	1475	1525	1600	1700	2200	2400	2600			
	7	975	1000	1050	1100	1150	1200	1260	1325	1425	1500	1550	1600	1700	1750	1800	1900	2800	3100	3400			
	8	1025	1050	1100	1150	1200	1250	1300	1375	1450	1525	1600	1650	1700	1750	1825	1900	3000	3300	3600			
	10																	3500	3700	4000			
48	4	775	800	825	950	1000	1050	1075	1125	1200	1300	1500	1575	1650	1700	1750	1800	2500	2700	2900			
	4 ½	800	825	875	950	1050	1100	1150	1200	1300	1450	1600	1650	1700	1750	1850	2000	2500	2700	2900			
	5	825	850	875	1000	1050	1100	1175	1225	1300	1450	1600	1700	1750	1800	1900	2000	2500	2700	3000			
	6	900	950	1000	1100	1175	1225	1275	1350	1450	1600	1750	1825	1900	1950	2000	2100	2700	2900	3200			
	7	—	1025	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2075	2150	2225	2300	3500	3700	4100			
	8	—	1150	1225	1300	1400	1450	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	3700	4000	4300			
	10																	4200	4400	4800			
54	4½	1050	1100	1200	1300	1375	1450	1525	1600	1700	1800	1925	2000	2100	2200	2300	2400	3000	3200	3500			
	5	1100	1150	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2150	2200	2300	2400	3100	3300	3700			
	6	1225	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2250	2325	2400	2500	2600	3800	4100	4500			
	7	1350	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	4000	4300	4700			
	8	1500	1500	1550	1650	1750	1825	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	4400	4700	5100			
	10	—	2000	2050	2130	2230	2330	2400	2500	2625	2750	2850	2950	3025	3100	3200	3300	5000	5300	5700			
60	4 ½	1300	1350	1450	1600	1700	1800	1900	2000	2100	2200	2300	2400	2450	2525	2600	2700	4100	4600	5100			
	5	1325	1400	1500	1625	1750	1850	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	4300	4600	5100			
	6	1450	1500	1600	1700	1800	1900	2000	2100	2250	2400	2500	2600	2700	2800	2900	3000	4600	4900	5400			
	7	1600	1650	1750	1900	2000	2100	2200	2300	2400	2500	2600	2750	2850	2975	3000	3200	4600	4900	5400			
	8	1700	1750	1800	1900	2000	2100	2200	2300	2500	2600	2700	2800	2900	3000	3100	3200	5200	5500	6000			
	10	—	2200	2300	2400	2500	2600	2700	2850	3000	3100	3200	3300	3400	3500	3600	3800	5800	6100	6600			

Pulleys and Shafts

Table 8-2. Welded steel wing pulley weights, pounds.

These are representative weights for calculating purposes. Since actual weights may vary, final shaft and pulley designs should be checked with the manufacturer.

Pulley		Belt width (inches)																					
Dia (in.)	Max. Bore (in.)	18		24		30		36		42		48		54		60		72		84		96	
		Pulley face width (inches)																					
		20	22	24	26	30	32	36	38	40	44	46	51	54	57	60	63	66	78	90	102		
8	1 3/8	40	45	50	55	60	65	70	75	80	85	90	100	105	110	120	125	135	160	190	225		
	2 1/2	50	55	55	60	70	75	80	85	90	95	100	110	115	120	125	130	140	160	200	230		
10	1 3/8	55	60	65	70	80	85	95	105	110	120	125	135	145	155	160	170	180	210	250	275		
	2 1/2	60	65	70	75	85	95	100	110	115	125	130	140	150	160	165	175	180	210	250	275		
12	1 3/8	80	85	95	100	115	125	140	145	155	165	175	190	205	215	225	230	240	290	335	380		
	2 1/2	80	90	95	100	115	125	140	145	155	165	175	190	205	215	225	235	245	290	335	380		
	3	90	100	105	115	130	140	150	160	170	185	195	210	220	235	245	260	270	315	365	410		
14	1 3/8	95	105	115	120	140	150	170	175	185	200	215	240	250	265	280	295	310	360	415	470		
	2 1/2	100	110	120	125	145	155	170	180	190	205	215	240	250	265	280	295	310	360	415	470		
	3	105	115	125	135	155	165	185	195	200	220	235	255	270	285	300	315	330	390	450	500		
	3 1/2	115	125	135	145	165	175	195	205	215	235	245	270	285	300	310	325	340	400	450	500		
16	2 1/2	115	125	135	145	165	175	200	210	220	240	250	275	290	300	320	335	350	400	460	520		
	3	120	130	140	150	175	185	205	215	230	250	260	290	305	320	340	355	375	450	500	570		
	3 1/2	130	140	150	160	180	195	215	230	240	260	275	300	315	335	350	365	385	450	520	600		
	4	145	160	170	180	200	215	240	250	260	280	300	325	340	360	375	390	410	480	550	625		
18	2 1/2	145	160	170	185	210	225	250	265	280	300	320	350	370	390	405	425	450	525	600	680		
	3	150	160	175	190	215	230	260	270	285	310	340	370	390	410	430	450	470	550	625	700		
	3 1/2	160	170	185	200	225	240	260	280	300	325	340	370	390	410	430	450	470	550	630	710		
	4	170	185	200	215	235	250	280	290	300	330	350	375	400	420	440	460	485	570	650	735		
	4 1/2	190	200	215	230	255	270	300	315	330	355	370	405	425	450	470	490	510	600	675	760		
20	2 1/2	160	175	190	205	230	250	275	290	305	335	350	385	400	430	450	475	500	600	675	760		
	3	165	180	200	215	240	255	285	300	315	350	360	400	425	450	470	500	515	600	675	760		
	3 1/2	175	190	205	220	250	265	290	310	325	355	370	405	430	450	475	500	520	600	700	780		
	4	190	200	220	230	260	280	310	325	340	370	390	420	440	465	490	510	530	625	700	800		
	4 1/2	210	225	240	250	280	300	325	350	360	390	400	450	470	490	510	540	555	640	720	810		
24	2 1/2	250	275	300	325	375	400	490	530	560	600	640	700	750	780	825	860	900	1000	1200	1400		
	3	280	305	335	360	410	435	490	530	560	600	640	700	750	780	825	860	900	1000	1200	1400		
	3 1/2	300	315	340	370	420	440	500	530	560	600	640	710	750	780	825	860	900	1000	1200	1400		
	4	300	325	345	375	425	450	500	545	570	620	650	710	750	780	825	860	900	1000	1200	1400		
	4 1/2	315	340	365	400	440	470	510	550	575	625	650	715	760	800	835	875	900	1000	1200	1400		
30	5	320	350	370	400	440	470	520	560	585	635	660	725	760	800	835	875	900	1100	1200	1400		
	2 1/2	400	435	470	500	575	610	685	710	785	850	900	950	1050	1100	1150	1200	1200	1500	1700	1900		
	3	400	435	470	510	580	610	690	750	790	860	900	975	1050	1100	1150	1200	1250	1500	1700	1900		
	3 1/2	400	435	470	510	580	610	690	750	790	860	900	980	1050	1100	1150	1200	1250	1500	1700	1900		
	4	410	450	480	520	590	620	700	760	800	875	900	1000	1050	1100	1150	1200	1250	1500	1700	1900		
	4 1/2	425	460	500	530	600	635	700	770	800	875	900	1000	1050	1100	1150	1200	1250	1500	1700	1900		
36	5	430	460	500	530	600	635	700	770	800	875	900	1000	1050	1100	1150	1200	1250	1500	1700	1900		
	6	500	530	565	600	660	700	760	800	850	920	955	1040	1100	1140	1200	1250	1300	1500	1700	1900		
	3	540	585	630	680	770	820	910	1000	1050	1150	1175	1300	1375	1450	1500	1600	1700	1900	2200	2500		
	3 1/2	540	585	630	680	770	820	910	1000	1050	1150	1175	1300	1375	1450	1500	1600	1700	1900	2200	2500		
	4	550	590	635	680	775	820	915	1000	1050	1150	1200	1300	1375	1450	1500	1600	1700	1900	2200	2500		
	4 1/2	560	600	650	700	800	830	925	1000	1050	1150	1200	1300	1375	1450	1500	1600	1700	1900	2200	2500		
	5	560	600	650	700	800	830	925	1000	1050	1150	1200	1300	1375	1450	1500	1600	1700	1900	2200	2500		
	6	625	675	700	750	850	900	1000	1050	1100	1200	1250	1340	1400	1475	1550	1600	1700	1900	2200	2500		
	7	675	700	750	800	900	950	1000	1100	1150	1200	1250	1350	1450	1500	1575	1650	1700	1950	2200	2500		
	8	700	740	775	825	900	950	1050	1100	1150	1225	1275	1400	1450	1500	1575	1650	1700	1950	2200	2500		

All the components that make up the pulley and shaft assembly must be integrated to provide a fully capable power transmission system. Several factors must be considered for proper application and economical design of conveyor pulleys intended for load conditions exceeding those necessary to conform to ANSI standards. For such engineered pulleys, the following information is required for proper design:

- Diameter, face width, straight or crowned
- Bearing centers
- Location of pulley; head, bend, snub, takeup, etc.
- Type of belt takeup; gravity, screw, etc.
- Type of conveyor belt
- Belt tensions on pulley
- Belt wrap angle on pulley
- Shaft diameter (at hub, bearing, drive) if predetermined
- Lagging specifications
- Overhung load with location on drive pulley shaft (if it exists)
- Horsepower
- Belt speed
- Starting mechanism (reduced voltage, fluid coupling, etc.)
- Special environmental conditions

Pulley Overloads

Normal running loads for standard pulleys should not exceed ratings in the ANSI B105.1 and 501.1 load tables. Starting and occasional peak loads should not exceed ratings by more than 50 percent. Overloads may result from such causes as starting, jam-ups, maladjusted take-ups, brakes, misalignment and excess amounts of material on the belt. Overload conditions may result in premature failure of either pulleys, shafting, or bearings.

Normal running loads for engineered class pulleys should not exceed the loads used for design. Starting and occasional peak loads should not exceed design loads by more than 50 percent. If starting or peak loads are expected to exceed design loads by more than 50 percent, that information should be provided to the pulley manufacturer to be considered in the pulley design.

Pulley Diameters

Standard steel drum pulley diameters are 6, 8, 10, 12, 14, 16, 18, 20, 24, 30, 36, 42, 48, 54, and 60 inches. Standard wing pulley diameters are 8, 10, 12, 14, 16, 18, 20, 24, 30, and 36 inches. All other diameters are considered special. These nominal diameters apply to straight and crown-face pulleys and are for bare pulleys only; they do not include any increases in diameter due to the application of lagging. Pulley diameters should be selected per belt manufacturer's recommendations, as described in Chapter 7, as well as for other drive or space considerations.

Permissible variations from nominal diameters of standard steel pulleys are based on face width as follows:

Table 8-3. Permissible face widths tolerances.

Pulley face (inches)	Variation			
	Over nominal		Under nominal	
	drum	wing	drum	wing
6-26	1/4	1/8	1/8	3/8
over 26-66	5/8	1/8	1/8	3/4

These limitations apply equally to straight-face and crown-face pulleys. The nominal diameter is measured at the midpoint of the pulley face width. The diameter is defined as the bare diameter exclusive of lagging. Listed variations may occur from one pulley to another. The permissible diameter variations listed are not to be construed as runout tolerance. Runout tolerance on diameter is measured at the midpoint of the bare pulley face and is as follows:

Table 8-4. Permissible runout tolerances for common applications.

Diameter (inches)	Maximum Total Indicator Reading
8-24	0.125
30-48	0.188
54-60	0.250

Engineered pulleys to be used with steel cable or high modulus belts are usually machined with a straight face having the following permissible runout tolerance:

Table 8-5. Permissible runout tolerances for high modulus belts.

Type	Maximum Total Indicator Reading
Unlagged Pulley	0.030
Lagged pulley (under lagging)	0.030
Lagged pulley (over lagging)	0.030

Pulley Face Widths

Pulley face width is the length of rim, wing or contact bar along the shaft axis. Standard pulley face width is normally equal to the belt width plus 2 inches for belt widths up to 42 inches and belt width plus 3 inches for belt widths over 42 inches. Engineered pulley face widths are generally 6 to 12 inches greater than the belt width to provide greater clearance. The conveyor belt should not wander beyond the edge of the pulley face. Refer to Chapter 7 for more information on pulley face widths.

Standard drum pulley face widths defined by ANSI standards are 20, 22, 26, 32, 38, 44, 51, 57, 63, and 66 inches, with standard wing pulley face widths also including 40, 54, and 60 inches. Face width variations from nominal, according to ANSI standards, are $\pm \frac{1}{8}$ inch for all size drum pulleys and $\pm \frac{1}{4}$ inch for all wing pulleys. The listed variations in face width may occur from one pulley to another. The permissible face width variations are not to be construed as an edge runout tolerance. Edge runout tolerance is specified by the individual pulley manufacturer.

Pulley Crown

See Chapter 7 for a discussion of the use of crowned conveyor pulleys. Straight-face pulleys are recommended for all installations using reduced ply, high modulus, low stretch belts, such as those with a carcass of steel cables or high strength tensile members. There are currently two types of pulley crowns available, the taper crown and the curve crown.

Taper Crown

On taper crown pulleys, the face forms a “V” with the rotating axis larger in diameter in the center of the pulley. This crown is expressed in inches of crown per foot of total face width, by which the diameter at the center of the face exceeds the diameter at the edge. Normal crowns of this type vary from 1/16 to 1/8 inch per foot of total face width.

Curve Crown

Curve crown pulleys have a long, flat surface in the center of the pulley with the ends curved to a smaller diameter. Except on short pulleys, the curved surface extends in approximately 8 inches from the edge.

Trapezoidal Crown

Trapezoidal crown pulleys have a flat surface in the middle portion of the pulley face with the ends tapered. Trapezoidal crown pulleys may be appropriate for wider face width pulleys.

Pulley Weights

Pulley weights must be used to determine pulley and shaft selection. Average weights for standard steel drum pulleys are given in Table 8-1, and weights for standard wing pulleys are listed in Table 8-2. Weights listed are averages. If more accurate values are required, it is necessary to obtain the actual pulley weight from the manufacturer, as there are some variations in manufacturing practices. Engineered pulley weights are dependent on the tensions encountered and can vary widely.

Pulley Lagging

Conveyor pulleys can be covered with some form of rubber, fabric, urethane, ceramic, or other material. Lagging is used on driving pulleys to increase the coefficient of friction between the belt and pulley. Lagging is also used to reduce abrasive wear on the pulley face and to effect a self-cleaning action on the surface of the pulley. Abrasive wear and material buildup can substantially decrease pulley life. Drive pulleys should always be lagged. Non-driving pulleys, especially on the carrying side of the belt, should be lagged whenever abrasive or buildup conditions exist.

Thickness and Attachment

Lagging thickness can vary from a few thousandths of an inch, as with a sprayed-on coating, to thicknesses of one to two inches, as with some solid rubber vulcanized

coatings. Common methods of attachment are bolting, cementing, tack welding, and vulcanizing. Vulcanized lagging is generally preferred for heavy duty or severe service applications. Bolted-on and welded-on slide type lagging has the advantage of being replaceable in the field. It can be obtained in various grooved and other specialized surface finish types. Bolted-on lagging usually consists of a rubber cover reinforced with multiple ply fabric construction similar to conveyor belting. The fabric plies are required to provide strength under the bolt heads. Slide-on lagging is constructed of rubber molded and vulcanized to steel backing plates. They are slid into steel retainer strips welded onto the pulley face. Replacement of the pads can be accomplished by sliding the old strip out and the new strip in without removing the pulley from its conveyor location.

Rubber Lagging Hardness

Rubber lagging used on drive pulleys normally has a durometer hardness of 60 ± 5 on the Shore A scale. The lagging used on non-driving pulleys may have a hardness of 45 ± 5 or 60 ± 5 , depending on the application. For snub and bend pulleys, which contact the carrying side of the belt, softer rubber tends to better resist buildup on the pulley face. With high tension belts, lagging of 70 ± 5 hardness is sometimes used.

Lagging Grooving

Drive pulleys which operate in wet or damp conditions are commonly grooved. These grooves generally take the shape of a herringbone or chevron shaped pattern cut into the lagging. Generally, the dimensions of the groove are $\frac{1}{4}$ inch wide by $\frac{1}{4}$ inch deep with a $\frac{1}{8}$ inch minimum thickness of material under the bottom of the groove. Grooves are usually spaced on $1\frac{1}{4}$ to $1\frac{3}{4}$ inch centers. In a chevron pattern, the grooves meet at the center of the pulley face, while in the herringbone pattern the grooves are offset by one-half the groove spacing. Figure 8.5 illustrates the herringbone pattern. In both patterns, the apex points in the direction of belt travel. There are also groove configurations and sizes of lagging which can be furnished to enhance belt tracking or reduce the accumulation of material on the pulley face.

High Tension Applications

With steel cable and other high modulus conveyor belts, lagging is almost always used on drive pulleys and is preferred on pulleys which contact the carrying side of the belt. For maximum pulley life and belt safety lagging can be used on all non-drive pulleys. Lagging for high tension applications should be solid vulcanized rubber machined concentric with the shaft.

Shafting

Suitable shafting to be used with a steel pulley cannot be selected independently of the pulley load rating. In fact, the load capacity of a given pulley is a function of the shaft that is installed in that pulley. The shaft and pulley must be treated as a composite structural assembly. This is because the structural rigidity of the assembly depends upon both the shaft and the pulley and their interaction.

The shaft diameter required for a pulley assembly is a function of two criteria, shaft diameter required for strength and shaft diameter required for deflection. Depending on the exact pulley assembly, either strength or deflection can be the determining factor for shaft diameter selection.

Shaft Materials

Pulley design is based upon the use of any commercial or standard shafting material, such as AISI C1018 or C1045 steel. Load ratings of standard or mine duty pulleys are not increased when higher strength shafting is used. High strength shafting is of value in cases where it permits the shaft ends to be turned down so that smaller diameter, high capacity antifriction bearings can be used. It is also sometimes of value on drive shafts to withstand the added torsional stresses. While the use of high strength steel increases the strength of the shaft, its use does not decrease deflection.

Resultant Radial Load

The resultant pulley radial load is the vector summation of the belt tensions, pulley weight, and weight of shaft. The force from weights always acts downward, and the forces from the belt act in the path of the belt and away from the pulley. A graphical representation, as illustrated in Figure 8.9, or trigonometric methods are simple means of obtaining the resultant load. When a drive pulley shaft has an overhung load outboard of the pillow block bearings, such as with a shaft mount reducer or chain drive, refer to the ANSI drum pulley standard B105.1 for inclusion of the effects of the overhung load.

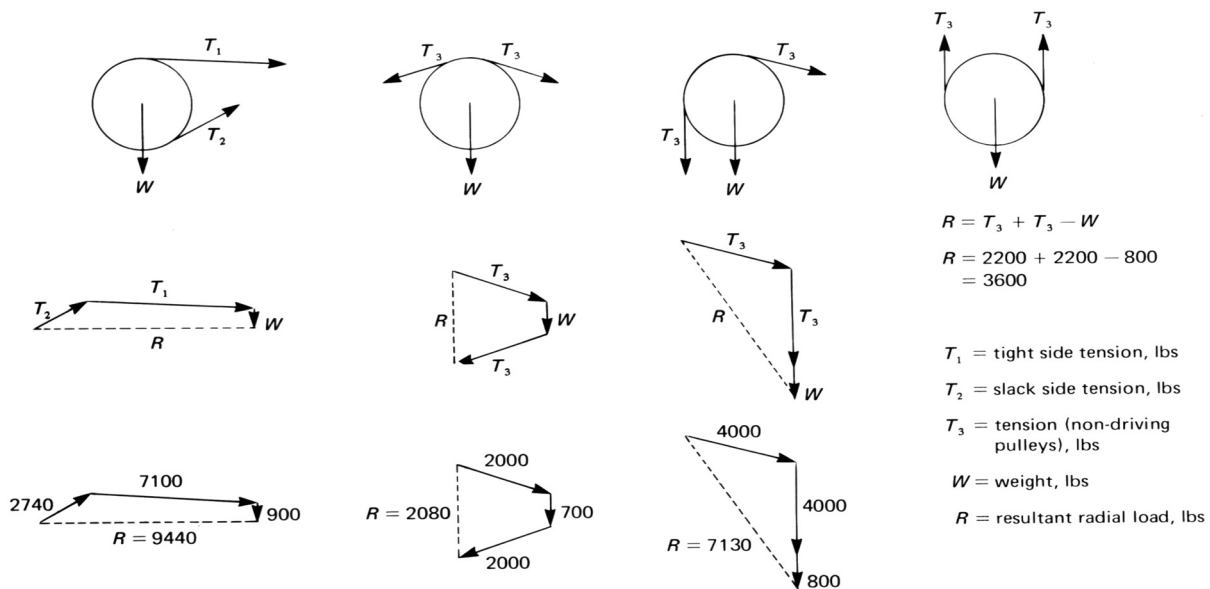


Figure 8.9 Graphical means of obtaining resultant radial load.

Shaft Sizing by Stress Limit

The equation given in standard B105.1 for the diameter of a pulley shaft loaded in bending and torsion (drive pulley with no overhung load) is:

$$D = \sqrt[3]{\frac{32FS}{\pi} \sqrt{\left(\frac{M}{S_f}\right)^2 + \frac{3}{4}\left(\frac{T}{S_y}\right)^2}}$$

The recommended values are:

FS , Factor of Safety = 1.5

S_f , Corrected shaft fatigue limit =

$$k_a k_b k_c k_d k_e k_f k_g S_f^*$$

Where:

k_a , surface factor = 0.8 for machined shaft

k_b , size factor = $(D)^{-0.19}$

k_c , reliability factor = 0.897

k_d , temperature factor = 1.0 for -70 to +400 degrees F

k_e , duty cycle factor = 1.0

k_f , fatigue stress concentration factor =

0.63 for steel (BHN < 200) with profiled keyway

0.77 for steel (BHN < 200) with sled runner keyway

0.50 for steel (BHN > 200) with profiled keyway

0.63 for steel (BHN > 200) with sled runner keyway

(The steels listed can be considered to be under 200 BHN)

k_g , miscellaneous factor = 1.0 for normal conveyor service

S_f^* = 29,000 for C1018

= 41,000 for C1045

= 47,500 for C4140 (annealed)

(S_f^* = 0.5 tabulated ultimate tensile strength)

S_y , yield strength = 32,000 for C1018

= 45,000 for C1045

= 60,500 for C4140 (annealed)

M = bending moment, pound inches

T = torsional moment, pound inches

For drive pulley shafts with overhung loads outboard of the pillow block bearings, such as a shaft mount reducer or chain drive, refer to the ANSI drum pulley standard B105.1 or vector methods for inclusion of the effects of the overhung load on M_B .

The method shown above can also be used to develop allowable bearing turn-down sizes by adjusting M_B for the correct moment arm and multiplying M_B and M_T by the stress concentration factors for shaft turndowns, available in many engineering texts.

For non-drive pulleys set $T = 0$. In the ANSI drum pulley standard B105.1, a value of 8000 psi is used for S_f for non-drive pulleys. In the ANSI wing pulley standard 501.1, a value of 6000 psi is used for S_f .

Shaft Sizing for Deflection

The pulley assembly is a structural unit, and the strengths of different components are interdependent. The shaft diameter, the bearing centers, the thickness of the end disc, the resultant loads on the pulley, the hubs, and the tapered bushings are all interconnected in this respect. A steel pulley with very thin end discs allows the shaft to deflect almost as it naturally would by virtue of the two loads applied to it through the hubs. A pulley made with very thick end discs has minimal shaft deflection. For this reason, only the pulley manufacturer can determine the actual deflection of the shaft at the end disc.

When selecting pulley assemblies, a shaft deflection limitation is required and varies for different types of pulley to shaft attachment methods. For preliminary calculations, the free shaft deflection, where the shaft is treated as a simple beam without any stiffening effects from the pulley, can be used for calculation. ANSI B105.1 and 501.1 specify free shaft deflection limits of .0023 in./in. for standard drum and wing pulleys. This represents a maximum deflection angle of 8 minutes at the pulley end disc.

For pulleys that must withstand higher belt tensions and higher resultant loads than those given in standards B105.1 or 501.1, pulley manufacturers design an engineered pulley. For engineered pulleys used in critical applications, such as steel cable belts or high tension belts for mining or power plant usage, the free shaft deflection should be limited to .0015 in./in. This represents a maximum deflection angle of approximately 5 minutes at the pulley end disc. The system designer often specifies a particular deflection limit.

The equation given in standard B105.1 for the free shaft deflection for any pulley assembly is:

$$\tan \alpha = \frac{RA(B - 2A)}{4E_y I}$$

Where:

- A = Moment arm for pulley (inches)
- B = Bearing centers (inches)
- R = Resultant pulley load (pounds)
- E_y = Young's modulus in psi
(29×10^6 for steel)
- I = Area moment of inertia of shaft in inches⁴
($.049087 D^4$)
- D = Diameter of shaft
- $\tan \alpha$ = Tangent of the angle made by the deflected shaft and its neutral axis before bending, at the pulley end disc

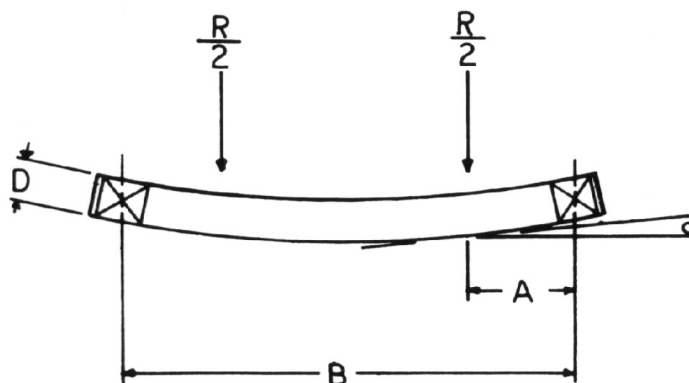


Figure 8.10 Shaft deflection.

For more information on shaft selection and allowable shaft deflection, consult a CEMA pulley manufacturer. It is suggested that complete pulley loading information be supplied with all pulley orders or requests for quotations.

Concave Vertical Curves:

Design; Graphical Construction; Precautions for Design

Convex Vertical Curves:

Design; Idler Spacing; Use of Bend Pulleys

Vertical curves in belt conveyors are used to connect two tangent portions which are on different slopes. They are of two basically different types: concave vertical curves, where the belt is not restrained from lifting off the idlers; and convex vertical curves, where the belt is restrained by the idlers.

Disregarding what may be the theoretically perfect shape for either curve, it is commercially satisfactory to consider them as arcs of a circle.

These curves occur on both the carrying and the return runs of the belt and in a variety of tension conditions. The reader is referred to Chapter 6, "Tension Relationships and Belt Sag Between Idlers," especially Figures 6.8 through 6.16, for various common belt conveyor profiles and tensions.

For simplification, the text and diagrams in this chapter are principally concerned with vertical curves on the carrying run of the belt conveyor.

Concave Vertical Curves

A conveyor belt is said to pass through a concave vertical curve when the center of curvature lies above the belt. (See Figure 9.1.) In such cases, the gravity forces of the belt and the load (if present) tend to hold the belt down on the idlers while the tension in the belt tends to lift it off the idlers. It is necessary to proportion the vertical curve so that the vector sum of these forces acts in a direction which allows the belt to stay down on the idlers and insures that the load will not be spilled. It is preferable that the belt does not lift off the idlers under any condition, including the starting of the empty belt.

If this is not practical, it is permissible to let the empty belt lift off the idlers if the following conditions are met: (1) Nothing above the belt will damage it (e.g., headroom of the structure, tunnel, skirtboards, guard rails, belt cover or machinery, etc.). Sometimes the empty belt can be protected from such sources of damage by locating flat idlers above the carrying strand. (2) Wind will not affect the proper training of the belt. (3) Lack of troughed support will not result in spillage as the loaded portion of the conveyor belt approaches the vertical concave curve.



Figure 9.1 Concave vertical curve.

Design of Concave Vertical Curves

Because of the considerations mentioned above, it is good practice to design vertical concave curves with sufficient radius to allow the belt to assume a natural path on the troughing idlers under all conditions.

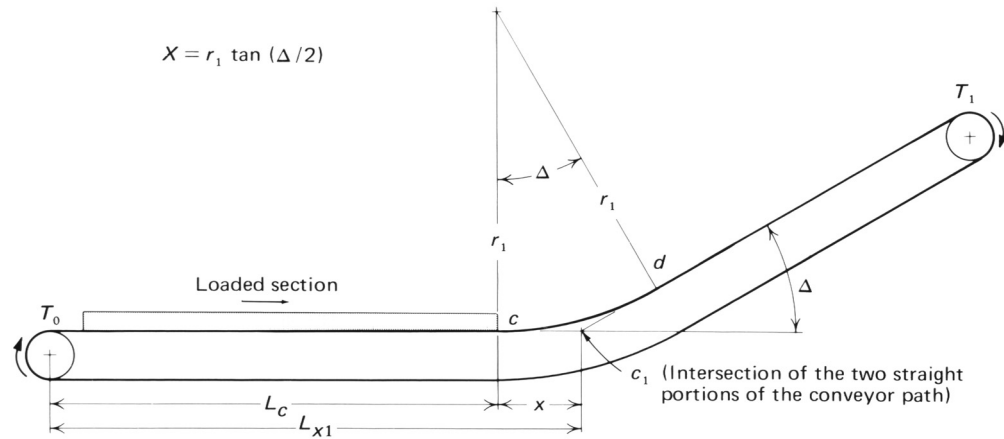


Figure 9.2 Profile of conveyor with concave vertical curve.

The illustration in Figure 9.2 makes it clear that the location of the beginning of the concave vertical curve, point c , tangent point of the curve, is indeterminate until the minimum radius is known. However, a close approximation can be made by assuming that the beginning of the concave vertical curve is at point c , in Figure 9.2. After determining the minimum radius from the following formulae, a second and exact calculation should be made.

The following formulae all involve point c , the start of the concave curve. But, for the first approximation, point c will be used.

To prevent the belt from lifting off the idlers with the belt conveyor running, the formula is:

$$\text{Minimum radius, } r_1 = \frac{1.11T_c}{W_b} \quad (1)$$

Where:

- r_1 = minimum radius, ft, to prevent belt from lifting off the idlers
- T_c = belt tension, lbs, at point c (or c_1)
- W_b = weight of belt per foot, lbs
- 1.11 = constant, based on maximum conveyor incline of 25° to horizontal

See Figure 9.3.

However, two hazards may exist. These require checking. The first involves the tendency of the belt edges to buckle when the tension in the belt is too low. The second is the possibility that the tension in the center of the belt may exceed the allowable tension in the belt.

Vertical Curves

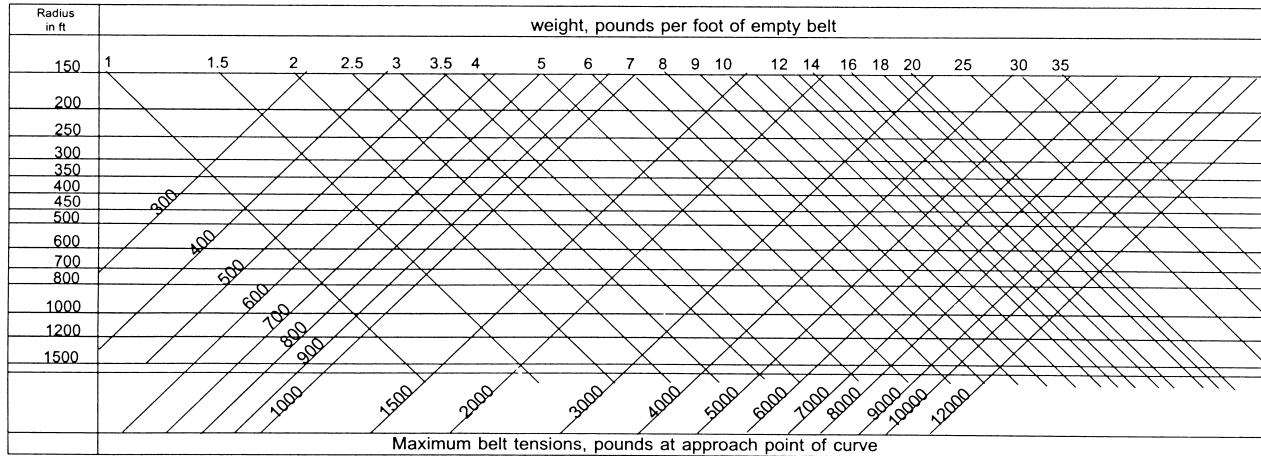


Chart is based on gradual acceleration of belt when starting the belt conveyor.

Figure 9.3 Recommended minimum radii for concave vertical belt conveyor curves.

To insure that the belt tension is sufficiently high to avoid zero tension in the belt edges at a concave curve, a check of the curve radius should be made by the use of the following formula for fabric constructions:

$$\text{Minimum radius, } r_1 = \frac{(\text{Factor A})(b^2)(B_m)(p)}{T_c - 30b} \quad (2)$$

For steel-cable belts, however, this radius can be reduced to permit a controlled buckling which experience has shown neither harms this type of belt or its splices nor causes excessive spillage. For steel-cable belts, use the following formula:

$$\text{Minimum radius, } r_1 = \frac{(\text{Factor A})(b^2)(B_m)(p)}{(T_c - 30b)(2.5)} \quad (3)$$

To prevent stressing the center of the belt beyond the rated tension of the belt, check the radius of the concave curve by using the following formula for both fabric and steel-cable construction:

$$\text{Minimum radius, } r_1 = \frac{(\text{Factor B})(b^2)(B_m)(p)}{T_r - T_c} \quad (4)$$

In these formulae:

r_1 = minimum radius of concave curve, ft

b = belt width, inches

p = number of plies in the belt

T_c = tension in belt at point c (or c_1), lbs

T_r = rated belt tension, lbs

B_m = modulus of elasticity of the conveyor belt, lbs per inch width per ply

Belt moduli vary widely among belt manufacturers because of the different fabric types used and the various ways of processing the fabrics and building the belt carcass. The modulus values calculated from the table below may vary considerably from

specific values given by manufacturers but, in most cases, they will be conservatively high and can be used for preliminary or estimating work.

Table 9-1. Belt modulus values.

Longitudinal or Warp Reinforcement	B_m Approximate Belt Modulus
Cotton	50 times rated tension
Nylon	70 times rated tension
Polyester	100 times rated tension
Rayon	100 times rated tension
Steel cable	400 times rated tension
(Rated tension is rating in pounds per inch width per ply)	

For the final design, accurate values should be obtained.

Factor A and Factor B depend upon the trough angle of the belt conveyor carrying idlers, as indicated below:

Table 9-2. Belt conveyor carrying idler trough angles.

Trough angle	Factor A	Factor B
20°	0.0063	0.0032
35°	0.0106	0.0053
45°	0.0131	0.0065

Formulae (2) and (3) are used for avoiding zero tension in the belt edges, and should be applied to the operating empty belt.

Formula (4) is used to prevent stressing the center of the belt beyond its rated tension. It should be applied to the condition where the belt is loaded from the tail pulley to the start of the curve and power is employed to start the belt from rest. Under starting conditions the allowable rated tension of the belt may be increased. See “Starting and Stopping Maximum Tensions” in Chapter 6, page 113.

Use the largest of the three radii calculated in formulas (1), (2) or (3), and (4) above. (See “Example Problem of Concave Vertical Curve,” page 248.) If formula (2) or (3) governs, investigate the possibility of increasing T_c by providing additional takeup weight.

Calculation of T_c Tension

The T_c belt tension can be determined by making the necessary additions to the tail tension, T_t , or subtractions from the head pulley tension, T_h . Refer to Chapter 6, “Belt Tension at Any Point, X , on Conveyor Length,” page 117, and Problem 4 in Chapter 6.

The decision to work forward from T_t , or to work backward from T_h , depends upon the complexity of the path of the conveyor belt from these points to point c (c_1 for the first approximation).

For the purposes of illustration, the following works forward from T_t . The value of the tail tension, T_t , can be computed for various conveyor configurations having

concave vertical curves from the formulae given in Chapter 6, associated with Figures 6.8B, 6.9C, 6.10C, 6.11B, 6.12C, 6.13B, 6.14B, 6.15C, and 6.16B.

Having obtained T_t , tension T_c then is determined as follows:

$$T_c = T_t + L_c[K_t(K_x + K_{y1}W_b) = K_{y1}W_m] \pm H_c(W_b + W_m)$$

Where:

T_c = belt tension, lbs, at point c (or c_1)

T_t = belt tension, lbs, at tail pulley

L_c = length of belt, ft, from tail pulley to point c (or c_1)

K_t = temperature correction factor (see Chapter 6, "Factor K_t ," Figure 6.1)

K_x = idler friction factor (see Chapter 6, " K_x —idler friction factor," pages 90-92)

$K_{y1} = K_y$ factor for the particular belt path from the tail pulley to point c (or c_1) (refer to Chapter 6, " K_y —Factor for calculating the force of belt and load flexure over the idlers," Tables 6-1 and 6-2)

W_b = weight of belt, lbs per ft

W_m = weight of material, lbs per ft

H_c = vertical distance, ft, if any, from the tail pulley to the point c (or c_1)

The formula above covers the condition where the belt is most likely to lift while running. When H_c is positive, this occurs when the belt is loaded from the tail pulley to point c and is empty forward of point c (i.e., there is no load forward of point c to hold the belt down on the troughing idlers). When H_c is negative, this could occur when the belt is empty.

Calculation of T_{ac} Tension at point c during acceleration

The effect of acceleration of the belt conveyor when starting from rest must be considered, as the tension in the belt at point c will be increased over the running tension T_c .

To prevent the belt from lifting from the idlers during acceleration at the start-up, it is necessary to calculate the acceleration forces and determine the total belt tension at the beginning of the curve. Refer to Chapter 6 for the effect of acceleration.

Where motors with higher than required horsepower are used, care should be taken in the calculation of acceleration forces to prevent underestimating the tension force in the belt at point c . If this is not done, the conveyor belt may lift off the idlers.

$$T_{ac} = T_c + T_a$$

Where:

T_{ac} = total tension, lbs, at point c during acceleration

T_c = tension, lbs, at point c during normal running

T_a = tension, lbs, induced in the belt by accelerating forces at any given point (in this instance, at point c)

The accelerating force at any point on the conveyor is in direct proportion to the mass being accelerated. Since the mass is the weight divided by the gravity acceleration, the accelerating force is also in direct proportion to the weights accelerated. Therefore,

$$T_a = F_a \left(\frac{W_c}{W_t} \right)$$

Where:

T_a = tension, lbs, induced in the belt by accelerating forces at any given point

F_a = total accelerating force, lbs, for concave vertical curve calculations, conveyor loaded from tail to point c only

W_c = total weight, lbs, to be accelerated by the belt at point c

$$= L W_b + W_{ri} N_{ri} + L_c \left(\frac{W_{ti}}{S_i} \right) + L_c (W_b + W_m) \\ + \text{equivalent pulley weights, lbs}$$

W_{ri} = equivalent weight, lbs, of moving parts of a single return idler

N_{ri} = number of return idlers

W_{ti} = equivalent weight, lbs, of the moving parts of a single troughing idler

S_i = troughing idler spacing, ft

L = total centers length, ft, of conveyor

L_c = length of conveyor, ft, from tail pulley to point c

L_2 = $L - L_c$ = remaining length, ft, of the conveyor from point c forward

W_b = weight of belt, lbs per ft

W_m = weight of material, lbs per ft

W_t = total equivalent weight in pounds of all moving conveyor parts, excluding drive and drive pulley, plus loaded portion from tail to point

$$c = W_c + L_2 W_b + L_2 \left(\frac{W_{ti}}{S_i} \right)$$

Like the formula for T_c , the above formulas apply to that condition where the belt is loaded from the tail pulley to point c , and where there is no load from point c to the terminal pulley. When the takeup is not near the discharge, the effect of the length of return run belt and the effect of the number of return run idlers should be reduced accordingly.

When the minimum radius has been calculated, based upon point c (or c_1 for the first approximation), the location of point c can be determined from the chart in Figure 9.4.

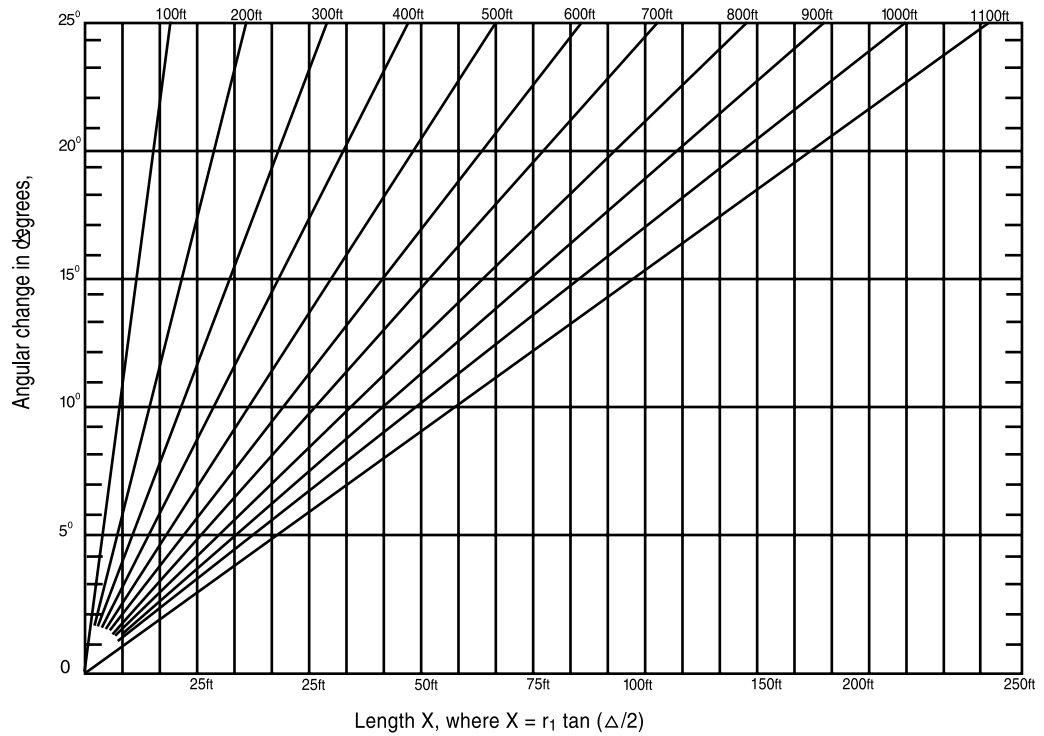


Figure 9.4 Length X for concave vertical curves.

Problem:
Determining the
Minimum Radius
of a Concave
Vertical Curve

To illustrate the method of determining minimum radius of the curve, the following problem is offered. (This is the same as Problem 4 in Chapter 6.) The profile is as indicated in Figure 9.5.

Conveyor Specifications:

Belt width = 36 inches, 7 ply, MP 70 nylon
 Belt modulus = $B_m = 4,900$ lbs per inch width per ply
 Length = $L = 4,000$ ft
 Belt weight = $W_b = 10$ lbs per ft
 Capacity = $Q = 800$ tph of material weighing 85 lbs per cu ft
 Weight of material = $W_m = 66.6$ lbs per ft
 Speed = $V = 400$ fpm
 Idlers = Class C6, 6-inch dia., 20° trough



Figure 9.5 Profile of concave vertical curve.

Weight of the Moving Parts of the Idlers (refer to Tables 5.13 and 5.14):

Troughing = $W_{ti} = 43.6$ lbs

Return = $W_{ri} = 37.6$ lbs

Idler spacing, troughing = $S_i = 4$ ft, return = 10 ft

$K_x = 0.427$

$K_y = 0.0255$ for 3,000-ft horizontal section

$K_y = 0.016$ for 800-ft inclined section

$K_y = 0.016$ for 200-ft horizontal section

From Problem 4, Chapter 6, $T_t = 1,287$ lbs and $T_{fcx} = 7,141$ lbs

T_c (tension at curve) = $1,287 + 7,141 = 8,428$ lbs

Therefore:

$$r_1 = \frac{1.11 T_c}{W_b} = \frac{(1.11)(8,428)}{10} = 936 \text{ ft}$$

Now, check against belt lifting during acceleration at the start:

$$W_c = L W_b + W_{ri} N_{ri} + L_c \left(\frac{W_{ti}}{S_i} \right) + L_c (W_b + W_m) + \text{pulley weight equivalents}$$

Inspection of the profile shows that six non-driving pulleys must be accelerated. Assume that these pulleys weigh 3,600 lbs, as in Problem 4, Chapter 6. Then,

$$\begin{aligned} W_c &= L W_b + W_{ri} N_{ri} + L_c \left(\frac{W_{ti}}{S_i} \right) + L_c (W_b + W_m) + \text{pulley weight equivalents} \\ &= (4,000)(10) + (37.6) \left(\frac{4,000}{10} \right) + 3,000 \left(\frac{43.6}{4} \right) + 3,000(10 + 66.6) + 3,600 \\ &= 40,000 + 15,040 + 32,700 + 229,800 + 3,600 \\ &= 321,140 \text{ lbs} \\ W_t &= W_c + L_2 W_b + L_2 \left(\frac{W_{ti}}{S_i} \right) = 321,140 + (1,000)(10) + 1,000 \left(\frac{43.6}{4} \right) \\ &= 342,040 \text{ lbs} \end{aligned}$$

The accelerating force, F_a , may be determined by assuming that the motor and controls will deliver an average accelerating torque of 180 percent of full load torque of the two motors, as stated in Problem 4, and the drive efficiency will be .94. See Table 6- 11.

Then, the accelerating force at the belt line becomes:

$$\frac{\text{Effective horsepower} \times 33,000}{V} = \frac{(1.8)(200)(0.94)(33,000)}{400} = 27,918 \text{ lbs.}$$

Again referring to Problem 4, Chapter 6, the effective tension when the conveyor is running fully loaded is $T_e = 14,055$ lbs. To determine T_e when only the horizontal portion is loaded, deduct resistance of load moving over inclined and upper horizontal sections and resistance to lift load.

$$\begin{aligned} &= (L - L_c)K_y W_m \pm H W_m \\ &= (1,000)(.016)(66.6) + 70(66.6) \\ &= 1,066 + 4,662 \\ &= 5,728 \text{ lbs} \end{aligned}$$

Therefore:

T_e , for the conveyor loaded only on the horizontal portion, is $14,055 - 5,728 = 8,327$ lbs.

The total equivalent force acting at the belt line and available for accelerating is $27,918 - 8,327 = 19,591$ lbs.

However, a portion of this will be necessary to overcome the inertia of the drive. This effect can be compensated for by converting the WK^2 of the drive to the equivalent weight at the belt line, and adding it to W_t . Referring to Problem 4, in Chapter 6, this equivalent weight of the drive is 55,615 lbs (see page 161).

Because the accelerating force is directly proportional to the total weight being accelerated, the actual accelerating force available to accelerate the conveyor then is calculated as:

$$F_a = 19,591 \left(\frac{342,040}{342,000 + 55,615} \right) = 16,851 \text{ lbs}$$

and:

$$T_a = F_a \left(\frac{W_c}{W_t} \right) = 16,851 \left(\frac{321,140}{342,040} \right) = 15,821 \text{ lbs.}$$

Therefore:

$$T_{ac} = T_c + T_a = 8,428 + 15,821 = 24,249 \text{ lbs.}$$

The minimum radius to prevent the belt from lifting during the calculated acceleration of starting the conveyor (loaded only from the tail to point c_1) may be found by substituting T_{ac} for T_c in the formula for the minimum radius:

$$r_1 = \frac{1.11 T_{ac}}{W_b} = \frac{(1.11)(24,249)}{10} = 2,692 \text{ ft.}$$

If a more accurate determination is required, recalculate the radius, based on the new T_{ac} , for the exact location of point c .

Checking for belt buckling during running conditions, and for overstressing the center of the belt during starting conditions, with the conveyor loaded only from the tail end to point c (or c_I), the following two conditions can occur.

For buckling of the belt, T_c for empty belt = $8,428 - (.0255)(3,000)(66.6) = 3,333$ lbs.

Therefore:

$$r_1 = \frac{(\text{Factor A})b^2(B_m)(p)}{T_c - 30b} = \frac{0.0063(36)^2(4,900)(7)}{3,333 - (30)(36)} = 124 \text{ ft}$$

For overstressing the center of the belt,

$$r_1 = \frac{(\text{Factor B})b^2(B_m)(p)}{T_r - T_c} = \frac{(0.0032)(36)^2(4,900)(7)}{31,752 - 24,249} = 19 \text{ ft}$$

It is assumed that, for the operating conditions in this example, the approximate value of $T_r = 17,640 \times 1.8 = 31,752$ lbs.

Therefore, the minimum concave radius requirement is 2,692 feet.

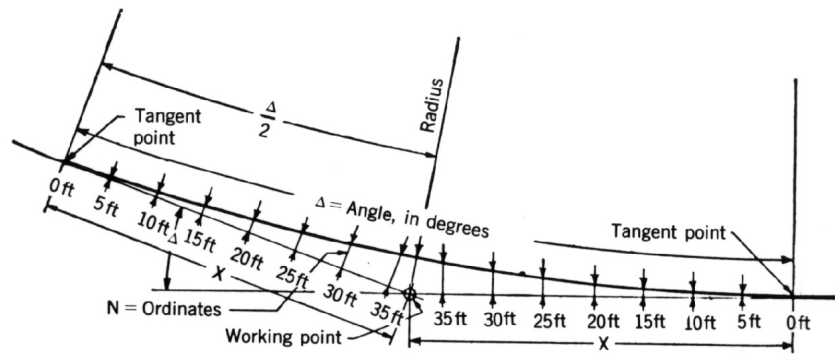
Graphical Construction of Concave Vertical Curve

After the minimum radius has been calculated and point c located, the concave curve can be graphically constructed, as indicated in Figure 9.6 and in Tables 9-3 and 9-4.

Example

The radius of the curve decided upon is 300 feet and the angle Δ is 20 degrees. After locating the working point, which is the intersection of the horizontal and inclined runs of the conveyor if extended to meet, the tangent points of the curve will be found to be 52 feet 10¾ inches (Dim, "X", Table 9-3) from the working point. By laying off points, starting from each tangent point, 5 feet, 0 inches apart towards the working point and then drawing ordinates through these points at 90 degrees from the lines representing the continuation of the horizontal and inclined conveyor runs and then measuring off the distance given in Table 9-4 on each respective ordinate, the curve may then be drawn through these points.

Figure 9.6 Method of plotting vertical curves. After the proper radius of the vertical curve is determined, and the exact angle of inclination of the belt conveyor decided upon, the conveyor can be laid out by using Tables 9-3 and 9-4.



Precautions for the Design of Vertical Concave Curves

With the trend toward stronger fabrics and new types of belt construction, the belt conveyor designer should consider the possibility of a lighter weight belt being used as a replacement at some future date. Because such a lighter belt would require a larger minimum radius, it is wise to design for the largest radius possible, considering economics and physical space requirements.

In general, the minimum radius of the vertical concave belt conveyor curve should not be less than 150 feet.

Convex Vertical Curves

A conveyor belt is said to pass through a convex vertical curve when the center of curvature lies below the belt (See Figure 9.7). In such cases, the gravity forces of belt and of load (if present), and the belt tension itself, press the belt onto the idlers.

When a troughed conveyor belt passes around the convex curve, the tension stress present is distributed across the belt so that the belt edges, being on a larger radius, are more highly stressed than is the belt center, where the radius of curvature is less. Similarly, the troughing idlers on a convex curve are more heavily loaded by radial pressures from the belt than those idlers not on the curve. A curve of sufficiently large radius holds these extreme stresses and loads within acceptable limits.



Figure 9.7 Convex vertical curve.

Convex Vertical Curves

Table 9-3. Location of tangent points on concave vertical curves.

Δ Angle of Inclination (degrees)	Dimension "X"– Distance from Tangent Point to Working Point, in feet and inches							
	Radius (ft)							
	150	200	250	300	350	400	450	500
5	6- 6 $\frac{5}{8}$	8- 8 $\frac{3}{4}$	10-11	13- 1 $\frac{1}{4}$	15- 3 $\frac{3}{8}$	17- 5 $\frac{1}{2}$	19- 7 $\frac{3}{4}$	21-10
6	7- 10 $\frac{3}{8}$	10- 5 $\frac{3}{4}$	13- 1 $\frac{1}{4}$	15- 8 $\frac{5}{8}$	18- 4 $\frac{1}{8}$	20-11 $\frac{1}{2}$	23- 7	26- 2 $\frac{1}{2}$
7	9- 2 $\frac{1}{8}$	12- 2 $\frac{3}{4}$	15- 3 $\frac{1}{2}$	18- 4 $\frac{1}{4}$	21- 4 $\frac{7}{8}$	24- 5 $\frac{1}{2}$	27- 6 $\frac{1}{4}$	30- 7
8	10- 5 $\frac{7}{8}$	13-11 $\frac{7}{8}$	17- 5 $\frac{3}{4}$	20-11 $\frac{3}{4}$	24- 5 $\frac{3}{4}$	27-11 $\frac{5}{8}$	31- 5 $\frac{5}{8}$	34-11 $\frac{1}{2}$
9	11- 9 $\frac{5}{8}$	15- 8 $\frac{7}{8}$	19- 8 $\frac{1}{8}$	23- 7 $\frac{1}{4}$	27- 6 $\frac{1}{2}$	31- 5 $\frac{3}{4}$	35- 5	39- 4 $\frac{1}{4}$
10	13- 1 $\frac{1}{2}$	17- 6	21-10 $\frac{1}{2}$	26- 3	30- 7 $\frac{1}{2}$	35- 0	39- 4 $\frac{1}{2}$	43- 9
11	14- 5 $\frac{3}{8}$	19- 3 $\frac{1}{8}$	24- 0 $\frac{7}{8}$	28-10 $\frac{5}{8}$	33- 8 $\frac{3}{8}$	38- 6 $\frac{1}{4}$	43- 4	48- 1 $\frac{3}{4}$
12	15- 9 $\frac{1}{4}$	21- 0 $\frac{1}{4}$	26- 3 $\frac{1}{4}$	31- 6 $\frac{3}{8}$	36- 9 $\frac{1}{2}$	42- 0 $\frac{1}{2}$	47- 3 $\frac{1}{2}$	52- 6 $\frac{5}{8}$
13	17- 1 $\frac{1}{8}$	22- 9 $\frac{1}{2}$	28- 5 $\frac{3}{4}$	34- 2 $\frac{1}{4}$	39-10 $\frac{1}{2}$	45- 6 $\frac{7}{8}$	51- 3 $\frac{1}{4}$	56-11 $\frac{5}{8}$
14	18- 5	24- 6 $\frac{3}{4}$	30- 8 $\frac{3}{8}$	36-10	42-11 $\frac{3}{4}$	49- 1 $\frac{3}{8}$	55- 3	61- 4 $\frac{3}{4}$
15	19- 9	26- 4	32-11	39- 6	46- 1	52- 8	59- 2 $\frac{7}{8}$	65-10
16	21- 1	28- 1 $\frac{1}{4}$	35- 1 $\frac{5}{8}$	42- 2	49- 2 $\frac{1}{4}$	56- 2 $\frac{5}{8}$	63- 3	70- 3 $\frac{1}{4}$
17	22- 5	29-10 $\frac{3}{4}$	37- 4 $\frac{3}{8}$	44-10	52- 3 $\frac{1}{4}$	59- 9 $\frac{3}{8}$	67- 3	74- 8 $\frac{3}{4}$
18	23- 9 $\frac{1}{8}$	31- 8 $\frac{1}{8}$	39- 7 $\frac{1}{8}$	47- 6 $\frac{1}{4}$	55- 5 $\frac{1}{4}$	63- 4 $\frac{1}{4}$	71- 3 $\frac{1}{4}$	79- 2 $\frac{1}{4}$
19	25- 1 $\frac{1}{8}$	33- 5 $\frac{5}{8}$	41-10	50- 2 $\frac{1}{2}$	58- 6 $\frac{3}{4}$	66-11 $\frac{1}{4}$	75- 3 $\frac{5}{8}$	83- 8
20	26- 5 $\frac{3}{8}$	35- 3 $\frac{1}{4}$	44- 1	52-10 $\frac{3}{4}$	61- 8 $\frac{1}{2}$	70- 6 $\frac{3}{8}$	79- 4 $\frac{1}{8}$	88- 2
21	27- 9 $\frac{5}{8}$	37- 0 $\frac{3}{4}$	46- 4	55- 7 $\frac{1}{4}$	64-10 $\frac{1}{2}$	74- 1 $\frac{5}{8}$	83- 4 $\frac{7}{8}$	92- 8

Table 9-4. Ordinate distances of points on concave vertical curves.

Distance from Tangent Point (ft)	Dimension "N"– Length of Ordinates in feet and inches, at Intervals of 5 feet from Tangent Point							
	Radius (ft)							
	150	200	250	300	350	400	450	500
5	0-1	0-0 $\frac{3}{4}$	0-0 $\frac{5}{8}$	0-0 $\frac{1}{2}$				
10	0-4	0-3	0-2 $\frac{3}{8}$	0-2	0-1 $\frac{3}{4}$	0-1 $\frac{1}{2}$	0-1 $\frac{3}{8}$	0-1 $\frac{1}{4}$
15	0-9	0-6 $\frac{3}{4}$	0-5 $\frac{3}{8}$	0-4 $\frac{1}{2}$	0-3 $\frac{7}{8}$	0-3 $\frac{3}{8}$	0-3	0-2 $\frac{3}{4}$
20	1-4	1-0	0-9 $\frac{5}{8}$	0-8	0-6 $\frac{7}{8}$	0-6	0-5 $\frac{3}{8}$	0-4 $\frac{7}{8}$
25	2-1 $\frac{1}{4}$	1-6 $\frac{7}{8}$	1-3	1-0 $\frac{1}{2}$	0-10 $\frac{3}{4}$	0-9 $\frac{3}{8}$	0-8 $\frac{3}{8}$	0-7 $\frac{5}{8}$
30		2-3 $\frac{1}{8}$	1-9 $\frac{3}{4}$	1-6	1-3 $\frac{1}{2}$	1-1 $\frac{5}{8}$	1-0	0-10 $\frac{3}{4}$
35		3-1	2-5 $\frac{1}{2}$	2-0 $\frac{1}{2}$	1-9	1-6 $\frac{1}{2}$	1-4 $\frac{1}{4}$	1-2 $\frac{3}{4}$
40			3-2 $\frac{5}{8}$	2-8 $\frac{1}{8}$	2-3 $\frac{1}{2}$	2-0	1-9 $\frac{3}{8}$	1-7 $\frac{1}{4}$
45			4-1	3-4 $\frac{3}{4}$	2-10 $\frac{3}{4}$	2-6 $\frac{1}{2}$	2-3	2-0 $\frac{3}{8}$
50				4-2 $\frac{3}{8}$	3-7 $\frac{1}{8}$	3-1 $\frac{5}{8}$	2-9 $\frac{3}{8}$	2-6 $\frac{1}{8}$
55				5-1	4-4 $\frac{1}{4}$	3-9 $\frac{1}{2}$	3-4 $\frac{1}{2}$	3-0 $\frac{1}{2}$
60					5-2 $\frac{1}{4}$	4-6 $\frac{3}{8}$	4-0 $\frac{1}{4}$	3-7 $\frac{3}{8}$
65						5-3 $\frac{3}{4}$	4-8 $\frac{3}{4}$	4-3
70						6-2	5-5 $\frac{3}{4}$	4-11 $\frac{1}{8}$
75							6-3 $\frac{3}{8}$	5-7 $\frac{7}{8}$
80							7-2	6-5 $\frac{3}{8}$
85								7-3 $\frac{3}{8}$
90								8-2 $\frac{1}{8}$

If a convex vertical curve is located where the belt tension is low, the distribution of stress across the belt may result in less than zero tensile stress at the center of the belt. This can produce buckling in the belt and possible spillage of the load.

Design of Convex Vertical Curves

The following equations are used to determine the minimum radius to use to prevent undesirable conditions such as belt buckling and load spillage:

$$\text{Minimum radius, } r_2 = \frac{(\text{Factor C})b^2(B_m)(p)}{T_r - T_c} \quad (5)$$

(to prevent overstress of belt edges)

$$\text{Minimum radius, } r_2 = \frac{(\text{Factor D})b^2(B_m)(p)}{T_c - 30b} \quad (6)$$

(to prevent buckling of the belt)

$$\text{Minimum radius, } r_2 = 12\left(\frac{b}{12}\right) \quad (7)$$

Where:

r_2 = minimum radius of convex curve, ft

b = belt width, inches

p = number of plies in the belt

T_c = tension in the belt at point c (or c_1), lbs

T_r = rated belt tension, lbs

B_m = modulus of elasticity of the belt, lbs per inch width per ply. For values of B_m , see discussion of concave vertical curve design, page 243.

Factor C and Factor D depend upon the trough angle of the carrying idlers, as indicated below:

Table 9-5. Trough angle of the carrying idlers.

Trough angle	Factor C	Factor D
20°	0.0063	0.0032
35°	0.0106	0.0053
45°	0.0131	0.0065

Equation (5) should be applied to the condition where the belt is being started from rest, with the belt loaded from the tail pulley to the convex curve. Under starting conditions, the allowable rated tension of the belt may be increased. See Chapter 6, “Starting and Stopping Maximum Tension,” (page 113).

Equation (6) should be applied to the condition where the belt is operating empty.

Always use the largest of the three values of the minimum convex curve radii determined by formulae (5), (6), and (7), above. See the problem, “Determining Minimum Radius of a Convex Vertical Curve,” page 256. If formula (6) governs, investigate the possibility of increasing T_c by providing additional takeup weight.

Idler Spacing On Convex Curves

Both the carrying and return idlers should be spaced so that the sum of the belt load, plus the material load, plus the radial resultant of the belt tension does not exceed the load capacity of the idlers.

The radial resultant of the belt tension can be calculated approximately as follows:

$$F_r = 2T_c \sin\left(\frac{\Delta}{2n}\right)$$

Where:

F_r = resultant force, lbs, on idlers at convex vertical curve, produced by the belt tension at the curve

T_c = tension in belt, lbs, at point c or c_l

Δ = change in the angle of the belt, degrees, between entering and leaving the curve

n = number of spaces between the idlers on the curve (must be an integral number)

$$\text{Arc length of curve} = 2\pi r_2 \left(\frac{\Delta}{360} \right), \text{ ft}$$

Troughing Idler Spacing on Convex Curves. The troughing idler spacing on a convex curve can be determined in the following manner:

$$\text{Maximum troughing idler spacing, } S_{ic} = \frac{(I_{lr} - F_r)}{(W_b + W_m)}$$

Where:

S_{ic} = maximum troughing idler spacing, ft, on the curve

I_{lr} = allowable load per troughing idler (i.e., troughing idler load rating, lbs); see Chapter 5

F_r = resultant force, lbs, on idlers at convex vertical curve, produced by belt tension at curve

W_b = weight of belt, lbs per ft

W_m = weight of material, lbs per ft

The above formula for maximum troughing idler spacing on the curve is subject to the following three conditions: (1) If the formula results in a troughing idler spacing on the curve greater than the normal idler spacing adjacent to the curve, S_{ic} is limited to values no greater than the normal troughing idler spacing. (For normal idler spacing, see Chapter 5, “Idler spacing,” page 64). (2) If the formula results in a troughing idler spacing greater than one-half of the normal idler spacing adjacent to

the curve, but less than such normal idler spacing, S_{ic} is limited to values no greater than the value given by the formula. (3) If the formula results in a troughing idler spacing less than one-half of the normal idler spacing adjacent to the curve, S_{ic} is limited to no less than one-half normal idler spacing adjacent to the curve. Solve for a new F_r . If possible, increase the radius of the curve to that based on this new F_r value.

There is also a practical limitation in determining the S_{ic} value. The idler spacing on the curve should be in integral and equal increments to simplify structural frame details. This further limits the actual value of S_{ic} .

If the length of arc of the curve (arc) is given in feet,

$$N = \frac{(\text{arc})}{S_{ic}}$$

Where:

n = number of spaces between idlers on the curve. Use the next largest integer.

*Problem.
Determining
Minimum Radius
of a Convex
Vertical Curve*

To illustrate the method of determining the minimum radius of a convex curve and the troughing idler spacing, the following problem is offered. (This is the same as Problem 4 in Chapter 6.) A profile of the conveyor is shown in Figure 9.5.

Conveyor Specifications:

Belt width = 36 inches, 7-ply, MP 70 nylon
Belt modulus, $B_m = 4,900$ lbs per inch width, per ply
Belt weight, $W_b = 10$ lbs per ft
 $T_r = (b)(p)(70) = (36)(7)(70) = 17,640$ lbs
Capacity, $Q = 800$ tph
Speed, $V = 400$ fpm
Material weight, $W_m = 66.6$ lbs per ft
Idlers = Class C6, 6-inch diameter, 20° trough,
Idler spacing, $S_i = 4$ ft
Maximum allowable idler load, $I_{lr} = 900$ lbs
Tension at curve, $T_c = 15,112$ lbs (see Problem 4, Chapter 6)
Assume 30,892 lbs during acceleration.

Using Equation (5) during acceleration of the belt:

$$\begin{aligned} r_2 &= \frac{(\text{Factor C})b(B_m)(p)}{T_r - T_c} \\ &= \frac{(0.0063)(36)^2(4,900)(7)}{(17,640 \times 1.8) - (30,892)} \\ &= 326 \text{ ft} \end{aligned}$$

Using Equation (6) when belt is running empty and $T_c = 4,388$ lbs:

$$\begin{aligned} r_2 &= \frac{(\text{Factor D})b^2(B_m)(p)}{T_c - 30b} \\ &= \frac{(0.0032)(36)^2(4,900)(7)}{4,388 - (30)(36)} \\ &= 43 \text{ ft} \end{aligned}$$

Using Equation (7):

$$r_2 = 12\left(\frac{b}{12}\right) = 36 \text{ ft}$$

Because Equation (5) yields the largest minimum radius, use 326 feet for the minimum radius of the convex curve.

Length of arc of curve in feet:

$$\text{arc} = 2\pi r\left(\frac{\Delta}{360}\right) = 2\pi(326)\left(\frac{5}{360}\right) = 28.4 \text{ ft}$$

Number of spaces between idlers:

$$n = \frac{\text{arc}}{S_i} = \frac{28.4}{4} = 7.10$$

however, the next greater integer = 8

Resultant idler load:

$$\begin{aligned} F_r &= 2T_c \sin\left(\frac{\Delta}{2n}\right)^\circ = (2)(15,112) \sin\left(\frac{5}{16}\right)^\circ \\ &= (2)(15,112) \sin .3125^\circ = (2)(15,112)(.00545) \\ &= 165 \text{ lbs (in round numbers)} \end{aligned}$$

Troughing idler spacing:

$$\text{Maximum } S_i = \frac{(I_{lr} - F_r)}{(W_b + W_m)} = \frac{(900 - 165)}{(10 + 66.6)} = 9.5 \text{ ft maximum}$$

The limitation for troughing idler spacing on convex curves, page 256, applies. The normal idler spacing adjacent to the curve is 4 feet. Therefore, the 4-foot idler spacing on the curve will be maintained.

Return Idler Spacing on Convex Curves. The spacing of return idlers can be determined similarly to the method used for troughing idlers. Use the resultant return idler load plus belt weight and then compare this value with the allowable load rating table in Chapter 5.

**Use of Bend
Pulleys for
Convex Curves**

A convex curve employing troughing idlers is recommended for all installations where space will permit for two reasons. First, the belt edge stress in a troughed belt is reduced by a properly designed convex curve. Second, there is less disturbance of the material on the belt as it passes through the change in belt profile, thereby reducing wear on the belt and idlers and preventing spillage over the edges of the troughed belt.

Bend pulleys on the carrying runs of troughed belts, in place of convex curves, are not generally recommended. A bend pulley should be used only in special cases, when space will not permit a properly designed convex curve and the belt conveyor is not sufficiently loaded to cause spillage of material over the edges of the flattened belt as it passes over the bend pulley.

Under these conditions, the diameter of the bend pulley should be large enough to insure retention of the material on the belt as the belt changes direction. The diameter required varies with the cosine Δ (angle of change in direction) and V^2 (square of the belt speed). This becomes quite large for belt speeds greater than 500 fpm. Naturally, this is another reason why troughing idlers are preferable.

The minimum diameter of the bend pulley, for a given belt velocity or speed, should be as listed in Table 9-6 below:

Table 9-6. Minimum bend pulley diameter.

Minimum diameter of bend pulley (inches)	Belt velocity or belt speed (fpm)
16	200
20	300
36	400
54	500

In no case should the diameter be less than the minimum value shown in Tables 7-13, 7-14, and 7-15 in Chapter 7.

A Guide to Steep Angle Conveying

Incline Limitations with Conventional Conveyors

Molded Cleat Belts

Large Cleat/Fin Type Belts

Pocket Belts

Totally Enclosed Belts

Sandwich Belts

Over the years a wide range of approaches to enhance the ability of belt conveyors to convey bulk material up steep inclines have evolved. Concerns about material degradation and conformity to terrain have opened up a market for belt conveyors that can lower material down steep inclines as well. This increased ability along with the already very adaptable path that they can now negotiate makes belt conveyors the most versatile method of moving bulk material between the required loading and unloading points.

For the approaches discussed below, nearly all of the material in previous chapters is totally relevant. Allowable material cross-sections and lump size restrictions differ from standard belt conveyor practice for all these approaches. For some the method of tension calculation must also be modified.

Incline Limitations with Conventional Conveyors

In Chapter 3, Table 3-3 lists maximum incline limits that conventional troughed conveyors with smooth surfaced belts can safely convey various bulk materials. These maximum angles generally range from 10-18 degrees depending on the bulk material. Beyond these recommended angles of incline material will want to:

- slide down the belt en masse
- slide internally on top of itself
- lumps will roll down the belt
- lumps will roll down over top of the fines



Figure 10.1 Example of molded chevron cleated belt.

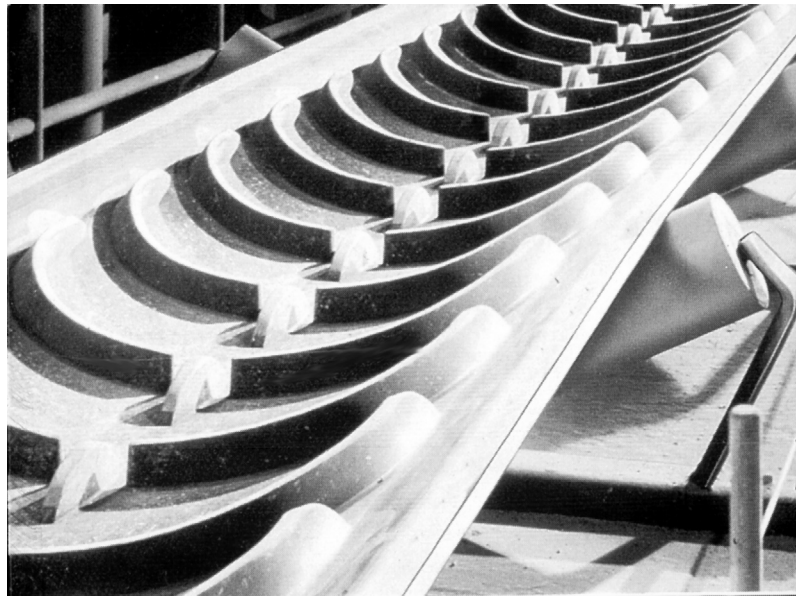


Figure 10.2 Example of molded “U”-shaped cleat belt.

Characteristics of bulk materials such as density, lump size, lump shape, internal angle of friction and coefficient of friction between the material and belt top cover are all factors that dictate the maximum incline angle up which material can be conveyed by a standard smooth covered belt conveyor. Rubber covers generally allow a greater angle of incline than PVC covers.

Light, fine, dry materials, such as sand or grain, can slide easily over a smooth covered belt as the individual particles are very small and do not indent into the rubber cover. Only the coefficient of friction between the belt and material limits the incline in this case. Material that is heavy and consists of large irregular lumps allows increased angles of incline. The edges of the lumps tend to indent into the belt cover and a mechanical holding action takes place. Mechanical interlocking of adjacent lumps also assists in increasing the allowable angle of incline. Most conveyors, however, handle material that is a combination of lumps and fines. The fines migrate to the belt surface as the belt runs, sometimes lowering the allowable angle of incline. In other instances however the fines can fill the voids between regular sized lumps, reducing their tendency to roll back down the belt. Small, uniform, spherical material such as iron ore pellets present a problem in that they can negotiate a given angle of incline when continuously fed, but the trailing end of a load will not negotiate the incline, presenting severe rollback problems.

Molded Cleat Belts

The angle at which dry fines will slide is generally less than the angle at which irregular lumps will roll. Simply making the belt cover irregular will therefore increase its ability to carry materials up slightly greater inclines. Most belt manufacturers are capable of molding patterns into the top cover of the belt. The most common patterns are a chevron or “U” shaped cleat, approximately $\frac{1}{4}$ inch to $1\frac{3}{8}$ inches high. Figures 10-1 and 10-2 show two examples. The patterns can either span the full

width of the belt, or consist of 6-inch wide shapes, 8 inches to 10 inches apart and alternately spaced across the belt width. Cleat arrangements are usually designed to allow the efficient draining of wet materials, to allow troughing with standard idlers, and to minimize problems when they contact support idlers along the return run. These cleats have more effect on the tendency of the materials to slide on the belt cover than the problems of material rolling or sliding on top of itself.

For this approach all the material in previous chapters applies except the recommended angles of incline in Table 3-3 and the belt speeds, load areas, capacities, etc. of Chapter 4.

Belt manufacturers recommend for ¼ inch high chevron cleats a maximum five degree increase in the allowable angle of incline over those values provided in Table 3-3 for smooth belt covers. Deep molded cleats on the order of 1 ¾ INCHES high can provide even greater incline angles. Table 10-1 provides a general guide to the angles of incline belt manufactures advertise as attainable with deep molded cleats. The allowable angle of incline for a specific cleat design and height depends largely on the material characteristics. These handling characteristics can vary widely. It is recommended that the material be tested and a CEMA member company be consulted when determining the safe maximum angle for a specific material.

The capacities as provided in Chapter 4 must be modified for belt conveyors using these types of belts at steep angles. The material tends to conform to its surcharge angle as measured in a vertical plane. This decreases the surcharge area and therefore the capacity. At normal angles of incline, the loss in capacity is minor and not usually taken into account. At these steep angles, however, it is recommended that the maximum allowable cross-sections, as calculated by the method in Chapter 4, be reduced by either Method 1 or Method 2, described below. Method 1 has been extracted from ISO Standard 5048 First Edition 1979-09-01. Method 2 is a deep cleated belt manufacturer's recommendation.

Method 1:

a) Where the angle of incline is less than the material surcharge on the belt, then the surcharge area is reduced.

$$A_s, \text{ reduced} = A_s \times \sqrt{\frac{\cos^2 \delta - \cos^2 \alpha}{1 - \cos^2 \alpha}}$$

$$\text{Total Reduced Area} = A_s, \text{ reduced} + A_b$$

b) Where the angle of incline equals or exceeds the material surcharge angle on the belt, the surcharge area is reduced to zero and only the zero degree surcharged is used.

$$A_s = 0$$

$$\text{Total Reduced Area} = A_b$$

Where:

α = angle of surcharge, degrees

δ = angle of incline, degrees

A_s = surcharge area, in² per Chapter 4, pg. 49

A_b = base trapezoid area, in² per Chapter 4, pg. 49

Table 10-1. Guide to maximum allowable angles of incline with deep molded cleat belts.

Material	Maximum Incline Angle
Alum, fine	26-27°
Alum, lumpy	26-30°
Ammonium nitrate	25-30°
Ammonium sulphate (granular)	25-30°
Asbestos ore or rock	25-30°
Asbestos, shred	30°
Ashes, coal, dry, ½" and under	28°
Ashes, coal, dry, 3" and under	28°
Ashes, fly	25-30°
Bagasse	30°
Bark, wood, refuse	30°
Barley	25-27°
Barytes, powdered	25-27°
Bauxite, ground, dry	30°
Bauxite, mine run	30°
Bauxite, crushed 3" and under	30°
Beans, castor, whole	25°
Beet pulp, dry	30°
Beets, sugar	25°
Beets, whole	25-27°
Brewers grain, spent, dry	25-27°
Brick, hard	25-30°
Bricks	25°
Cement	35°
Cement, Portland	25-35°
Cement, Portland, aerated	20°
Cement, rock, (see limestone)	30-32°
Cement, clinker	30-35°
Chaff, lumpy	30°
Charcoal	30°
Chips, paper mill, softwood	30°
Cinders, blast furnace	28-30°
Cinders, coal	28-35°
Clay, calcined	30-32°

Table 10-1. Guide to maximum allowable angles of incline with deep molded cleat belts.

Material	Maximum Incline Angle
Clay, dry	35°
Clay, wet	30°
Clinker	35°
Coal, crushed	30°
Coal, washed	35°
Coal, briquettes	25°
Coal, bituminous, mined 50 mesh or under	30-32°
Coal, bituminous, mined, run of mine	30-32°
Coal, bituminous, mined, slack, ½" and under	30-32°
Coal, bituminous, stripping, not cleaned	30-32°
Coal, Lignite	30-32°
Coke, loose	30-32°
Coke, petroleum calcined	30-32°
Coke breeze, ¼" and under	30°
Concrete, 2" slump, wet	30-32°
Concrete, 4" slump, wet	30-32°
Concrete, 6" slump, wet	30-32°
Concrete, in place, stone	30-35°
Copper ore	30-35°
Copper ore, crushed	30-35°
Copper sulfate	25°
Corn, ear	28-30°
Corn, shelled	25-28°
Cottonseed, dry, de-linted	25°
Cottonseed, dry, not de-linted	30°
Cottonseed cake, crushed	30°
Dolomite, lumpy	30-32°
Earth, as excavated, dry	40°
Earth, wet, containing clay	40-45°
Feldspar, ½" screenings	30°
Feldspar, ½" to 3" lumps	30°
Fertilizer, pellets	25°
Fertilizer, fine grained	30°
Fluospar, ½" screenings	30°
Foundry refuse, old sand cores, etc.	30°

Table 10-1. Guide to maximum allowable angles of incline with deep molded cleat belts.

Material	Maximum Incline Angle
Glass	20-25°
Grain	20-25°
Grain, distillery, spent, dry	25-27°
Granite, ½" screenings	30-32°
Granite, 1½" to 3" lumps	30-32°
Granite, broken	30-32°
Graphite, flake	25°
Gravel	30-35°
Gravel, bank run	30°
Gravel, dry, sharp	32°
Gravel, pebbles	20°
Gravel, washed	25°
Gypsum dust, aerated	26-28°
Gypsum dust, non-aerated	28-20°
Gypsum, 1½" to 3" lumps	30°
Guano, dry	28°
Ice, crushed	23-25°
Ilmenite ore (Titanium ore)	30°
Iron ore	30-32°
Iron ore, crushed, ½"	30-32°
Iron ore, pellets	25-30°
Iron oxide, pigment	28°
Kaolin clay, 3" and under	28°
Lead ores, fine	30°
Lead oxides	30°
Lignite, air dried	30°
Lime, ground, ⅛" and under	30°
Lime, hydrated, ⅛" and under	28°
Lime, hydrated, pulverized	25-28°
Lime, pebble	28°
Limestone, crushed	30°
Limestone, dust	28-30°
Loam, mud, clay, dry	35°
Loam, mud, clay, wet	30°
Lumps, mixed to 2"	35°
Lumps, mixed to 4", sized to 3"	30°

Table 10-1. Guide to maximum allowable angles of incline with deep molded cleat belts.

Material	Maximum Incline Angle
Lumps, mixed to 6", sized to 4"	28°
Manganese ore	30-32°
Manganese sulphate	28-30°
Marble, crushed, ½" and under	30-35°
Molybdenite, ore	30°
Mud, dry	35°
Mud, wet	30°
Peanuts, in shells	30°
Phosphate, acid, fertilizer	28°
Phosphate triple super, ground fertilizer	30°
Phosphate rock, broken, dry	30°
Phosphate rock, pulverized	30°
Phosphate sand, wet	30°
Potash ore	30°
Potatoes	30°
Pyrites, iron, 2" to 3" lumps	28°
Quartz, ½" screenings	30°
Rock, crushed	30-35°
Rock, soft, excavated with shovel	30-35°
Rubber, pelletized	30-35°
Rye	30°
Salt	30-35°
Salt, common dry, coarse	30°
Salt, common dry, fine	28°
Salt cake, dry, coarse	30°
Sand, bank, amp	30-45°
Sand, bank, dry	25-36°
Sand, dry, fine	25°
Sand, foundry, prepared	30-35°
Sand, foundry, shakeout	28°
Sand, Gravel, mixed	30°
Sand, wet	35-45°
Sandstone, broken	32°
Sinter	30°
Slag, blast furnace	30°
Soap chips	20°

Table 10-1. Guide to maximum allowable angles of incline with deep molded cleat belts.

Material	Maximum Incline Angle
Soapstone, talc, lumps	30°
Soda ash, briquettes	32°
Soil	30°
Soybeans, cracked	25°
Soybean cake, over ½"	28-30°
Sugar, granulated	30°
Sugar, raw, cane	30°
Sulphur, lumpy	30-35°
Sulphur, ore	30-35°
Titanium ore	30-35°
Triple super phosphate	30°
Walnut shells, crushed	30°
Wheat	25-30°
Wood chips	30°
Zinc ore, crushed	30°
Zinc ore, roasted	30°

Method 2 is a deep-cleated belt manufacturer's recommendation. See Table 10-2 below.

Table 10-2. Steep angle conveying capacity reduction, Method 2.

Total Area Correction Factor						
Angle of Inclination	15°	20°	25°	30°	35°	40°
Slightly rolling and coarse materials	0.89	0.81	0.70	0.56	----	----
Sticky materials	1.00	0.93	0.85	0.68	0.58	0.47

The angle of incline can be maximized by minimizing any disturbing forces the material sees as it travels over the troughing idlers. Slower belt speeds and less sag than recommended in Chapter 4 and 6 should be used. A maximum belt speed of 300 to 400 fpm along with a maximum belt sag of 1 percent is generally recommended.

To minimize the risk of vibrations, the distance between the return idlers should not be a multiple of the cleat spacing. Return snub or bend pulleys are not recommended. In applications where they must be used, a large diameter pulley and low belt tensions are recommended.

The advantages of this approach are:

1. Economical method of increasing the angle of incline.
2. No additional mechanical components over a standard belt conveyor.

The disadvantages of this approach are:

1. The allowable angle of incline is limited.
2. The cleats will wear off more rapidly than a standard belt cover and the belt may lose its inclined conveying capabilities.
3. At higher belt speeds, as the cleats pass over the return rollers, vibrations can occur that cause accelerated cleat wear and a reduction in return roll bearing life.
4. The belt is more difficult to clean than standard belt conveyors. Belt brushes, belt beaters, air knives, water sprays, or a wash box must be used.

Molded cleat belts are usually restricted to short conveyors where few or no return idlers are needed and either the material does not stick to the surface or where the carryback is acceptable.

Large Cleat/Fin Type Belts

There are various fabricators who can either hot or cold bond or mechanically fasten large cleats or fins onto the rubber top cover of conveyor belts after manufacture. To allow the belt to run in a troughed position on the carrying side and a flat position for the return, these cleats/fins involve various shapes and configurations. Figure 10.3 shows one such arrangement. Note the special return supports that are required. The cleats/fins allow the conveying angle to increase substantially, up to approximately 45 degrees. The conveying capacity drops off rapidly, however, as the angle increases.

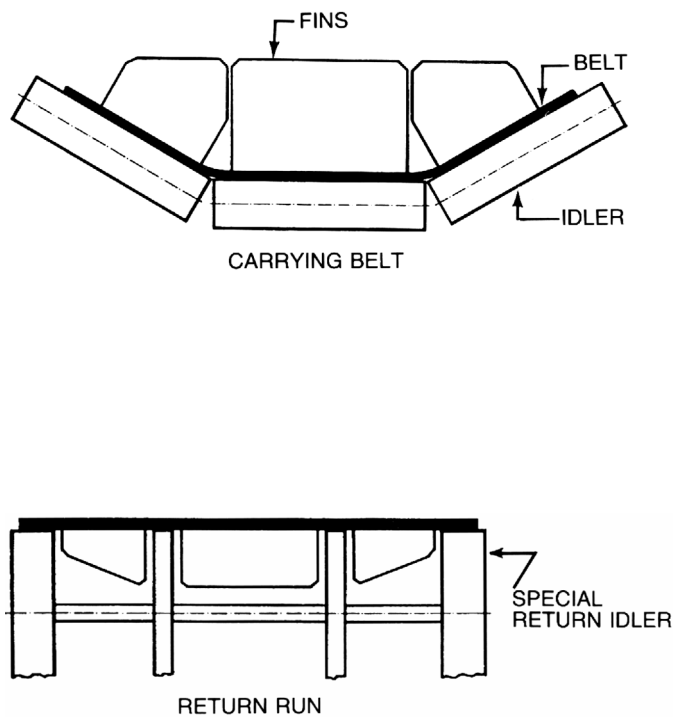


Figure 10.3 Large cleat/fin belt illustration.

All material in previous chapters is relevant to the design of these types of conveyors except the calculation of capacity. To calculate the allowable material volume per unit length on the belt, the cleat/fin height and spacing along with the material's dynamic angle of repose and the conveyor incline angle must be known. Allowing for a safe edge distance of $0.55b + 0.9$ inches or more and a safe clearance between the top of the material and the top of the cleat, the material unit volume on the belt can be calculated. It is recommended that a CEMA member company be consulted when calculating capacities.

Similar to the molded cleat design above, slow belt speeds of 300 to 400 fpm and a maximum belt sag of 1 percent are recommended.

The advantages of this approach are:

1. Economical method of increasing the angle of incline.
2. No additional mechanical components over a standard belt conveyor.

The disadvantages of this approach are:

1. The allowable angle of incline is limited.
2. The cleats or fins are susceptible to damage.

3. Special return belt supports must be designed to support the specific cleat/fin design.
4. The belt is difficult to clean. Belt beaters, air knives or water sprays must be used.

Large cleated/finned belts are usually restricted to short conveyors where few or no return idlers are needed and either the material does not stick to the surface or the carryback is acceptable.



Figure 10.4 Example of pocket belt partitions.

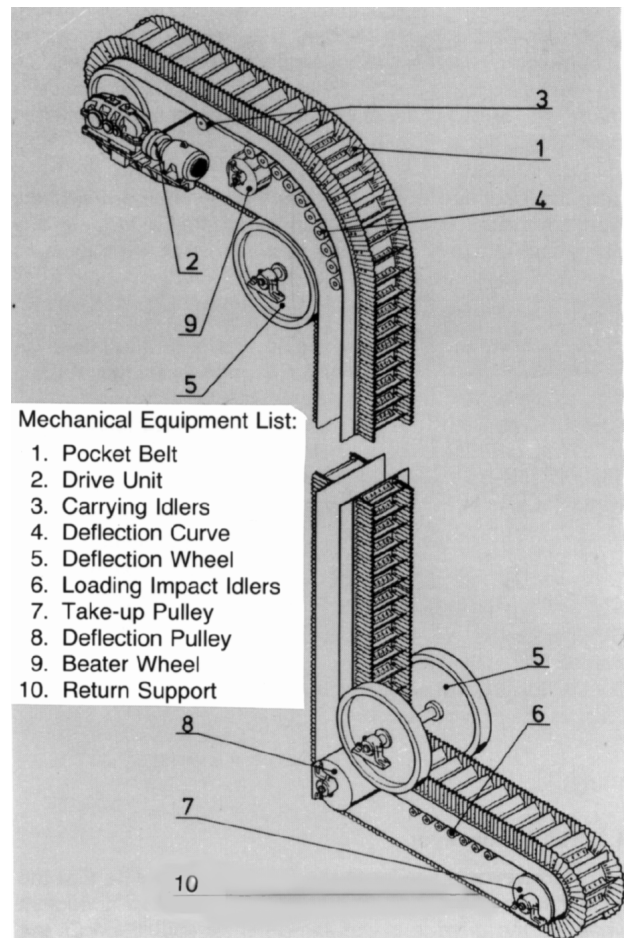


Figure 10.5 Typical pocket belt for vertical elevating.

Pocket Belts

To enhance the large cleat/fin approach, corrugated sidewalls are added to the edges of the belt to form complete rectangular partitions or “pockets” for the material to ride in as shown in Figure 10.4. These types of belts have many trade names, but

are known by the generic term “pocket belts.” The sidewalls allow the belt to convey a large cross-section of material without troughing idlers being required. The sidewalls also increase the load carrying capacity over the large cleat/fin design, which loses capacity fairly quickly as the incline increases. Most belts for this approach use a cross-rigid carcass belting to prevent bowing or sagging at deflection wheels or return supports.

Figure 10.5 illustrates the mechanical equipment required for a typical vertical system using these types of belts.

The belt, deflection wheels, and return supports are special, but many items such as the drive, tail/take-up and forward bend pulleys, flat belt idlers, belt beater, and side guide idlers are standard belt conveyor items. Figure 10.6 illustrates a typical return support.

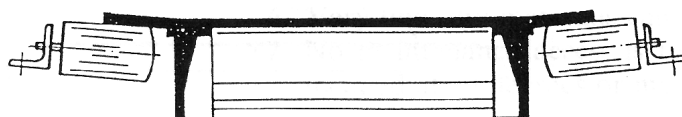


Figure 10.6 Typical return support for pocket belt.

There are numerous cleat designs. Cleat spacing is a function of lump size and angle of incline. It is not within the scope of this manual to detail the method of calculating capacities or belt tensions for this approach. Figure 10.7 does provide a general idea of how capacities are calculated using the materials surcharge angle and the cross-section of the cleat.

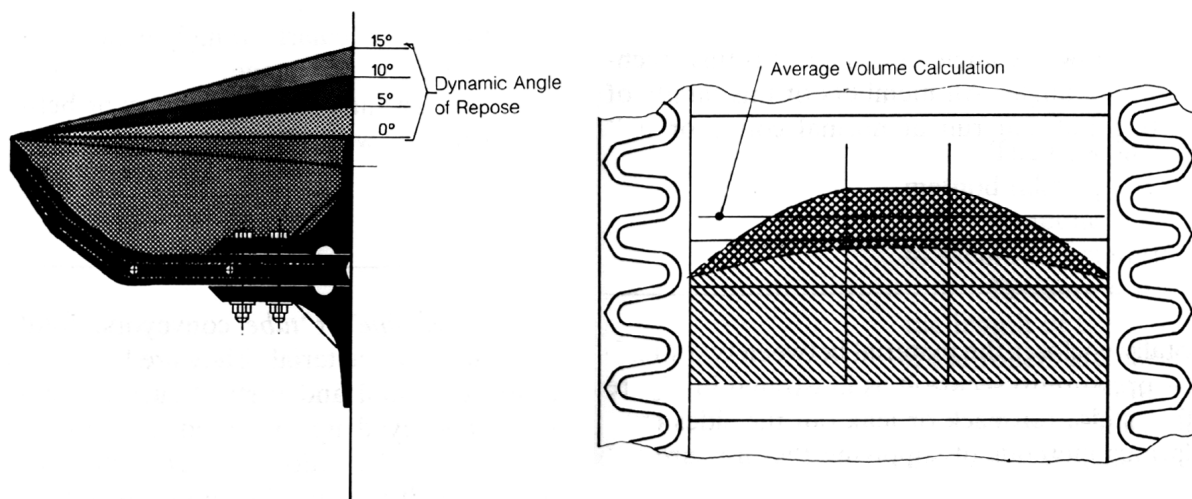


Figure 10.7 Illustrated method of calculating pocket belt capacity.

With proper care the tension calculations of Chapter 6 can be modified to handle these types of systems. It is recommended, however, that a CEMA member company be contacted for assistance in designing these systems.

The advantages of this approach are:

1. Incline angles to 90 degrees are possible.
2. Fewer mechanical components are required, particularly at steep angles of incline.
3. Medium capacity, high lifts can be accomplished.
4. Vertical units can be twisted about their vertical axis. See Figure 10.8.
5. Relatively small radii concave and convex bends can be used.
6. Very fine, dry materials can be elevated vertically.
7. No maintenance access is required up the vertical portion of units inclined at 70 degrees or more.

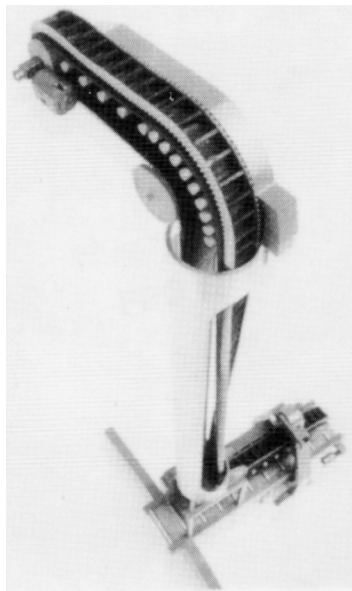


Figure 10.8 Pocket belt twisted about its vertical axis.

The disadvantages of this approach are:

1. The belt is more expensive than standard conveyor belting.
2. Lump size is somewhat limited particularly at high angles of incline.
3. Sidewalls and cleats are susceptible to damage.
4. The belt is more difficult to clean than standard conveyor belts. Belt beaters, air knives, water spray or a wash trough must be used.

Totally Enclosed Belts

A family of belt conveyors known as “pipe” or “tube” conveyors, “fold” belts, and “suspended” belts totally enclose the material. They are best known for their ability to negotiate tight horizontal and vertical curves. They also, however, allow material to be conveyed up increased angles of incline. Research has shown that generally as the ratio of material cross-sectional area to contact perimeter with the belt (technically known as the hydraulic radius) decreases, the greater the allowable angle of incline. It is this mechanism that allows the increase in the angle. All members of this family of conveyors use proprietary belting, but can run at normal conveyor belt speeds.

Sandwich Belts

The sandwich belt principal totally encloses the material between two standard conveyor belts and applies pressure to secure it as illustrated in Figure 10.9. This insures it will neither slide, roll back, or leak out the sides even when conveyed vertically. Various methods of supplying the pressure are commercially available. These methods can be used alone or in combination to best advantage.

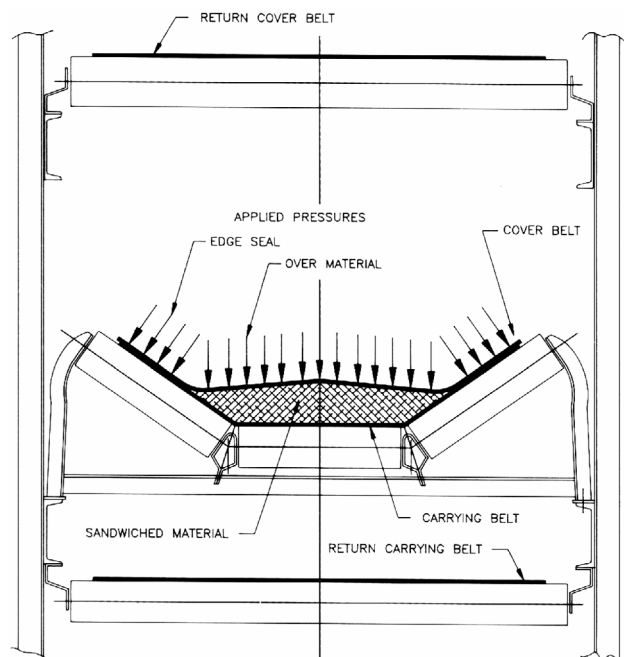


Figure 10.9 Typical sandwich belt cross section.

Simply laying a conveyor belt on the material is the simplest approach, but supplies a limited amount of pressure that quickly diminishes as the angle of incline increases. Special lead-filled belts have been used in the past to obtain the required weight, but are difficult if not impossible to obtain today.

The transverse stiffness of the belt alone can be used where the combination of angle of incline, belt width, material density, material internal friction, and material

to belt friction are appropriate. The transverse stiffness is utilized by using rollers at the edges of the belt so the belt has to deflect between them in order to allow passage of the material. To accomplish this curvature requires that a pressure be developed to overcome the belt stiffness. This pressure in turn secures the material to the extent that it can be elevated vertically.

The tension in the belt itself can be used to secure the material. This method takes the sandwich through a continuous convex curve so that the radial component of the outer belt tension supplies the required pressure to secure the material.

Pressure can be applied mechanically with rollers across and along the length of the sandwich belts. Although the pressure applied is discrete even if the rollers are closely spaced, the transverse and longitudinal stiffness of the belt will help to distribute the pressure enough to secure the material.

Belt tensions and system power are calculated with a modified CEMA method. Capacities are calculated on a reduced CEMA cross-sectional area. The reduced area is required to provide the seal zone at the sides of the sandwich. A reduction is also required due to the pressure from the cover belt reducing the materials surcharge angle. Sandwich belts are run at normal belt conveyor speeds or higher to compensate for the reduced belt capacity. Lump size, at full material cross-section for the belt, is generally limited to the material cross-sectional depth between the belts. Larger lumps can be accommodated by reducing the material load cross-section. This removes the possibility a lump will sit high on top of material cross-section and lift the cover belt so high as to open the belts at the edges causing spillage.

Almost all sandwich belts use convex curves to transition to an incline and back. The curve is inverted when going from the horizontal to an incline and it is right side up when going from an incline to the horizontal. Concave curves can be used to transition to an incline, but large curve radii and high applied pressures are required to prevent belt lift off.

For small radii vertical convex curves, nylon carcass belts are preferred due to their low modulus of elasticity. The allowable curve radius can be calculated using the method in Chapter 9 for convex curves. The allowable radius can be reduced by using special low modulus nylon carcass belting, long center roll troughing idlers, and low troughing angles, usually 20 degrees. The limitations on belt edge and center tension used in the Chapter 9 calculations may be eased to reduce the curve radius. It is recommended that a CEMA member company be contacted when this is required.

Figure 10.10 illustrates the various profiles that are available for sandwich belts. These profiles can be combined in one system in order to adapt to the required path.

Figure 10.11 illustrates a C-profile application that feeds a bin. The C-profile has found widespread use on board self-unloading ships.

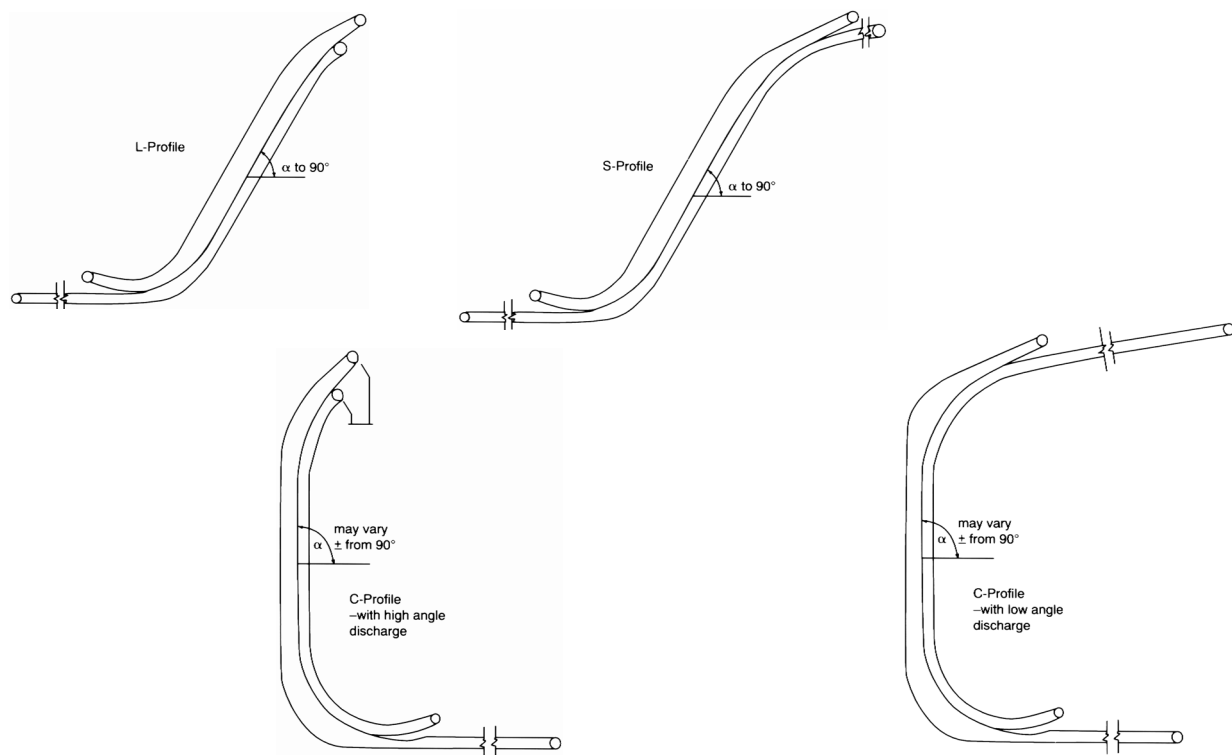


Figure 10.10 Available sandwich belt profiles.

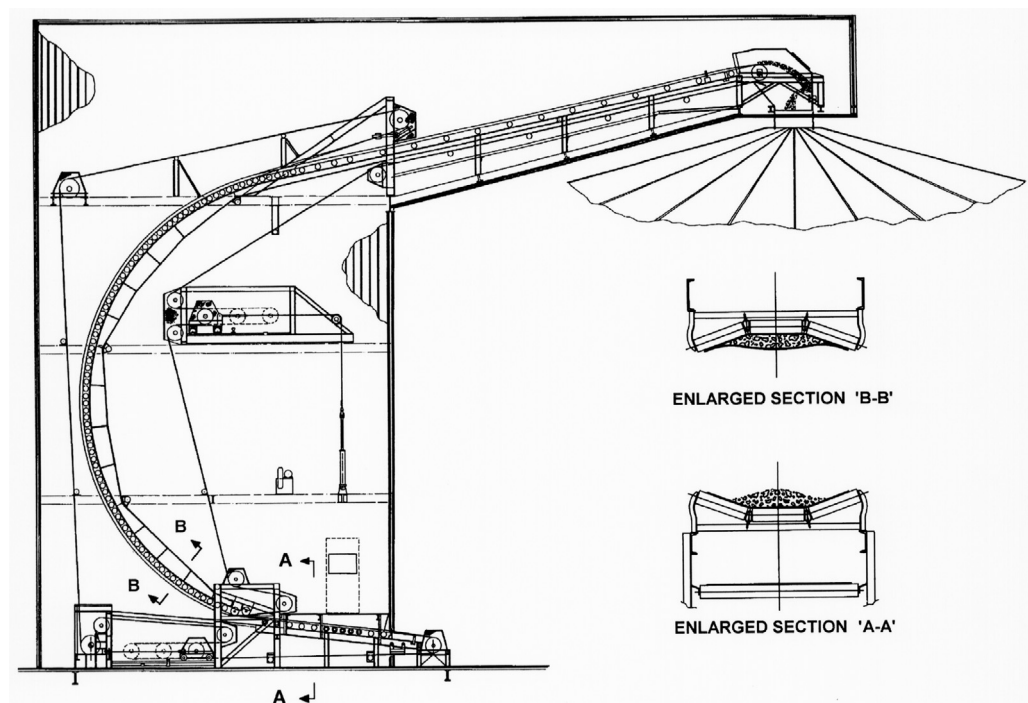


Figure 10.11 C-profile sandwich belt feeding a bin.

An illustration of a self-unloading system with a long tunnel loading zone utilizing multiple gates can be found in Figure 10.12.

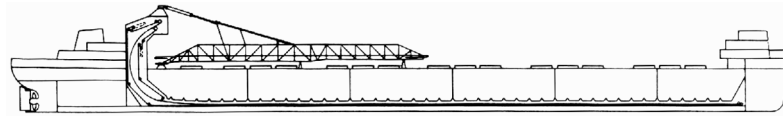


Figure 10.12 C-profile sandwich belt on board a self-unloading ship.

Figure 10.13 illustrates the components in a typical sandwich belt elevator. The type of unit illustrated uses the transverse stiffness of the belts alone to apply the required pressure to contain and elevate the material. Where conditions warrant, such as belt width and material density, spring loaded pressure rolls are added to supplement the pressure available from the belt's cross-stiffness.

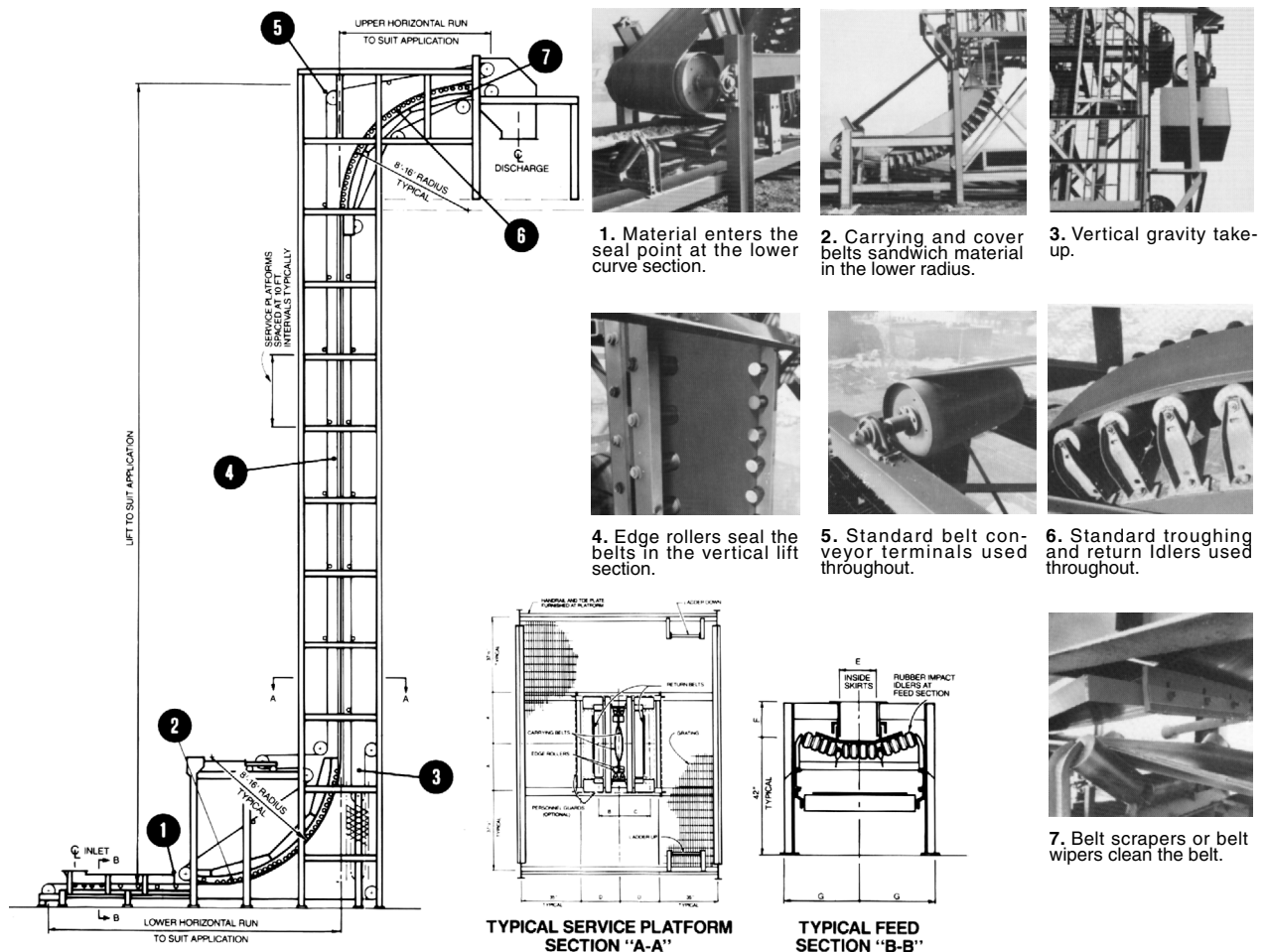


Figure 10.13 Typical sandwich belt elevator components.

Figure 10.14 illustrates the profile and cross-section of a typical sandwich belt system that utilizes closely spaced spring loaded pressure rollers and standard troughing idlers up the incline. This design of incline can be taken to 90 degrees.

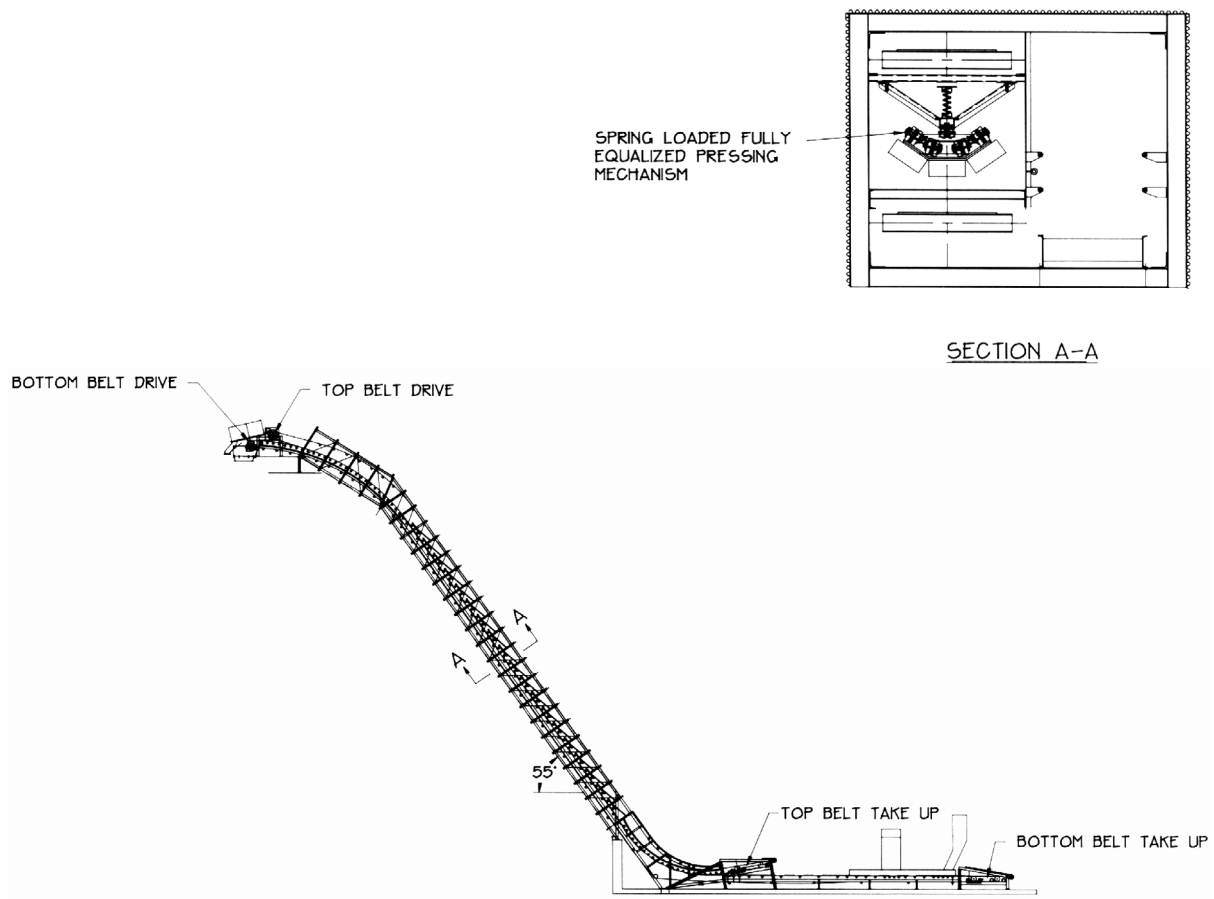


Figure 10.14 L-profile sandwich belt with spring loaded pressure rolls.

Figure 10.15 shows a system in an open pit mining application conveying 4000 tph of primary crushed copper ore up a 35.5 degree incline.



Figure 10.15 Sandwich belt in an open pit mining application.

The advantages of this approach are:

1. Incline angles to 90 degrees are possible.
2. Standard conveyor belting and components are used.
3. High capacity, high lifts can be accomplished.
4. Large lumps can be accommodated.
5. Belts can be easily cleaned using standard belt cleaners.
6. The approach can be incorporated locally in long conveyors when advantageous.

The disadvantages of this approach are:

1. More mechanical components than other types of steep angle conveyors are required.
2. Compared to pocket belt systems, relatively large radii concave or convex curves must be used, particularly for high capacity and/or high lift systems.
3. Very fine, dry bulk materials such as cement cannot be elevated effectively using this method.
4. Even units inclined at 90 degrees require maintenance access up the vertical portion.

It is highly recommended that you consult with the individual manufacturers of steep angle belt conveyors for more information and specific inquiries.

Belt Takeups, Cleaners, and Accessories

Belt takeups

Belt Stretch

Takeup Movement

Manual Takeups

Automatic Takeups

Cleaning Devices

Analyzing Carryback

Cleaners and Conveyors

The Systems Approach to Belt Cleaning

Maintaining Cleaning Pressure

Belt Cleaners and Horsepower Requirements

Belt Cleaners and Top Cover Wear

The Importance of Maintenance

Specialized Cleaning Systems

Handling the Removed Material

Belt Turnovers

Pulley Wipers

Return-run Belt Cleaning

Belt Conveyor Accessory Equipment

Weather Protection

Spillage Protection: Wing-type Pulleys

Tramp Iron Detectors

Conveyor Belt Scales

Sampling Devices

Belt Takeups

All properly designed belt conveyors require the use of some form of takeup device for the following reasons:

1. To insure the proper amount of slackside tension, T_2 , at the drive pulley to prevent belt slippage.
2. To insure proper belt tension at loading and other points along the conveyor (necessary to prevent loss of troughing contour of the belt between idlers, thus avoiding spillage of the material from the belt).
3. To compensate for changes in belt length due to stretch.
4. To allow belt storage for making replacement splices (without which storage, small sections of new belt would have to be added, requiring two splices for each splice repair).

Belt Stretch or Elongation

Any conveyor belt can be expected to have several types of stretch or elongation:

Elastic stretch. This is that part of a belt's stretch which occurs during changes in belt tension, including starting acceleration or braking deceleration. This stretch is almost entirely recovered when the applied pull or stress is removed.

Constructional stretch. This is due more to the type of fabric weave than to the textile material used. In a conventionally woven fabric, the warp strands which are crimped tend to straighten out as the load is applied. This results in belt growth, a portion of which is nonrecoverable.

Permanent length change. This includes changes in length caused by elongation in the basic fiber structure. It also includes the portions of the elastic stretch and constructional stretch which are nonrecoverable.

Takeup Movement

The required length of takeup movement depends upon several factors:

1. Type of starting or braking. Across-the-line starting or uncontrolled braking will require considerably more allowable takeup movement than controlled acceleration or deceleration.
2. Frequency of starts and stops of a fully loaded belt.
3. Whether a run-in period with metal fasteners can be utilized. If so, this would conveniently allow for removal of any nonrecoverable length change before making a final vulcanized splice.
4. Stretch and elongation characteristics of the belting being used.
5. Running tensions.

The takeup should provide sufficient movement to accommodate acceleration or deceleration surges without having the takeup strike against its stops. It should also

allow for some “live” storage of belting so that, in case of an accident, the belt can be repaired (or a damaged portion removed) without requiring two splices. In addition, takeup movement should provide for changes in belt length due to stretch or shrinkage.

Care must be taken to ensure the takeup’s even movement. It is important that the movement be balanced across the width of the belt to provide even tension on the belt and maintain belt alignment. For that reason the takeup must remain free to function without accumulations of fugitive material or damage, which could interfere with movement. Takeup pulleys that are not guided to operate horizontally are a significant source of belt misalignment.

Manual Takeups

Manual takeups have the advantages of compactness and low cost; however, because the takeup of the belt is at best periodic with this type, the resulting belt tensions are almost always too high or too low. Therefore, manual takeups are recommended only where an automatic takeup is not practical because of space limitations or in the case of relatively short, light-duty belt conveyors, where takeup considerations are not as critical.

In high-tonnage inclined conveyors, such as stackers, the use of manual takeups for any length of belt is not normally recommended. Such cases should be referred to a CEMA member.

The most commonly used manual takeup is the screw takeup, which is illustrated in Figure 11.1. Other common types include ratchet and jack-operated takeups. Screw takeups are generally available with 12, 18, 24, 30, and 36 inches of takeup travel.



Figure 11.1 Manually adjusted screw takeup assembly.

The main problem with the use of a manual takeup is that it requires a vigilant and careful operator to observe when additional takeup is required, and then adjust the takeup just to the point where the proper tension is applied to the belt. The problem is made difficult by the absence of an accurate index for judging how much tension is needed, or how much has been provided by a particular adjustment.

With new belts, it is necessary to provide a large amount of slack-side tension to prestretch the belt. This prevents slack in the belt from accumulating behind the drive pulley and causing belt slippage when the conveyor operates. With manual takeups, the operator must rely on personal judgment in making the suitable adjustment.

Manual Takeup Location

Manual takeups normally are located at the opposite end of the conveyor from the drive. This is the most convenient and least costly arrangement as it involves no extra pulleys. However, it is possible to locate manual takeups elsewhere on the return run of the belt conveyor, if required.

Automatic Takeups

Automatic takeups are the more desirable type for use on any belt conveyor. They can be installed horizontally, vertically, or on an incline. They can be either gravity-operated or power-operated by hydraulic, electric, or pneumatic means. The most common type is the gravity takeup. Other types of automatic takeups can be given consideration when special conditions, such as space limitations or portability, are involved.

Shown in Figures 11.2 and 11.3 are a horizontal automatic gravity takeup at a tail pulley and a vertical automatic gravity takeup for use near a head pulley. Figure 11.3 illustrates a vertical automatic gravity takeup on an inclined conveyor.

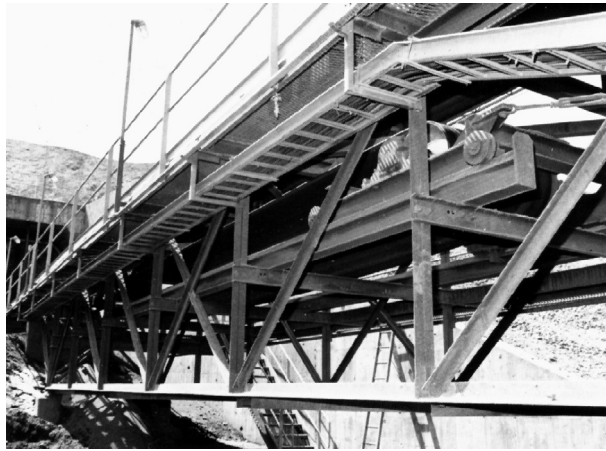


Figure 11.2 Horizontal automatic gravity takeup. Counterweight and cables not shown.

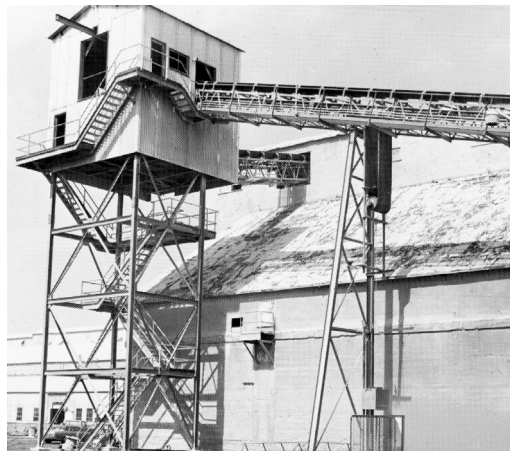


Figure 11.3 Vertical automatic gravity takeup on an inclined conveyor.

Belt Takeups

Recommended Takeup Movement

The foregoing points have been considered in making recommendations for the amount of takeup movement required and for the position of the takeup in making the final splice.

The values for takeup movement listed in Table 11-1 are generally suitable for most conveyor applications. A reduction or increase in these values will depend upon factors which include environmental and operating conditions as well as belt selection. Therefore, it is advisable to consult the manufacturer of the belt under consideration before takeup requirements are determined.

Table 11-1. Recommended takeup movement.*

Conveyor Centers (ft)	Takeup Movement (ft)			
	Multiple-ply Belt**	Reduced-ply Belt	Steel-cable Belt	Solid-woven Belt***
50 or less	1.5	1.5	--	
100	3	3	--	
200	5-6	6	--	
300	7-8	8	--	
500	10-14	14	--	
700	13-18	18	--	
1,000	18-25	25	7	
1,500	25-34	34	8	
2,000	30-40	40	10	
2,500	35-47	47	12	
3,000	39-54	54	15	
3,500	42-59	59	17	
4,000	45-64	64	20	
4,500	48-70	70	22	
5,000	50-75	75	25	
*Allows for 3/4 of the total movement for belt stretch. **The larger allowance is recommended for belts having 30-50 RMA ratings. ***Consult belting manufacturer.				

Automatic Takeup Location

Automatic takeups may be located at any place in the return run of the belt conveyor. The prime consideration is where the automatic takeup will work best in relation to the drive, to keep belt tensions at a minimum. Other considerations, such as available space, maintenance conditions, and the economics of the location, should also be taken into account.

Generally, the most inexpensive location for an automatic takeup is at the tail of an inclined conveyor. At this point, no additional pulleys will be involved. On steeply inclined conveyors, the weight of the takeup pulley assembly and belt may provide sufficient slack-side tension to prevent drive pulley slippage, without the need for additional counterweight.

On long, horizontal, or slightly inclined conveyors, the automatic takeup should be located near the drive, where it will act quickly enough to prevent slippage of the belt on the drive pulley during acceleration at startup. If the takeup is located else-

where, its movement must be calculated to be sure that it exceeds the rate at which the belt will be deposited in the takeup. Refer to Chapter 6, "Belt tensions for various profiles," for usual takeup locations on various conveyors.

Automatic Takeup Force Requirements

An automatic gravity takeup must provide a force equal to twice the required belt tension, at the place where the takeup is installed. This force usually is supplied by a counterweight composed of steel, cast iron, concrete, or some other heavy material equal to the force required. The force may be somewhat less in magnitude and multiplied appropriately by the mechanical advantage of a system of ropes and sheaves. Some means of adjustment to the weight force should be provided, as actual operating conditions may change the force requirements originally calculated in the design of the belt conveyor.

To calculate the required force of the automatic takeup or the weight force of a gravity takeup, the following formula can be used:

$$W_g = \frac{2T + W_f - W_p}{R_1}$$

Where:

W_g = required force, lbs, provided by the takeup; in gravity takeups, it is the weight force, lbs

T = belt tension, lbs, at the point where the takeup is located

W_f = force, lbs, to overcome friction of the takeup carriage, ropes, sheaves, or other frictional resistance. **Note:** When solving for T from a given W_g , W_f can work to trap takeup force in takeup system. In this case, make W_f negative.

W_p = component of weight force of takeup carriage, wheels, pulley, shaft, shaft bearings, etc., acting in direction of resultant pulley load; where elements move horizontally, W_p becomes zero

R_1 = mechanical advantage ratio, if any mechanical advantage is provided

In a hydraulically or pneumatically operated automatic takeup, the force is calculated as above. As with a gravity automatic takeup, the force should be adjustable to meet unforeseen operating conditions. This can be done by varying the hydraulic or pneumatic pressure to suit the actual operating conditions.

Cleaning Devices

Many bulk materials conveyed on belts are somewhat sticky. Portions of the material will cling to the conveying surface of the belt and not discharge with the rest of the load at the unloading point. This residual material is carried back on the return run, eventually falling off at various points along the belt line to cause housekeeping and maintenance problems. The carryback leads to excessive wear, build-up on return idlers, misalignment of the belt, and possible damage by forcing the belt against some part of the supporting structure, adversely affecting conveyor opera-

tions and plant efficiency. In addition, accumulations of material on the ground or clouds of dust in the air can present a health and safety hazard. Therefore, it is desirable to clean the belt.

To minimize the risk of carryback material being released from the belt into the general plant environment, cleaning should take place as close to the discharge point as possible without being in the trajectory of the unloading material. The point is to get the belt as clean as possible before the belt-borne material can be dislodged into the air or transferred to rolling components such as snub pulleys and return idlers.

Analyzing Carryback

The material that sticks to the belt is typically the belt's cargo in its worst state. Carryback is finer and has higher moisture content than the nominal material characteristics.

When considering the construction of new conveyors, it may be desirable to include in the design requirements a specification for cleaner performance and the amount of carryback allowed to pass the cleaner. As a conveyor is designed, its plan should spell out a specific amount of material allowed to be carried back, with a sufficient number of effective cleaners to hold carryback to this required level.

To determine how much carryback is actually present, a sample can be collected from a given portion on the belt. This sample can be collected using a putty knife to manually scrape the belt, or with a catch pan that is affixed with scraper blades. This material, when collected and weighed, can be multiplied by the schedule of operation to establish an accurate measurement of the material carried back on the belt.

By weighing the amount of material collected as an average and multiplying that weight by the width and speed of the belt, you can determine how much carryback material is on the belt.

For example, carryback of .03 ounces per square yard of loaded belt width can result in over 2.5 tons per week of fugitive material from a single conveyor belt. Here's the calculation:

$$\begin{aligned} &810 \text{ feet per minute} = 270 \text{ yards per minute} \\ &\times .03 \text{ oz per square yard} \\ &\times 60 \text{ minutes per hour} \\ &\times 8 \text{ hours per shift} \\ &\underline{\times 5 \text{ days per week}} \\ &= 1,458 \text{ pounds per week for one shift per day} \end{aligned}$$

or 2,916 pounds per week for two shifts per day

or 4,347 pounds per week for three shifts per day.

For around-the-clock, 7-days-per-week operation, the carryback would total 5,103 lbs or 2.55 tons.

While this total seems high, actual measurement of carryback on many belts can be in the neighborhood of 0.15 to 0.30 ounces per square yard of loaded belt surface. (This yard of loaded belt width assumes a 54-inch wide belt loaded to two-thirds of the belt width (36 inches).) This material can be released from the belt unless the belt is properly cleaned. Because there are so many variables in cleaning a belt, it is virtually impossible to reach 100% cleaning efficiency over an extended period of time.

Cleaners and Conveyors

Selecting a belt cleaner for a given application requires the assessment of a number of factors including belt speed, material transported and ambient temperatures, material size, and moisture content. It is important to design the cleaning system for the problems presented by “worst” material conditions, rather than “ideal” or even “normal” operating conditions. That way, the cleaning system will be better able to cope with changes in the material without becoming totally ineffective.

After a belt cleaner is installed, periodic inspection, adjustment, and maintenance will be required. The conveyor must be designed with required clearances to allow cleaners to be installed and serviced. The cleaners themselves should be designed for both durability and maintainability.

The foremost requirement of cleaner design is that it must minimize the risk of injury to the belt, splice, or cleaner itself. It would be counterproductive if a cleaner that is installed to enhance conveyor performance had the potential to damage the belt.

It is important that cleaners be installed out of the flow of the main material body and that the cleaned materials cannot adhere to the blades and structure. Cleaners should be applied at a point where the belt is supported by a pulley, pressure roller, or other device to prevent the belt from being forced up by the cleaning action (resulting in less efficient cleaning).

Typically, cleaners are made with blades that are less than the full width of the belt. (For example, a 72-inch belt may use a cleaner with a blade that is 66 inches wide.) This difference allows minor variations in belt tracking and does not affect cleaning performance, since typically the full width of the belt is not used to carry material.

Cleaning is improved by the use of vulcanized splices on the belt. Improperly installed mechanical fasteners can catch on cleaners and cause them to jump and vibrate or “chatter.” If mechanical splices are used, they should be recessed to the supplier’s recommendations. In addition, the belt cleaner mount should incorporate some relief mechanism that allows the blades to be pushed momentarily away from the belt and then return to cleaning position.

In addition to belt cleaners at the conveyor discharge, pulley cleaners and tail protection plows may also be required.

The Systems Approach to Belt Cleaning

Frequently, more than one “pass” at the belt must be made to effectively remove carryback material. Effective cleaning usually requires use of multiple cleaners. These systems can be composed of multiple cleaners located on the face of the pulley or a single cleaner on the pulley with one or more secondary cleaners positioned at a point after the belt leaves the discharge pulley. See Figure 11.4.

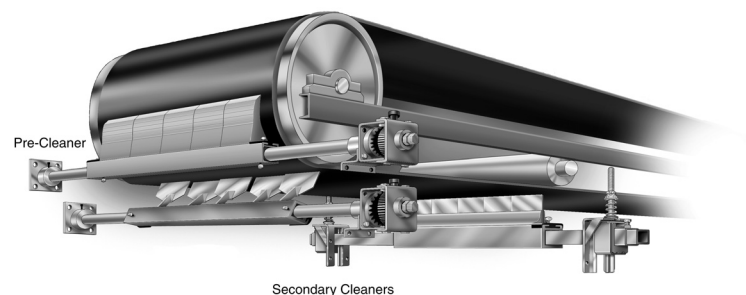


Figure 11.4 Multiple belt cleaner system.

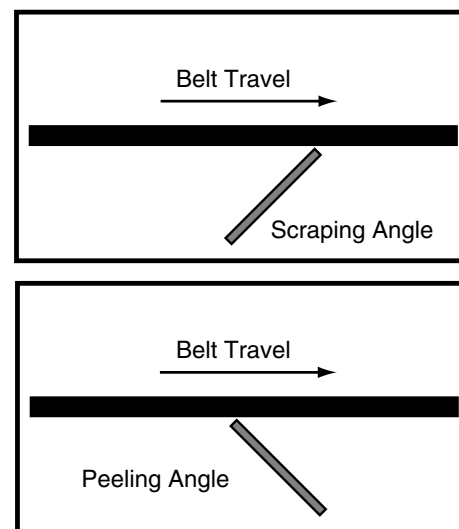
The first cleaner in either system will remove 90 percent or more of the carryback, allowing subsequent cleaner(s) to perform the final, precision, or mop-up removal of adhering fines without being overloaded with the mass of carryback. The cleaning efficiency of this system can be as high as 95 percent.

Pre-cleaners

Installed on the face of the head pulley just below the material trajectory, the primary cleaner acts as a “doctor” or “lopping” blade, shearing off most of the carryback. It leaves only a thin skim of fines on the belt to be removed by the subsequent cleaner(s). Because of its position on the head pulley, the material it removes is discharged into the main flow chute.

The position of the pre-cleaner blade in relation to the belt places it in a peeling position. See Figure 11.5.

Figure 11.5 Cleaning angles.



To avoid or minimize the risk of this dangerous (to belt, blade, and splice, position, pre-cleaners commonly use resilient elastomer blades (urethane or rubber) rather than metal and be only lightly tensioned against the belt. Lower blade-to-belt pressure means that the tensioning system will be able to relieve its blade — to bounce the blade away from the belt — when a mechanical splice moves past the cleaning edge, to reduce risk of damage. The low cleaning pressure also provides the benefits of improving blade life and reducing belt wear.

Secondary Cleaners

The closer to the conveyor’s discharge point the carryback is removed, the lower the potential for the release of fugitive material into the plant. Consequently, it is usually best to install the initial secondary cleaner so blades contact the belt as the belt leaves the head pulley. Additional cleaners should be positioned against other pulleys or hold-down rollers for optimum cleaning effect.

The blades themselves can be of a hard material — steel or tungsten carbide, for example — that resists heat build-up from its friction against the belt’s surface. Some operations prefer to avoid the application of a metal blade against the belt, so urethane or rubber is used but is less efficient.

The angle of the blade against the belt is an important consideration. A scraping angle is often used for secondary cleaners, as blades in a peeling position are quickly

honed to extreme sharpness. See Figure 11.5. These sharpened blades raise the risk an untrained or “in a hurry” adjustment will apply too much pressure to allow instant release from belt obstructions such as mechanical splices. The end result could be damage to the belt, splice, or the belt cleaner itself.

Construction Materials

Cleaning blades are available in a wide variety of materials; each conveying operation should specify blades that best suit its operating characteristics. Pre-cleaner blades are formed from urethane or rubber. Secondary blades are typically mild steel, stainless steel, heat-treated steel, tungsten carbide, or urethane. A number of urethane formulations are available, including high-performance versions that provide added resistance to high temperatures, industrial chemicals, or high-abrasion materials.

Strips of belting should not be used for belt cleaning because there may be embedded steel cables or fines of abrasive material caught in the belt carcass, both of which can cause excessive wear of the moving belt.

Maintaining Cleaning Pressure

Research by the U.S. Bureau of Mines indicates 11 to 14 psi blade pressure for secondary scraper-style cleaners is the optimum pressure for cleaning efficiency, blade life, and belt wear.

This research shows that applying the cleaner with too much pressure supplies no benefit and will increase blade and belt wear and power consumption. Undertensioning results in rapid blade wear and reduced cleaning efficiency. Maintaining blade-to-belt pressure in the optimum range is critical to efficient and effective cleaning performance.

Blade-to-belt pressure is maintained by some manner of tensioner. Tensioner devices range in sophistication from locking collars and concrete block counterweights to torsion storage devices and engineered air spring systems connected to the plant air supply. The choice depends on conveyor and cleaner specifications and plant preferences.

When adjustment or retensioning is required, the belt should be stopped and properly locked out. The cleaner and tensioner unit should be designed to facilitate any required service.

Belt Cleaners and Horsepower Requirements

Applying one or more cleaners against the belt increases the drag against the belt and so raises the conveyor’s power consumption. However, the consequences of not installing and maintaining effective cleaners prove a more serious drain of conveyor power, through the power consumption arising from idlers with material accumulations or seized bearings.

As reported by the Bureau of Mines, over-tensioning the cleaners is of doubtful cleaning value and will dramatically affect horsepower requirements.

Calculations for the impact of belt cleaners on conveyor power requirements are detailed in Chapter 6. These calculations should be reviewed whenever cleaning systems are changed or upgraded.

Belt Cleaners and Top Cover Wear

A research study has examined the issue of whether belt cleaners adversely affect the life of the belting. The study indicated that a belt cleaner can induce wear of the belt, but the rate of wear was approximately one-half of the wear seen when the belt runs through abrasive carryback without benefit of cleaners.

The Importance of Maintenance

Often, you get out of a belt cleaner what you are prepared to put into it. If you install a cleaner and never look at it again, its performance will deteriorate over time. If inspection, adjustment, and maintenance are provided on a timely basis, cleaning will continue to match original standards. Proper maintenance of belt cleaning systems will reduce wear on the belt and cleaner blades, prevent damage, and ensure efficient cleaning action.

Maintenance requirements should be reviewed when a cleaning system is specified. Some manufacturers are designing their equipment to allow easy maintenance. In addition, it is important that as the conveyor is designed and the cleaning system is installed to provide sufficient access and clearance to allow the required maintenance activities.

In some cases, it is best to let a plant's maintenance crew pick the belt cleaner, as they will then have made a commitment to it and will work harder to maintain its performance.

Specialized Cleaning Systems

In addition to conventional "bladed" belt cleaners, there are a number of alternative systems that may be selected to fit plant preferences, conveyor specifications, and cleaning requirements. These options include:

Rotary Brush Cleaners

These could be free-wheeling (driven by the motion of the belt) or powered by an electric motor. See Figure 11.6. They can be effective on dry materials; however, they sometimes encounter problems when material builds up in the bristles of the brush. Efficiency is in the range of 90 percent.

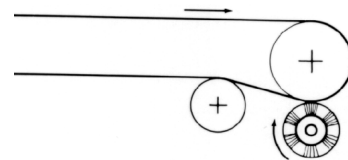


Figure 11.6 Rotary brush cleaner.

Pneumatic (Air Knife) Cleaners

These direct a stream of air over the face of the pulley to shear off carryback. See Figure 11.7. This system can be effective in removing dry or extremely wet materials; their disadvantages include the continuing expense of providing the air stream, and the creation of additional airborne material that can settle someplace else in the plant or build up in the chute. Efficiency is in the 75 percent range.

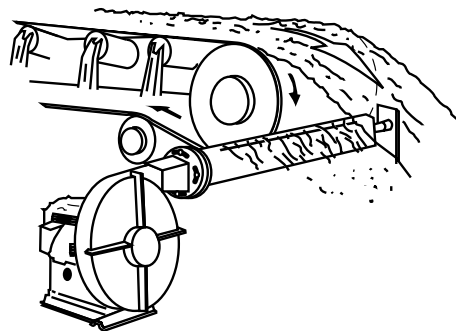


Figure 11.7 Pneumatic belt cleaner.

Belt Washing Stations

Two styles of cleaning system use water. See Figure 11.8. The first directs a low-pressure (2 to 5 psi) mist of water at the belt to assist in the scraping process. The second washing system use a higher pressure (60 to 100 psi) spray of water to remove the carryback from the belt. After the nozzles wet the belt, one or more cleaner assemblies with urethane or metal blades removes the material and water from the belt. Either style of “wash box” can provide effective cleaning, but poses the problem of disposing of the slurry of removed material and water. In some cases, the water can be filtered and reused; in other cases, both water and material can be returned to the conveyed material load. The low-pressure system uses less water, and consequently reduces the water/sludge handling problem. Both systems must be carefully engineered to avoid problems in freezing weather. Cleaning efficiency is in the range of 98 percent.

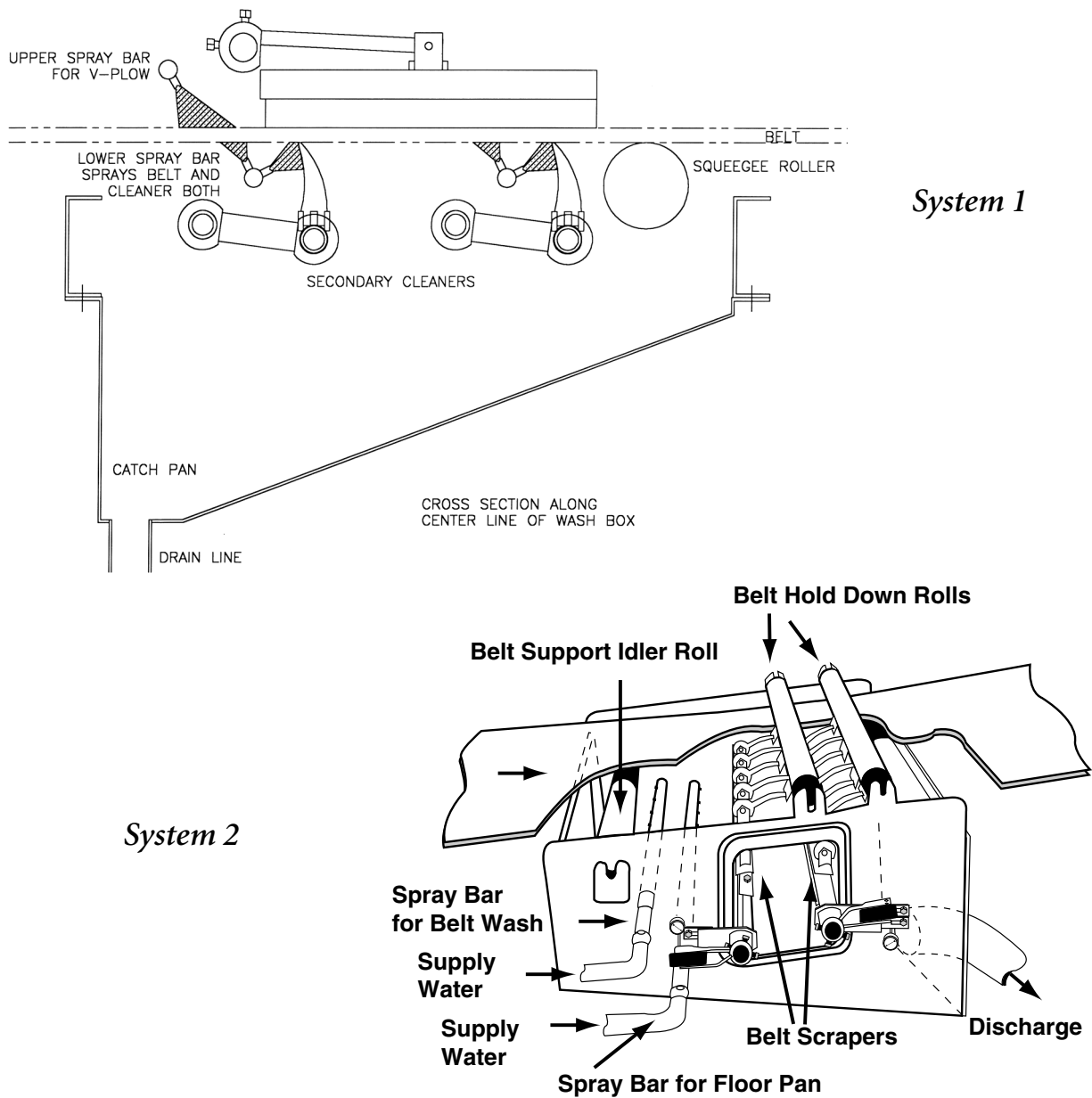


Figure 11.8 Belt washing stations.

Cleaners for Special Applications

Manufacturers now offer a variety of cleaners to fit applications with particular requirements. Some of these specialty systems include food-grade cleaners, cleaners utilizing electrically heated blades to remove extremely sticky material, high-temperature cleaners for continuous duty in conditions up to 250° F, and cleaners for belts with ribs, cleats, or chevrons.

Handling the Removed Material

The fact that carryback clings to the belt past the discharge point indicates that it has different characteristics than the rest of the load. The particles tend to be finer, with a higher moisture content. Even after its removal from the belt, this material presents serious capture, handling, and disposal problems.

As noted above, it is usually best to locate the belt cleaner(s) on the conveyor return as far forward — as close to the discharge point — as possible, without being in the trajectory of the discharged material. This gives the belt most, if not all, of its cleaning before fugitive material can be released into the plant. In addition, cleaning on the head pulley minimizes the amount of handling required for the material cleaned off the belt.

Ideally, the conveyor's discharge chute is sufficiently large enough so material cleaned from the belt can fall through the same chute, where it is collected and reunited with the main stream of material. If the main chute is not large enough, an extra fines chute or dribble chute should be added under the secondary cleaners to prevent the cleaner's encapsulation in these sticky, cementitious materials. Care must be taken to prevent this smaller chute from bridging over or clogging with the sticky fines. Measures to prevent dribble clogs include the use of low-friction liners or a small vibrator to keep the material in motion.

Special scavenger or fines conveyors are sometimes used to return the belt scrapings to the main material stream. These devices — often vibratory conveyors or drag-scraper feeders — are particularly common in installations where multiple secondary cleaners are installed at some distance from the discharge pulley. See Figure 11.9.

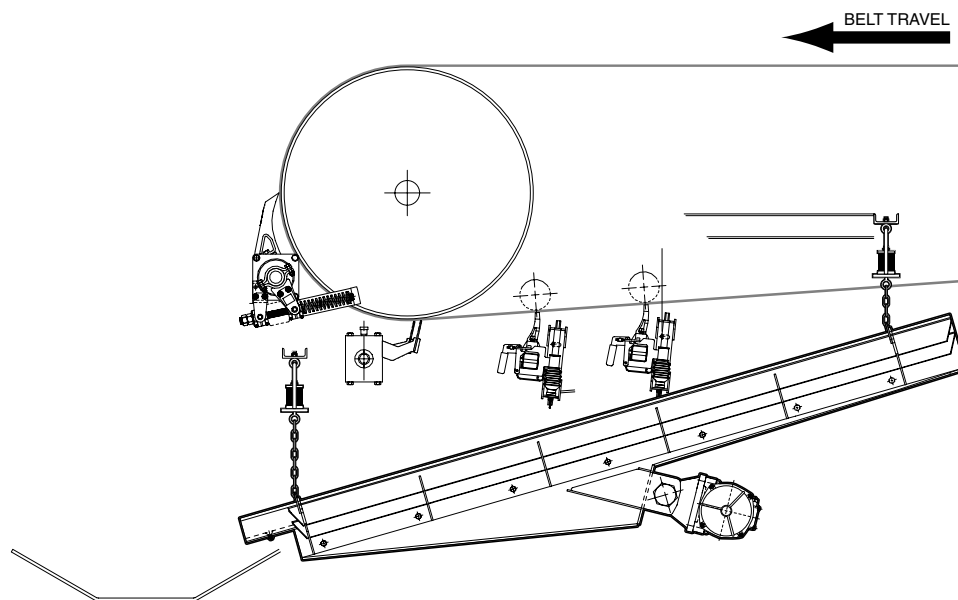


Figure 11.9 Vibrating scavenger conveyor to return fines removed by secondary cleaners to main material flow.

Belt Turnovers

To eliminate the problems caused by a dirty belt in contact with return idlers, the belt can be twisted 180 degrees after it passes the discharge point. This brings the clean surface of the belt into contact with the return idler rolls. The method of accomplishing this is illustrated in Figure 11.10. The belt must be turned back again 180 degrees before it enters the tail section to bring the carrying side of the belt up at the loading point.

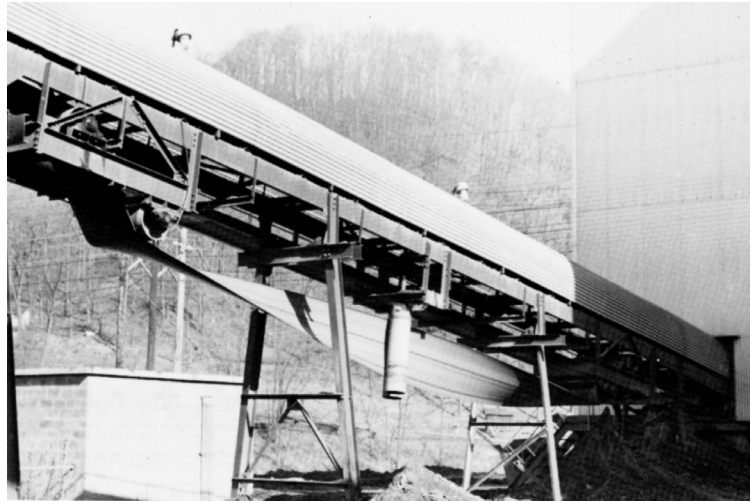


Figure 11.10 Belt turnover scheme.

As this twisting of the belt induces abnormal tensions in the carcass of the belt, a CEMA member should be consulted for the proper location of the snub pulleys that turn the belt. (These are not shown in Figure 11.10.) The distance required to accomplish the 180° turnover of the belt is approximately 12 times the belt width at each end of the belt.

Belt turnovers do not eliminate the need for a belt cleaning system. Turning the belt over will often dislodge residual material from the belt. To minimize the chance of carryback remaining on the belt to be released when the belt is twisted, a conventional belt cleaning system (consisting of a pre-cleaner at minimum) should be installed at the conveyor's discharge.

Because fugitive materials is released by the act of twisting the belt, the turnover system may result in getting the “clean” side of the belt dirty. In addition, this scheme can increase abrasive wear on what is usually the belt's thinner undersurface, perhaps leading to the purchase of a new belt or more rugged and more expensive grade of belting.

Turnovers are generally installed on long belts after the conveyor's take-up system or snub-pulley system, and so they generally leave plenty of opportunity for carryback to fall from the belt into the conveyor system before the turn-over system inverts the belt.

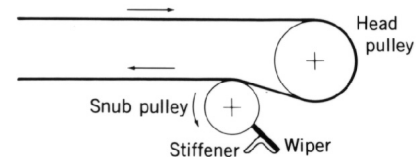
Pulley Wipers

Materials that will stick to the belt can often transfer and adhere to any snub or bend pulleys that contact the dirty side of the belt. Therefore, pulley-cleaning devices may be as necessary as belt cleaners. Lagging snub and bend pulleys reduces buildup and wear on expensive pulleys.

To clean adhering material from a pulley, a scraper is typically mounted on the ascending side of the pulley, about 45 degrees below the horizontal center line. This is

shown in Figure 11.11. This position allows material scraped from the pulley to fall free. Care must be taken to provide a dribble chute or an accessible area where this cleaned material can accumulate for later removal.

Figure 11.11 Snub pulley wiper.



Most commercially available belt cleaners can be applied as pulley cleaners; consult a CEMA member company for information.

Return-Run Belt Cleaning

Even though a belt conveyor is carefully designed, spillage from the carrying side of the belt may occur at the loading point and elsewhere along the belt. This fugitive material can land on the return run of the belt, eventually becoming trapped between the belt and tail pulley. This leads to damage to, or misalignment of the belt. If such spillage occurs, it might land on the return run of the belt and become trapped between the belt and tail pulley, causing possible damage to, or misalignment of, the belt.

Two devices used to prevent this difficulty are deck plates and tail protection plows.

Deck Plates

Deck plates or decking consists of flat, bent, or curved metal sheets installed between the belt's carrying and return runs. These plates act as a shield to deflect any fugitive material. This construction, common to many well-designed belt conveyors, protects the return run of the belt, both from any spilled material and from the weather.

The three styles of decking shown in Figure 11.12 are used more or less according to the designer's preference and the nature of the supporting structure. Both the round and inclined styles allow material that has landed on the decking to slide off to the outside of the conveyor.

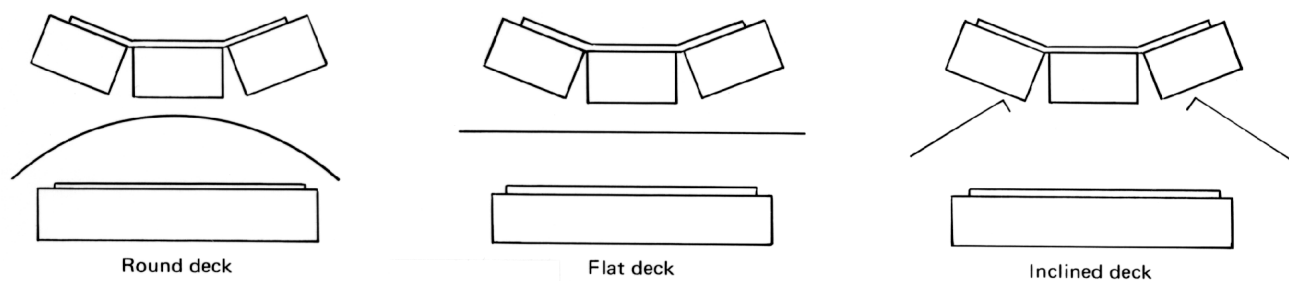


Figure 11.12 Three typical styles of belt conveyor decking.

Sometimes decking is applied only at the loading point and forward of it for about the next 50 feet down the belt line. If this design is used, then a tail protection plow, as described below, should also be provided. If decking is used, provisions for periodic cleaning should be considered.

Tail Protection Plows

Installed on the upper side of the return belt just before it enters the tail pulley, a tail protection plow (or return belt scraper) removes any material spilled onto the return run. It thus prevents this spilled material from becoming trapped between the belt and the tail pulley.

Depending on conveyor configuration and load, it may be desirable to install return belt scrapers to protect additional pulleys.

This scraper is typically a diagonal plow or “V”-plow using a rubber or urethane blade to push any fugitive material off the belt. In most cases, the plow rides on the belt, tensioned against the belt either by a tensioning device or simply by its own weight, free to float up and down with any fluctuations in belt travel. Both styles of tail protection plow should be located between the tail pulley and the first return idler. See Figure 11.13. The plow should be secured with a safety cable to prevent it being carried into the pulley should it become dismounted.

On belt conveyors that travel only in one direction, the plow can be a “V”-plow. The point of the “V” is aimed away from the head pulley, so any loose material carried on the belt’s inside surface is deflected off the conveyor by the wings of the plow.

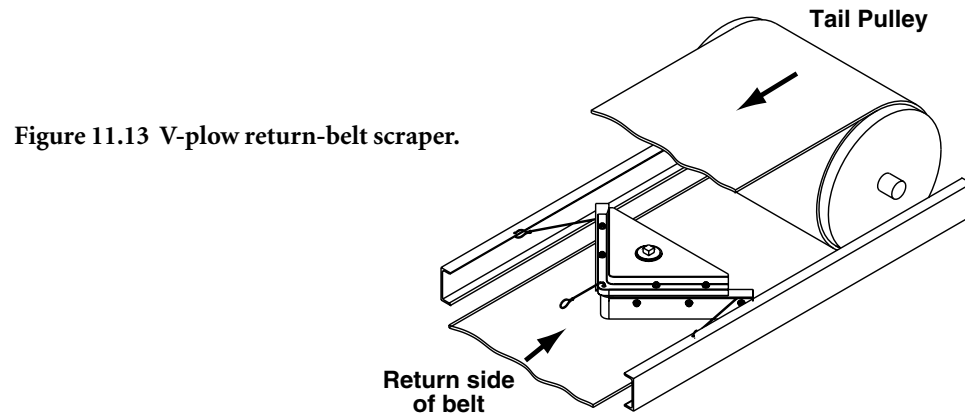


Figure 11.13 V-plow return-belt scraper.

If the conveyor has two directions of movement — through either reversing operation or accidental rollback — the return belt scraper should be a diagonal plow that provides protection regardless of the direction of belt movement. Diagonal plows are installed across the belt at an angle of 45 degrees to the direction of travel. See Figure 11.14.

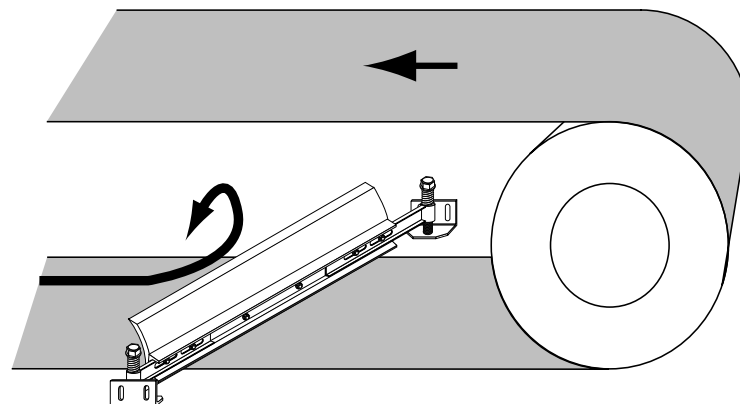


Figure 11.14 Diagonal plow return belt scraper.

A diagonal plow can be installed on a one-direction conveyor if there is a reason to collect or discharge the material on a specific side of the belt. In this case, the diagonal plow should be installed so its discharge end — the side of the belt where the cleaned material should be collected — is closest to the pulley.

If the conveyor is truly two-directional — so either pulley can serve as a discharge pulley — diagonal plows should be installed at each end.

Belt Conveyor Accessory Equipment

After selecting and designing the major components of a belt conveyor system, it is still necessary to consider a number of secondary items which are discussed here as belt conveyor accessory equipment. This includes wing-type tail pulleys, tramp iron detectors, in-process weighing and sampling devices, and weather and spillage protection devices such as housings, wind breaks, wind hoops, and decking.

Weather Protection

The need to protect belt conveyors from the weather varies with the climate, the material handled, and the type of operation. The material being handled may be affected by rain, or, if the material contains moisture, by freezing temperatures. Rain on the pulley side of the belt or on the drive pulleys may cause slippage between the belt and the pulley. Ice and snow on these same surfaces can completely stop a belt conveyor. Also, the life of rubber covers may be shortened by the effect of intense sunlight.

Strong winds can lift conveyor belts off their idlers and cause severe belt training problems. On conveyors with narrow belts, and on those handling light materials, it is possible for the wind to lift the belt, spill the load, and cause serious damage to the belt. This is particularly true of tripper belts, where the belt leaves the troughing idlers and rises on a rather long incline to the tripper head pulley. Wind may also create a nuisance by blowing fine materials from the conveyor belt.

When the belt conveyor is not contained within an enclosed gallery (which obviously offers the most complete protection for the conveyor parts and maintenance personnel), one or more of the following belt protections can be used.

Half-Covers

Half-covers usually are semicircular in shape and are made of flat or corrugated sheets, commonly either of galvanized iron or aluminum. They are fastened to the stringers and are situated over the belt, as shown in Figure 11.15. The arrangement and fastening of the covers should permit convenient access to permit servicing the belt and idlers. The belt conveyor covers frequently are hinged at one side.

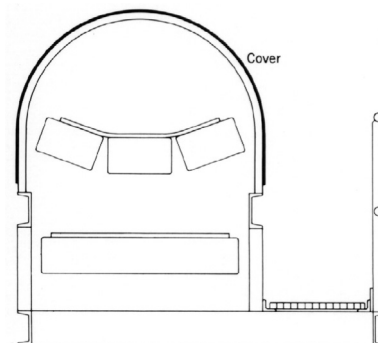


Figure 11.15 Half covers over conveyor belt.

Three-quarter covers

As the name implies, these covers are less than a complete semicircle. On the walkway or access side of the belt conveyor, to facilitate inspection, the lower edge of the cover may terminate slightly above the high edge of the troughed belt. Figure 11.16 shows three-quarter covers over a conveyor belt.

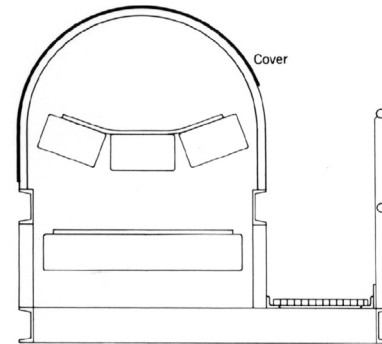


Figure 11.16 Three-quarter covers over conveyor belt.

Wind breaks

Where protection is needed from wind alone, it can be provided by installing a suitably reinforced metal sheet on the windward side of the conveyor stringers. This sheet should extend above and below the stringers to protect both carrying and return runs of the belt. See Figure 11.17.

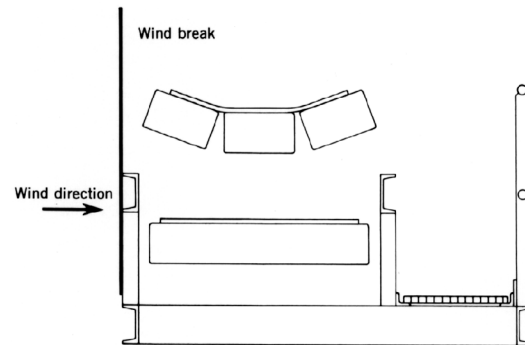


Figure 11.17 Wind break on belt conveyor.

Wind hoops

In some instances, it is necessary only to prevent the empty or lightly loaded belt from being blown off the idlers. This can be accomplished by providing suitable hoops, spaced at regular intervals along the belt conveyor, as shown in Figure 11.18.

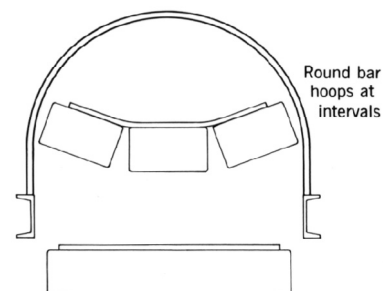


Figure 11.18 Example of wind hoops on a belt conveyor.

In some operations, the conveyor may be shut down for long intervals of time. If the belt is not otherwise protected it can be lashed to the stringers at frequent intervals by light rope ties.

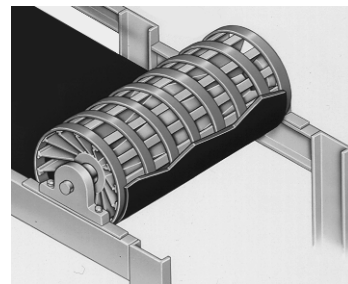
Spillage Protection: Wing-Type Pulleys

A wing-type pulley is a cast or fabricated pulley with a discontinuous surface (see Figures 8.7 and 8.8). Between the cross bars are inclined valley-shaped recesses that prevent fine or granular material from being caught between the tail pulley and the return belt. The material gathers in the valley-shaped recesses and falls out of the open ends as the pulley revolves.

If a conveyor handling such fine or granular material is likely to spill some of it on the return belt, the wing pulley is an effective device for removing spillage without belt damage. Wing-type pulleys also have been used as vertical gravity takeup pulleys, with the same benefits.

Care must be used when applying a wing-type tail pulley, because the intermittent, on-again, off-again contact pattern between the “wings” and the belt can impart an up-and-down fluctuation to the belt line. This vibration makes it difficult to effectively seal the loading zone, resulting in additional material spillage. Wrapping a band of steel in a spiral around the wing pulley allows the pulley to provide the self-cleaning benefit without creating oscillation in the belt line. See Figure 11.19.

Figure 11.19 Spiral-wrapped wing-type pulley.



Tramp Iron Detectors

In Chapter 8, the use of magnetic pulleys for removing tramp iron is discussed. There are other methods of removing or detecting tramp iron. One is to suspend a stationary magnet over the belt, alone or in combination with a magnetic pulley.

Another method is to arrange a short conveyor above and at right angles to the belt conveyor. Here, a stationary magnet, located between the runs of the cross belt, extends beyond one side of the conveyor below. Material that is picked off the belt conveyor is held against the underside of the cross belt by the magnet. As soon as the belt is removed from the magnetic field, the material drops into a container. This device, although employing a stationary magnet, is self-cleaning.

A third approach, sometimes preferred, is to pass the belt through a magnetic field of adjustable intensity. The device is electrically arranged so that a disturbance of this magnetic field, by passage of a preselected size of iron particle, will shut down the conveyor belt.

Conveyor Belt Scales

For in-plant process functions, the continuous weighing of a wide variety of bulk materials on a moving conveyor belt is accomplished through the use of belt scales. Three types of scales are manufactured specifically for this purpose: the all-mechanical system, the electronic scale, and the nuclear scale.

The first two types operate by weighing the actual material as it passes over a weigh bridge, measuring the belt speed, and integrating these measurements into a rate output. Figure 11.20 illustrates a typical electronic load cell type of belt scale.

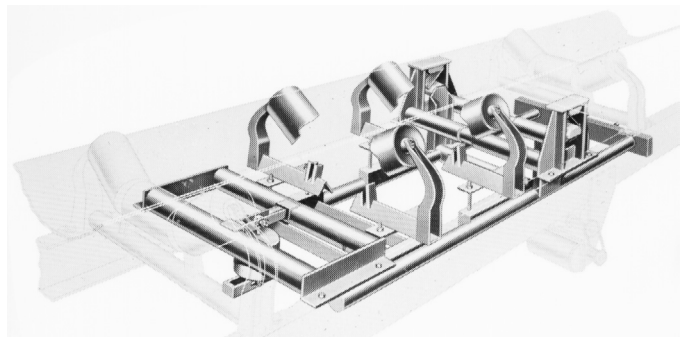


Figure 11.20 Typical electronic load cell type of belt scale.

The nuclear scale measures the relative density of the material on the moving belt and compares this with the known density of the material. A conversion of density to weight is computed and integrated with belt speed to achieve a rate output.

The various manufacturers of belt scales have designed their equipment for compatibility with standard belt conveyor design practices, so that scales are easily adaptable to both existing and newly designed conveyor systems. Scale accuracy is generally considered to be within $\frac{1}{2}$ of 1 percent if properly installed in accordance with the manufacturer's recommendations.

Sampling Devices

Sampling is a process of obtaining a small portion of a material which is representative of the whole. Samples are taken both to determine acceptable quality and to control operations and inventory. The most accurate method of sampling is to stop the loaded belt, insert a template to fit the curve of the belt, and shovel or sweep off a precalculated quantity of material. This must be done a specific number of times, depending upon the size of the lot being sampled. This large sample is then reduced and sent to the laboratory for analysis.

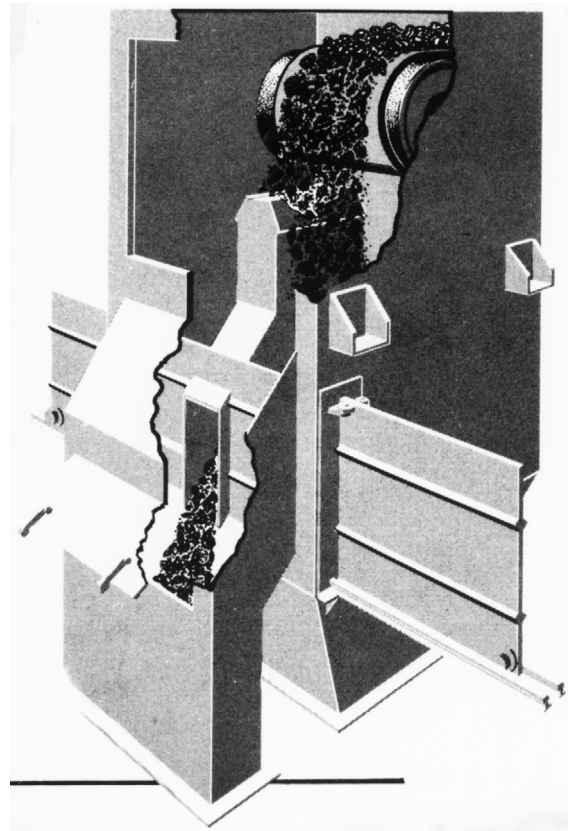


Figure 11.21 Typical sampling arrangement.

However, for most operations, this method would prove impractical if performed continually. An acceptable alternative is to cut the stream of material completely at a right angle to the flow of the stream at a suitable discharge point in the conveying system. Figure 11.21 illustrates a typical sampling arrangement. A cutter device traverses the stream at a right angle removing a “typical” cross section at the conveyor load and comes to park completely out of the stream. The cutter is driven across the stream by means of a hydraulic cylinder. Electric motor and pneumatic drives are also sometimes used.

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Conveyor Loading and Discharge

Loading the belt

Direction of loading

Transverse belt displacement

Loading an inclined belt conveyor

Impact at loading point

Loading chutes and skirtboards

Loading chutes

Skirtboards

Feeders

Methods of discharging from the belt

Discharge over-the-end pulleys

Discharge chutes

Lowering chutes

Trippers

Plows

Discharge trajectories

Calculating and plotting normal material trajectories

Horizontal, inclined, and declined conveyor belt trajectories

Plotting the trajectory

Examples of trajectories

The successful operation of a belt conveyor requires: first, that the conveyor belt be loaded properly; second, that the material carried by the belt be discharged properly. These two requirements are very important and must be given most careful consideration if the belt conveyor is to function as intended.

Loading the Belt

While the loading of material onto a belt conveyor involves many considerations, of prime importance is the placing of the material centrally on the belt in such a manner that the material velocity in the direction of belt travel is, as nearly as possible, equal to the velocity of the belt itself.

If the material isn't delivered onto the belt at the belt speed, there will be a turbulence in the mass of material at the loading point. A build-up in volume may form at this point. This material turbulence is a function of the velocity difference.

Provided that the material is delivered centrally onto the belt, the ideal loading is attained when the forward velocity in the direction of the belt is exactly the same as the velocity of the belt. When this condition exists, there is minimum wear on the belt cover, minimum power is required to operate the belt, the material takes the proper load shape quietly without spillage, and minimum degradation or dusting of the material is assured. While this ideal condition is difficult to reach in actual practice, it is well worth the effort to attain it as closely as possible.

It is important that the belt be stable for loading. The placement of material on the belt should not be done while the belt is in transition from flat to troughed. If the proximity of the tail pulley to the loading zone prevents the completion of troughing, it is possible to partially trough the belt (for example, to 20 degrees) before material loading, and then complete the troughing (to 35 degrees or 45 degrees) outside the loading zone.

Other factors of loading, listed below, also must be considered.

Direction of Loading

There are only two possible directions — in the direction of belt travel, or transverse to the direction of belt travel.

Loading in the Direction of Belt Travel

This type of loading is best, as it presents the simplest and easiest design problem. The flow of material down a loading chute onto the moving belt surface can be arranged so that the forward velocity of the material is nearly the same as the belt velocity.

The material flow can be directed centrally onto the belt so that the shape of the load is symmetrical. The skirtboards at the loading point can be of minimum length. Proper loading of the belt is not appreciably affected by fluctuations in the feed rate. The loss of height at the transfer from one conveyor to another is at a minimum with this direction of loading.

Unfortunately, loading of belt conveyors in the direction of belt travel is not one of the usual layout configurations.

A short, speed-up belt conveyor can be used to advantage between the loading chute and the loading point of a long, high-speed belt conveyor, as shown in Figure 12.1. Use of the short, high-speed conveyor avoids the wear that otherwise would occur to the cover of the long and costly high-speed belt because of the acceleration of the loaded material.

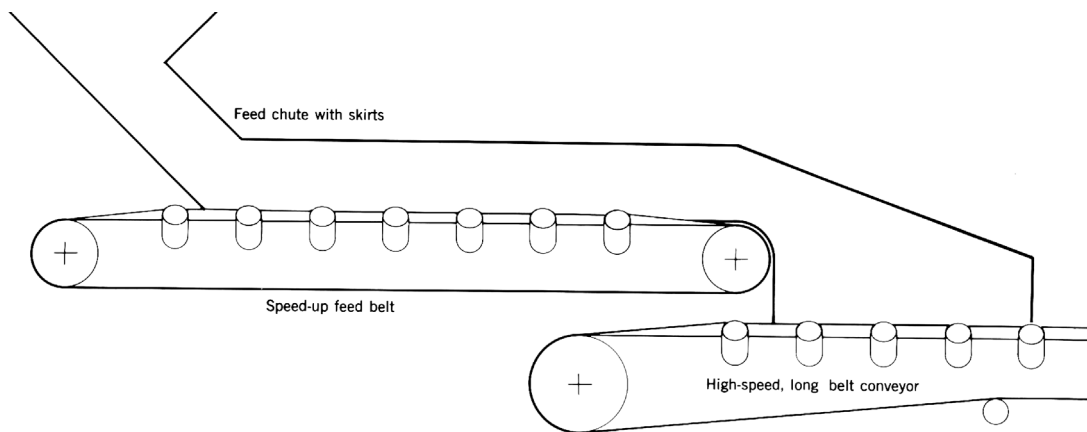


Figure 12.1 Speed-up belt conveyor.

The length of this speed-up conveyor should be sufficient to bring the speed of the material very close to the speed of the long belt conveyor. The cover of the speed-up belt should be thick enough to accept the wear caused by the acceleration of the material leaving the loading chute.

If the impact of the material leaving the loading chute is high, the speed-up conveyor should be flat. It should run on flat, impact-absorbing idlers or an impact-absorbing slider bed. Continuous skirtboards with an effective edge sealing system should be provided. This flat belt can be made of the required plies and thickness of cover to meet the impact, without consideration of belt-troughing capabilities.

If the impact condition is not severe, a lighter-bodied belt can be used, with a cover thick enough to withstand the wear. This lighter belt can be carried on troughing idlers, as illustrated in Figure 12.1.

Flat or troughed, speed-up conveyor belts must be considered expendable and therefore must be periodically replaced.

Loading Transverse to the Direction of Belt Travel

This type of loading is found very frequently in belt conveyor layouts. The horizontal angle that a belt conveyor makes with the succeeding belt may be less or more than 90°.

Any angularity at the transfer presents the problem of turning the flow of material, so that finally, as the material passes to the succeeding belt conveyor, its velocity in the direction of the succeeding belt travel is as close to the speed of this belt as possible. As the angularity of transfer increases from a small angle to 90 degrees or more, the design of the loading chute becomes a more difficult problem, from the standpoint of both desirable material velocity and central loading of the receiving belt.

With angular transfers up to 90 degrees, the height required at the transfer point increases. Also, it is increasingly difficult to load the belt centrally at all rates of material flow. With large angularity, in spite of careful loading chute design, field adjustments sometimes are required to center variable loads on the receiving belt. Skirtboards at the loading point may have to be higher and longer, in order to retain the material on the belt without spillage until the material is accelerated to the belt

speed and assumes the proper load shape. There is more wear on the belt cover, and there is likely to be increased wear on the loading chute itself.

With careful consideration of loading chute design, a compromise of all the factors can be effected which will nearly balance the disadvantages of this direction of loading.

Horizontal angularities of transfer greater than 90 degrees should be avoided. While it is possible to effect a transfer of material from one belt conveyor to another where the angularity exceeds 90 degrees, the design of the loading chute becomes very difficult. Considerable height must be lost at the transfer point. And, it seldom is possible to obtain an acceptable material velocity in the direction of belt travel. The hazards of belt cover wear and loading chute wear are thus accentuated.

Transverse Belt Displacement

If the material being loaded on a belt conveyor is moving in a direction other than directly in the line of belt travel, the belt is likely to be displaced transversely on its supporting idlers.

Likewise, if the material is not uniformly spread on the belt and piles up against one of the skirtboards, the belt can be displaced transversely on the idlers. Such belt displacement causes belt-training difficulties and may result in spillage of the material over the edge of the belt beyond the skirtboards.

Loading an Inclined Belt Conveyor

If the belt conveyor is inclined (slanting upward) at the loading point, it is difficult to attain the minimum difference between the velocity of the material and the velocity of the belt. The greater the incline and belt speed, the greater the difficulty. Thus, the material loaded on an inclined portion of a belt conveyor is subject to much turbulence before it can be accelerated to the speed of the belt. Consequently, the loading skirts may have to be higher and longer to retain the material and avoid spillage. The wear on the belt cover is great if the material is at all abrasive or has sharp-edged particles or lumps.

Conveyors declined (slanting downward) also present loading problems. Careful consideration of the material and belt velocities is essential to satisfactory loading.

Where possible, it is desirable to make the loading portion of the belt conveyor horizontal. This horizontal loading portion is then connected to the succeeding inclined or declined portion by means of a suitable vertical curve.

Impact at Loading Point

Because in a vertical plane the direction of flow of the material being loaded on a belt conveyor is almost never exactly in the direction of belt travel, some impact is encountered as the material engages the surface of the belt. Severe, uncushioned impact can damage the belt cover and weaken the belt carcass.

Very fine materials, even if heavy, do not cause much impact. But the belt can be deflected between the idlers unless the belt is supported by closely spaced idlers or belt support cradles. Such belt deflection allows material to pass under the skirtboards with resulting spillage over the belt edges, at or beyond the loading point.

Lumpy materials, especially if the lumps are heavy, can cause appreciable impact on the belt. If the lumps are sharp, the belt cover may be nicked or cut. Heavy lumps also tend to crush the belt carcass, thus weakening it.

To reduce this deflection and damage, impact-absorbing idlers or impact cradles should be used. They should be located under the loading point of the belt so the

major impact force strikes the belt between the supporting idlers rather than over any one of them, or in the middle of the cradle, rather than at either end. See Figure 12.2.

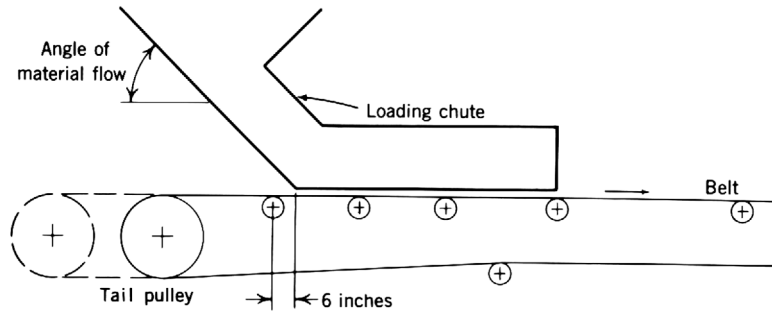


Figure 12.2 Typical loading chute.

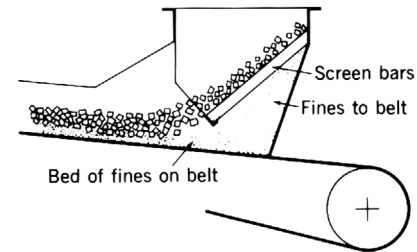


Figure 12.3 Grizzly or screened loading chute (after Hetzel.)

Where the material consists of a mixture of fines and lumps, the loading chute can be arranged, so it first deposits the fines on the belt, and then deposits the larger pieces. This allows the layer of fines to cushion the impact of the lumps. This is accomplished by incorporating a grizzly bar or screen (Figure 12.3), or a notched or wedge shaped loading opening in the chute. (Figure 12.4).

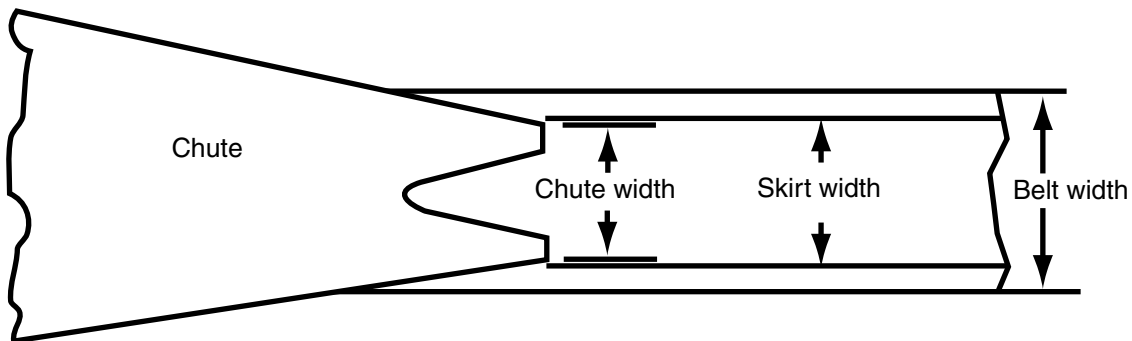


Figure 12.4 Notched or wedge shaped loading opening.

Impact problems, especially with lumpy materials, are accentuated when the loading portion of the belt conveyor is inclined. It may be advantageous for the belt to be horizontal through the loading zone, and then be inclined after it is loaded and the material has stabilized.

Loading Chutes and Skirtboards

Loading chutes and skirtboards can seriously damage conveyor belts if not properly designed and installed. Therefore, their design should be given careful consideration. Chutes and skirtboards should be fastened securely, to position them accurately with respect to the belt. Care must be taken to ensure the belt does not rise up off the idlers at startup or when running empty. The skirtboard should be of proper length and sufficient clearance above the belt to combine safety for the belt with proper control of material and spillage prevention.

Skirtboards should extend to a point where the material has stabilized at belt speed. Greater distances are required to stabilize the material when loading on an incline. Therefore, longer skirtboards are required for the loading portions of the inclined belt conveyors.

Loading Chutes

The design of loading chutes is worthy of careful study. Where the loading of the belt is to be accomplished in the direction of the belt travel, loading chutes can be quite simple. Where the loading of the belt must be accomplished in an angular direction, the design becomes more complex. The design of successful transverse loading chutes and skirtboards for high-speed belt conveyors requires much care and ingenuity.

Obviously, the loading chute must be inclined in order to give the material flow a desirable forward velocity. If the material is fine and contains some moisture, the chute must be made steep enough so that the material will slide rapidly. However, if the material is lumpy, the steepness of the chute is limited to that angle at which the material will slide satisfactorily, but not bounce and tumble. High lump velocities may be controlled by the use of baffle bars or chains hung in the path of the lumps.

Multiple-angle chutes, curved chutes, and sometimes covered chutes can be employed to impart a uniform sliding action to the material. If sufficient velocity cannot be imparted to the material in the proper direction it may be necessary to reduce the speed of the receiving belt conveyor. This is done to obtain the minimum difference between the forward velocity of the material flow and the belt velocity. However, such a compromise may result in a wider, more costly belt.

Loading chutes can be made of metal or other materials. Metal chutes are the most common. For abrasive materials, the chute can be lined with abrasion-resisting, removable plates or other material, such as ceramic liners. For corrosive materials, corrosion-resistant metal coatings, rubbers, synthetics, or fused-on glass linings can be used.

Width of Loading Chutes

The width of a loading chute should be no greater than two-thirds the width of the receiving belt. On the other hand, the inside width of the loading chute should be at least two and a half to three times the largest dimension of uniformly sized lumps, when they represent a considerable percentage of the material flow. Where lumps and fines are mixed, the inside width of the chute may be as small as two times maximum lump size.

These proportions are essential to the proper loading of the belt and to the prevention of interlocking and jamming of lumps in the chute. Thus, the width of the loading chute could, in some cases, determine the width of the belt on the receiving conveyor.

Loading Chutes and Skirtboards

Grizzly or Screened Loading Chutes

Where a mixture of lumps and fines is to be handled, heavy loading impact on the belt should be minimized in one of two ways: (1) by arranging the chute to place a layer of two fines on the receiving belt ahead of the loading of the lumps, or (2) by using curved, perforated, or notched chute bottoms, or grizzly chutes. The latter sometimes are called screened chutes. Figure 12.3 shows a grizzly or screened chute for loading a slightly inclined belt conveyor. These measures will probably not eliminate the need for a shock-absorbing belt support system, such as an impact bed or impact idlers.

When loading a declined belt conveyor, it is best to load the fines ahead of the lumps for another reason. If the fines make a layer on the belt in which the lumps subsequently may be imbedded, the lumps will be prevented from bouncing and cascading down the declined, moving conveyor belt.

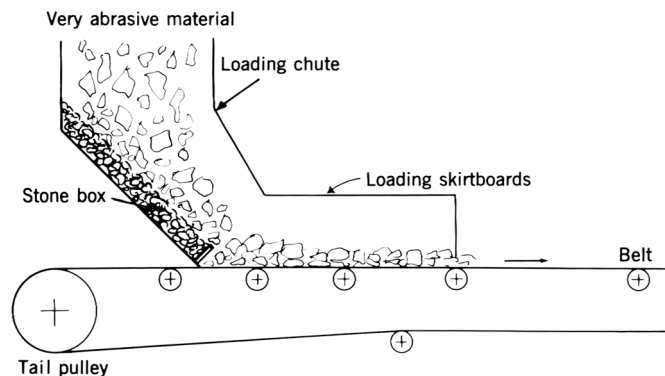
Details of a Loading Chute

The back or bottom plates of chutes used for loading should be fitted rather closely to the belt and should be provided with adjustable rubber edging to prevent leakage of fines. Such rubber edging or skirtboard seal also prevents lumps from getting under or behind the back plate and jamming between the back plate and belt.

Stone Box Chutes

If the material to be handled is severely abrasive and the speed of the receiving belt is slow, it is possible to arrange the chute bottom so it forms a box in which some of the material is retained. The loading material will then flow over this retained material, wearing the material rather than the chute walls. This arrangement commonly is called a “stone box” or “rock box” and is most often seen in gravel, rock, and ore handling. Figure 12.5 shows a typical stone box loading chute.

Figure 12.5 Stone box loading chute.



Back of the Loading Chute

The back or bottom plates of chutes used for loading should be fitted rather closely to the belt and should be provided with an adjustable sealing strip to prevent material rolling out the back of the loading zone. This sealing strip also prevents lumps from getting under or behind the back plate and pinched between the back plate and belt. A tail box installed on the back of the chute allows the sealing strip to

be inside the box, allowing any carryback or residual material sticking to the belt to pass through while keeping the loading material on the belt. See Figure 12.6.

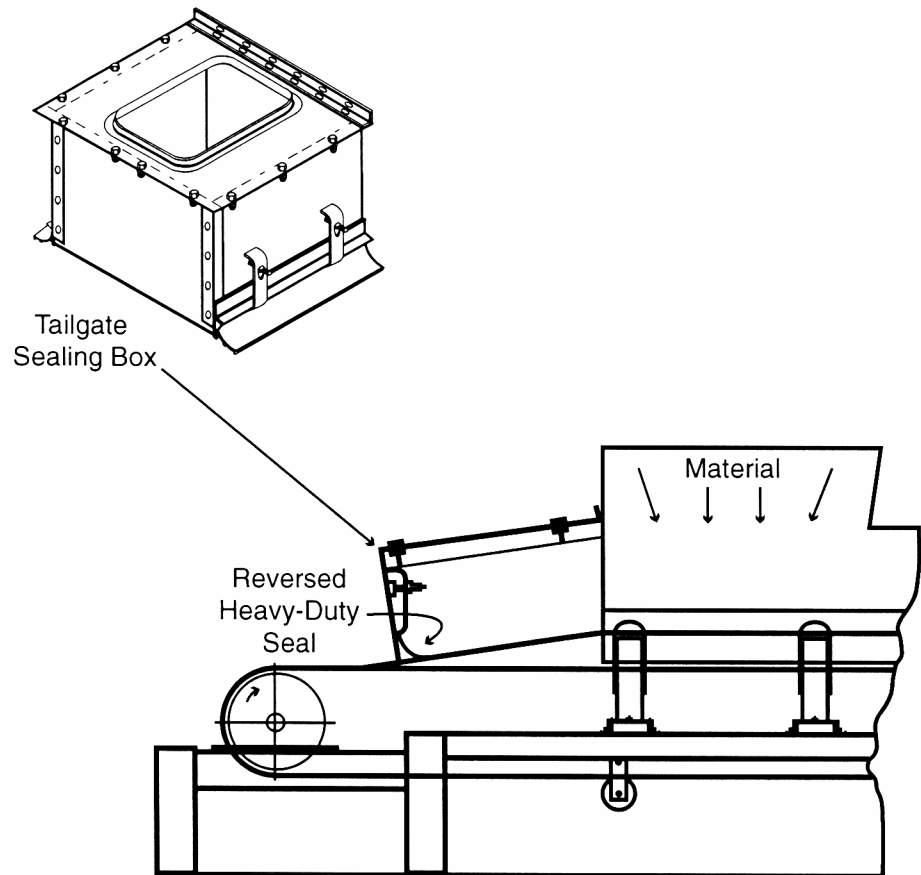


Figure 12.6 Tailbox.

Chute Liners

The use of a wearable liner inside the chute helps to centralize the loading material and preserves the flexible elastomer sealing strips from the forces of the material as it loads. Available in a deflector or straight pattern (Figure 12.7), these liners isolate the sealing strips from the weight of the material load, enabling the strips to effectively keep fines on the belt.

Deflector designs have been commonly used as they offer a centralizing effect on the falling material. However, this design can reduce chute capacity, and in load zones with impact, it can catch bouncing materials inside the deflector angle. The more common profile is now straight liners.

Materials for this wear liner should be matched to the characteristics of the material and load passing through the chute. This liner is sacrificial material, designed for replacement as it is consumed by abrasive wear.

Skirtboards

To retain the material on the belt after it leaves the loading chute and until it reaches belt speed, skirtboards are necessary. These skirtboards usually are an extension of the sides of the loading chute and extend roughly parallel to one another for some distance along the conveyor belt. The skirtboards normally are made of metal,

although wood sometimes is used. The lower edges of the skirtboards are positioned some distance above the belt. The gap between the skirtboard bottom edge and the belt surface is sealed by a more or less flexible elastomer sealing strip, attached or clamped to the exterior of the skirtboard. See Figure 12.8. Figures 12.9 and 12.10 show typical skirtboards arrangements for flat and troughed belts.

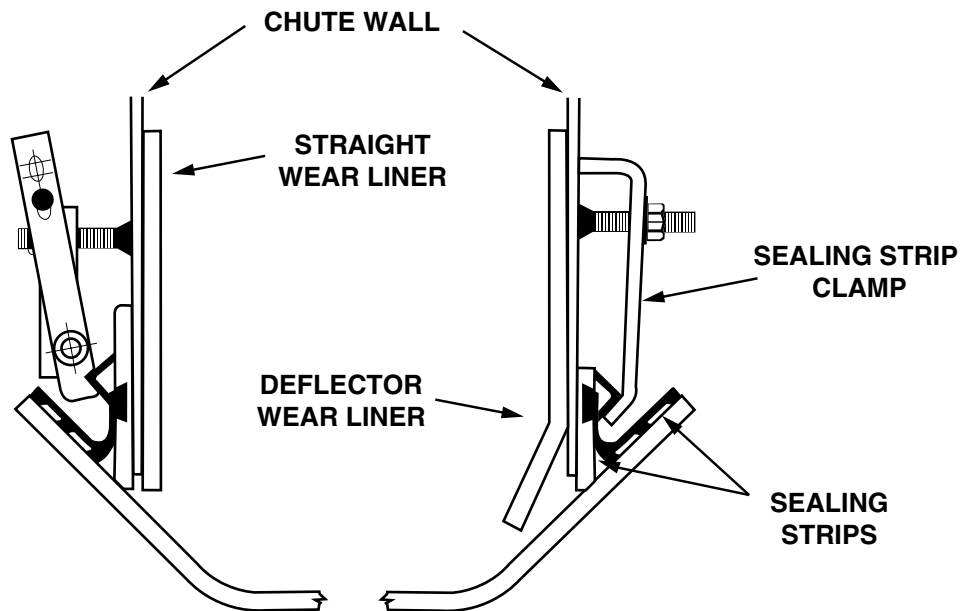


Figure 12.7 Deflector or straight chute liner patterns.

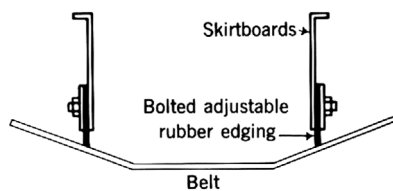


Figure 12.8 Typical application of skirtboard on troughed belt.

Figure 12.9 Continuous skirtboards on a flat belt. Here, frequent skirtboard adjusting is required so that the edging just touches the belt surface. Improper pressure of the rubber skirtboard edging may overload the belt conveyor driving motor.





Figure 12.10 Skirtboards on a troughed belt. Usually, when the loading is in the direction of the troughed belt travel (right), the skirtboard length is a function of the difference between the loading material velocity and the belt velocity.

To avoid the entrapment of material lumps between skirtboards, sealing strips, and belt, the skirtboards should be installed so they open in the direction of belt travel. This gradual widening, often less than one inch (sometimes, even less than one-half inch) over the entire length of the transfer point, provides a relief mechanism for any material that could become entrapped and risk gouging or abrading the moving belt. Rather than being pinched between the skirtboard or lining and the belt, the lumps of material are pulled free by belt motion. It is critical that these openings form a straight line, without any jagged or saw-toothed pattern which could capture material.

Commonly used proportions and details of skirtboards and elastomer sealing strips are as follows:

Spacing of Skirtboards

The maximum distance between skirtboards customarily is two-thirds the width of a troughed belt, ($0.666b$) However, it is desirable, when possible, to reduce this spacing to one-half the width of the troughed belt ($0.5b$) especially for free-flowing materials, such as grain.

On flat belts, depending on how well the belt is trained centrally, how well it is supported by idlers or a loading plate beneath the belt, and how effectively the edge sealing system is maintained, the space between the skirtboards may be only a few inches less than the belt width. Such spacing commonly is used when handling damp or prepared molding sand, or similar materials with minimal slump upon leaving the end of the loading area.

Length of Skirtboards

Usually, when the loading is in the direction of the troughed belt travel the skirtboard length is a function of the difference between the velocity of the loading material—at the moment the material reaches the belt — and the belt speed. For the installation where this difference is small, the length of the skirtboard can safely be 2 ft for each 100 fpm of belt speed, with a minimum length of 3 ft. Otherwise, the skirtboard should be long enough to allow the load to settle into the profile it is to maintain for the rest of its conveyor journey. The need for a dust suppression or collection system may require an increase in the length of skirtboard for use to establish a plenum.

Where belt conveyors with trippers are arranged with inclined loading sections, the skirtboards should extend to the bend pulley or to the first of the group of idlers at the convex vertical curve. This is done to maintain the shape of the material load on the belt, right up to the beginning of the curve.

Skirtboards preferably should terminate above an idler rather than between idlers.

Height of Skirtboards

The height of skirtboards must be sufficient to contain the material volume as it is loaded on the belt. Table 12-1 lists accepted, minimum skirtboard height for 20, 35° and 45 degree three-equal-roll troughing idlers. The requirement to control airborne dust driven off by the forces of loading material may require an increase in the height of skirtboard. In addition these systems may require the lengthwise extension of the skirtboard and its covering, as well as the use of mechanical systems for dust suppression and/or collection.

Skirtboard Clearance over Belt

To minimize fugitive material, skirtboard should come as close to the belt as possible, consistent with safe operation of the belt. This clearance preferably should increase uniformly in the direction of belt travel. As previously explained, this steadily increasing gap permits lumps or foreign objects to move forward without becoming jammed between the lower edge of the skirtboards and the belt.

Greater clearance can be used, but this necessitates wider and thicker rubber-edging strips, particularly for long skirtboards.

Skirtboard Sealing Strips

To prevent the leakage of fines through the clearance between the lower edge of the skirtboards and the moving belt, it is common practice to edge the skirtboard exterior with a flexible sealing strip. These strips, typically an elastomer such as rubber or urethane, are connected to the skirtboard in such a manner to allow for adjustment due to wear. (Refer to Figure 12.8.)

Strips of used conveyor belting should never be used for a skirtboard seal, as the belt carcass may have picked up abrasive fines which can damage the cover of the moving belt.

Sealing strips can be installed vertically or at an angle; it can be installed so it seals on the outer edge of the belt, or so it turns back under the skirtboard. Edging installed at an angle provides a better seal between idlers as the belt flexes under load. However, care must be exercised in design to combine good sealing with minimizing friction and belt cover wear.

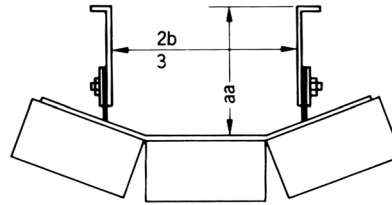


Table 12-1A Minimum uncovered skirtboard height for 20° three-equal-roll troughing idlers.

20° Three-Equal-Roll Troughing Idlers									
Belt width b	aa , Skirtboard height, inches								
	Lump size, inches								
	2*	4	6	8	10	12	14	16	18
18	5.0	5.0	5.0						
24	5.5	5.5	6.0	6.6					
30	5.8	6.3	7.0	7.6	8.3				
36	6.7	7.3	8.0	8.6	9.3	10.0			
42	7.7	8.3	9.0	9.6	10.3	11.0	11.6		
48	8.7	9.3	10.0	10.6	11.3	12.0	12.6	13.3	
54	9.7	10.2	11.0	11.6	12.3	13.0	13.6	14.3	15.2
60	10.7	11.3	12.0	12.6	13.3	14.0	14.6	15.3	16.2
72	12.7	13.3	14.0	14.6	15.3	16.0	16.6	17.3	18.2
84	14.7	15.3	16.0	16.6	17.3	18.0	18.6	19.3	20.2
96	16.7	17.3	18.0	18.6	19.3	20.0	20.6	21.3	22.2

Table 12-1B Minimum uncovered skirtboard height for 35° and 45° three-equal-roll troughing idlers.

35° and 45° Three-Equal-Roll Troughing Idlers									
Belt width b	aa , Skirtboard height, inches								
	Lump size, inches								
	2*	4	6	8	10	12	14	16	18
18	7.0	7.0	7.0						
24	7.5	7.5	8.0	8.6					
30	8.8	8.8	9.5	10.1	10.8				
36	9.7	10.3	11.0	11.6	12.3	13.0			
42	11.2	11.8	12.5	13.1	13.8	14.5	15.1		
48	12.7	13.3	14.0	14.6	15.3	16.0	16.6	17.3	
54	14.7	14.8	15.5	16.1	16.8	17.5	18.1	18.8	19.7
60	15.7	16.3	17.0	17.6	18.3	19.0	19.6	20.3	21.2
72	18.7	19.3	20.0	20.6	21.3	22.0	22.6	23.8	25.0
84	21.7	22.3	23.0	23.6	24.3	25.0	25.6	27.2	28.7
96	24.7	25.3	26.0	26.6	27.3	28.0	28.6	30.6	32.5
*For material that is all fines, use skirtboard heights in 2-inch lump column.									

Edging installed at an angle provides a better seal between idlers as the belt flexes under load. However, care must be exercised in design to combine good sealing with minimizing friction and belt cover wear.

Where the characteristics of the material permit — such as uniform lump sizes greater than 1 inch with no fines — the elastomer sealing strip can be omitted, but only if the skirtboards are not too close to the edge of the belt. Skirtboard edge sealing strips should be adjusted frequently so that the seal just touches the belt surface. Forcing the strip hard against the belt cover can require additional power to move the belt. On conveyors with continuous skirtboards, improper pressure of rubber skirtboard edging may overload the motor driving the conveyor belt.

There are many theories on the hardness required for skirtboard seals. In general, the hardness of the seal should be sufficient to allow it to perform its designed mechanical function. More important than hardness is the selection of a skirtboard seal that is sacrificial and wears before wearing the belt.

Damage near the edge of the belt below the skirtboard is often blamed on the sealing system. However, this damage may not always be the result of too much sealing pressure, but rather abrasion from material entrapped in pinch points. The careful installation of liners, skirtboards, and sealing strips so they form a smooth line, without jagged or sawtooth edges, and which gradually opens in the direction of belt travel can go a long way to eliminate this entrapment damage to the belt.

Skirtboard Covers

Skirtboards can be covered to minimize dusting. Of course, the skirtboard must be high enough to contain the system's peak load and maximum lump sizes. The top edges of the skirtboards can be externally flanged, and the cover fastened to these flanges.

If skirtboard covers are used, especially on skirtboards for inclined belt conveyors, the portion of the cover adjacent to the feed chute should be generously slanted to meet the chute. This is necessary to make room for material which is not yet moving at belt speed and thus avoid a jam in the material flow at the end of the feed chute.

Designs for optimum control of dust typically specify skirtboards be enclosed, to allow for the required dust suppression or collection systems.

Skirtboards for Intermediate Loading Points

Where a belt is loaded at more than one point in the conveyor's length, care must be used in the arrangement of the skirtboards at the intermediate loading points. Because the load of the material tends to flatten and spread as the belt moves, skirtboards at intermediate loading points must be designed to let previously loaded material pass freely. Usually, intermediate-loading point skirtboards are spaced closer together, with a generous clearance above the profile of the previously loaded material. The additional material is centered on the belt and placed on top of the earlier load in the attempt to minimize spillage.

Spillage may occur at intermediate loading points, even with the most careful design of the skirtboards, because of fluctuations in initial loading. Dust creation at intermediate loading points is almost inevitable.

Often, when loading points are relatively close together, it is better to continue the skirtboards between the loading points than to invite the fugitive material problems which can be created by use of relatively short lengths at the intermediate loading points. Continuous skirtboards are good insurance against spillage.

Sometimes the use of a wider-than-normal belt or a more deeply troughed belt will facilitate loading without spillage at intermediate loading points.

Friction Against Skirtboards

The additional conveyor power required to overcome the friction of the material against the skirtboards and to overcome the friction of the skirtboard sealing strips against the belt is discussed in Chapter 6.

Feeders

The cross-sectional areas and capacities given in Chapter 4, Tables 4-2 through 4-5 are based on a continuous, even flow of material. Some variations in the flow may be permitted, provided these variations do not appreciably affect the average flow.

However, intermittent or irregular feeding of the material to the belt will result in alternate empty and overloaded portions of the belt. Such a condition usually causes a loss of capacity and, very likely, spillage of material over the edges of the belt along the overloaded portions.

Some means of feed regulation must be employed, particularly when a belt conveyor is to be loaded from hoppers, bins, or piles. Material discharged to a belt conveyor from other conveyors requires only a suitable transfer chute. However, the rate of feed must be established somewhere in the conveying system.

Feeders can be of various types, such as screw, belt, drag-scraper, apron, reciprocating plate, vibrating, rotary vane or drum, rotary disc, or table feeders.

Apron, drag-scraper, and vane feeders provide a slightly pulsating feed, unless the spacing of the apron pans, drag-scraper bars, or vanes is small in comparison with the volume of material being fed. However, chutes from these feeders to the belt conveyor usually smooth out the pulsations. Retarding devices, suspended in such chutes, help to reduce the pulsation peaks.

The choice of feeders depends upon the characteristics of the material handled, the manner in which the material is stored, and the tonnage rate of feed. Ten basic feeders which are commonly used are briefly discussed below and illustrated in Figures 12.11 through 12.23.

Screw Feeders

A conveyor screw can be located at the bottom of a storage bin to control and regulate the flow of most materials uniformly and continuously, except those materials containing large lumps or highly aerated fines, or those which tend to pack. See Figure 12.11.

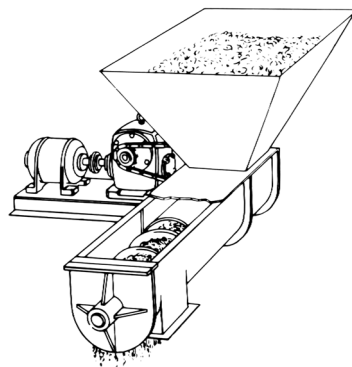


Figure 12.11 Screw feeder.

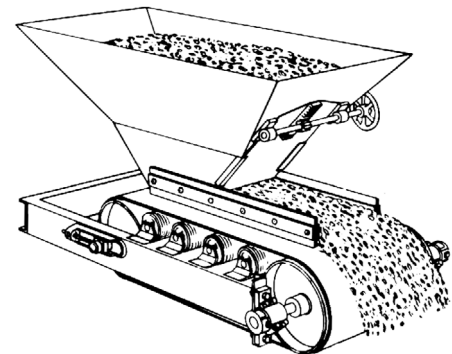


Figure 12.12 Typical belt feeder.

Belt Feeders

A belt feeder is a very short belt conveyor, installed under a storage facility. Generally, the belt is flat and is supported on closely spaced idlers or on a smooth slide plate. Belt feeders are used extensively for handling fine, freeflowing, abrasive, and friable materials. See Figure 12.12.

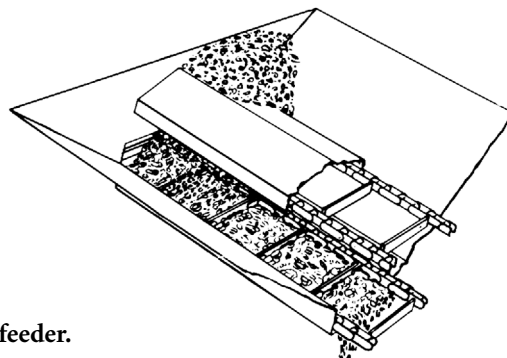


Figure 12.13 Typical drag scraper feeder.

Drag-Scraper Feeders (Bar Drag Feeders)

A drag-scraper feeder consists of a succession of plates or bars mounted between two strands of conveyor chain. The plates or bars drag along the bottom of a trough. A drag-scraper feeder is a simple and compact arrangement for controlling the feed of fine or small-lump material. See Figure 12.13.

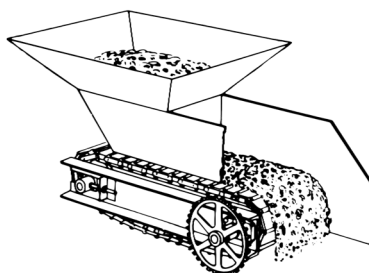


Figure 12.14 Apron feeder.

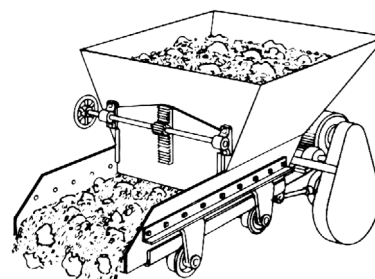


Figure 12.15 Single-plate feeder.

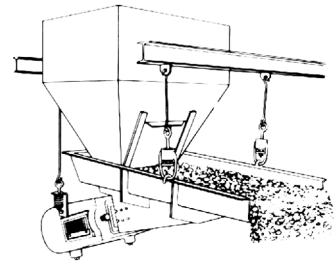
Apron Feeders

Apron feeders are used for handling abrasive, heavy, and lumpy materials. The material is carried on overlapping metal plates or “pans,” mounted on or between strands of conveyor chain. The chains usually are fitted with rollers which ride on metal tracks. See Figure 12.14.

Reciprocating Plate Feeders

The reciprocating plate feeder is one of the oldest devices for feeding materials from bins or hoppers. This feeder consists of a reciprocally driven plate or pan, operating horizontally or slightly declined, and positioned under a head of material in a bin. The reciprocating element can be double-ended, arranged so that each end is under a separate opening in the same bin. Plate feeders will handle fines, a mixture of lumps and fines, or small-sized lumps. See Figure 12.15.

Figure 12.16 Typical electrical vibrating feeder.



Vibrating Feeders

A vibrating feeder consists of a pan or trough to which is imparted a vibrating motion so that the material is impelled in a definite, controlled flow. Normally, it is positioned under the opening in the bottom of a bin or a hole under a storage pile. See Figure 12.16.

Vibrating feeders successfully handle a wide range of materials. However, their use should be avoided where the material is of such nature that it will stick to and build up on the surface of the pan or trough.

Rotary-Vane Feeders (Pocket Feeders)

A rotary-vane feeder consists essentially of a shaft-mounted vane or pocket within a snugly fitted, enclosed case. Rotary-vane feeders, also called pocket feeders, provide an intermittent feed of free-flowing fine or small-lump material. See Figure 12.17.

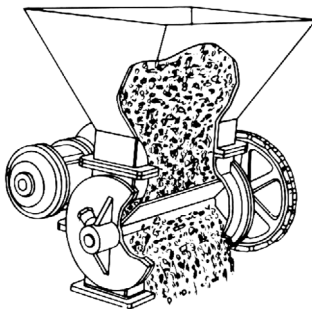


Figure 12.17 Typical vane or pocket feeder.

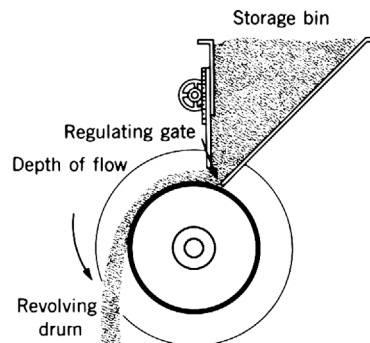


Figure 12.18 Rotary drum feeder.

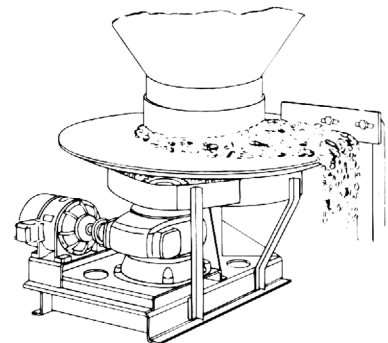


Figure 12.19 Rotary-table feeder.

Rotary-Drum Feeders

Rotary-drum feeders provide a fairly accurate control of the rate of feed (in limited applications) of fine or small-lump material. See Figure 12.18. Where the material handled is sticky or otherwise not free-flowing, the use of drum feeders should be avoided.

Rotary-Table Feeders (Disc Feeders)

A rotary-table, or disc, feeder consists of a horizontal circular plate, with or without a central cone, rotating under a circular opening in the bottom of a conical bin or hopper. The material on the plate is plowed off by an adjustable blade and falls over the edge of the table or disc. Rotary-table feeders are used for materials that have a tendency to arch. See Figure 12.19.

Traveling Rotary-Plow Feeders

Traveling rotary-plow feeders are suitable for use in tunnels, under storage piles, or under long storage bins. The plowing mechanism consists of a number of curved arms, operating on a vertical axis, and arranged to sweep material off a narrow shelf running the length of the storage pile or bin. See Figure 12.20.

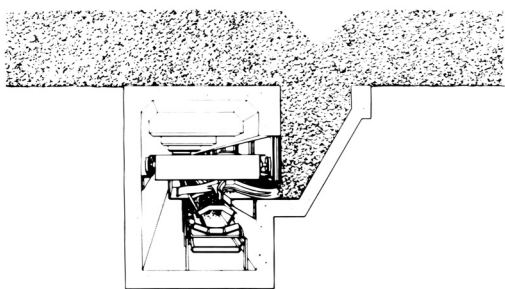


Figure 12.20 Traveling rotary plow.

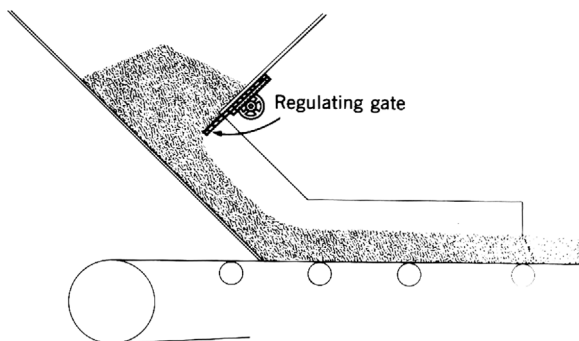


Figure 12.21 Typical regulating gate feeder.

Feed-Control Gates

If the material is fine and very free-flowing, the judicious use of a simple, adjustable cut-off gate usually will satisfactorily control the flow of such material. These gates are used in reclaiming tunnels beneath storage piles or under overhead bins. They are made in several designs, plain or with attached loading chute and skirt-boards, and with or without separate cut-off or regulating gates. Several tunnel gates and chutes can be arranged in succession to blend different grades of material on the belt conveyor. See Figures 12.21 and 12.22. Feed-control gates should not be used if the character of the bulk material is variable, as it will be practically impossible to maintain the proper gate adjustment.

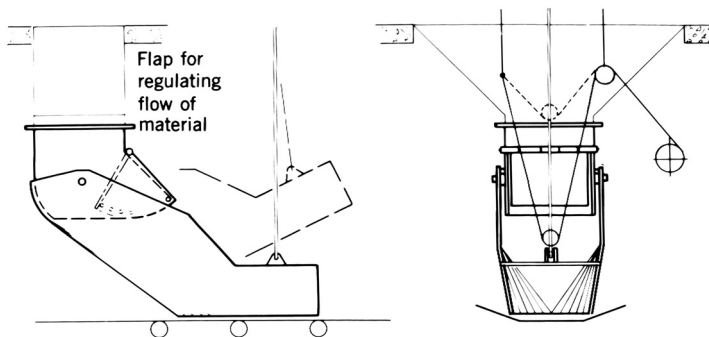


Figure 12.22 Tunnel gate.

Methods of Discharging from the Belt

Materials carried by a belt conveyor can be discharged from the belt in different ways to effect certain desired results. The discharge can be accomplished at a definite point or points, or it can extend alongside the belt conveyor, on one or both sides, for considerable distances. The flexibility of discharge arrangements of belt conveyors facilitates their use in the maximum fill of long bins and in the creation of large and variously shaped storage piles.

The simplest method of discharge from a conveyor belt is to let the material pass over an end pulley and fall onto a pile. By adding a suitable chute, the discharge can be directed, as desired, to a pile, a bin, or another conveyor. A fork in the discharge chute, with a gate, will permit the material to flow simultaneously in two directions or alternately in either direction.

If several specific points of discharge are required, the conveyor belt can be passed over fixed trippers, which will affect the discharge at these points. A movable tripper can be used to discharge material at many points on either side of the belt, or to one side, continuously or intermittently.

When a flat belt is used, it is possible to position plows at the desired discharge points and direct the material from the belt to either side or simultaneously to both sides of the belt. Such plows can be fixed in position, or can be movable to affect a wider range of discharge points. A troughed belt can be temporarily flattened to permit discharge by a plow.

Whenever the material is discharged over an end pulley, the speed of the belt and the diameter of the end pulley are factors which determine the path of the discharged material. This path is called a trajectory. The shape of discharged material trajectories is important when designing discharge chutes, and when the material falls freely without a chute onto a stockpile.

Care in the design of discharge chutes will pay off handsomely in the successful operation of a belt conveyor. Ingenious combinations of discharge chutes and gates will facilitate diversion of the material in desired directions, collection of the material adhering to the belt, avoidance of spilled material, and control of dusting in the case of dry or powdery and fine materials.

Various common discharge arrangements are illustrated in Chapter 2, Figures 2.11 through 2.18.

Discharge Over-the-End Pulleys

This simple discharge is commonly used. The trajectory of the discharged material should be investigated to ascertain where the material will fall. Provision must be made so that the pile will not rise to the end pulley and cause the vulnerable cover of the belt to scrub against the top of the pile.

Dusting of dry discharged material may be a problem. Figure 12.23 shows a simple over-the-end pulley discharge.

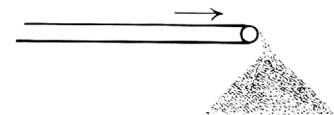


Figure 12.23 Discharge over-the-end pulley.

Discharge Chutes

The character of discharge chutes is limited only by the ingenuity of the belt conveyor designer. Discharge chutes may be extremely simple or very complex. The prime requirement is that the chute collects all the material discharged and cleaned from the belt. The carrying surface of the belt should be free from any adhering material as it passes back over the return idlers.

Some materials will part freely from the surface of the belt as the belt flexes in passing over the end pulley. Other materials will adhere to the belt. For a discussion of belt-cleaning devices, see Chapter 11.

The upper end of the discharge chute must enclose the cleaning device and catch the material cleaned from the belt. Otherwise, a separate dribble chute can be employed.

Gates frequently are provided in the discharge chute to divert the flow of the discharged material.

If the material handled is abrasive, provisions must be made in the discharge chute for renewing the wearing surfaces. This is particularly true if the material is heavy and has sharp-edged lumps. Corrosion-resistant materials may be required for certain chemicals, to avoid both damage to the chute and contamination of the chemical.

Some of the most common forms of discharge chutes are illustrated and described below.

A typical simple discharge chute is shown in Figure 12.24. The top of the chute can be provided with a cover, or even an air suction vent, if dusting of the material is a problem. The upper edge of the bottom plate, or the “lip,” should be far enough under and back of the pulley to catch all the material, even when the belt is barely moving. To avoid abrading or gouging the belt cover, the arrangement must be such as to eliminate the danger of a hard lump becoming wedged between the lip and the belt on the pulley.

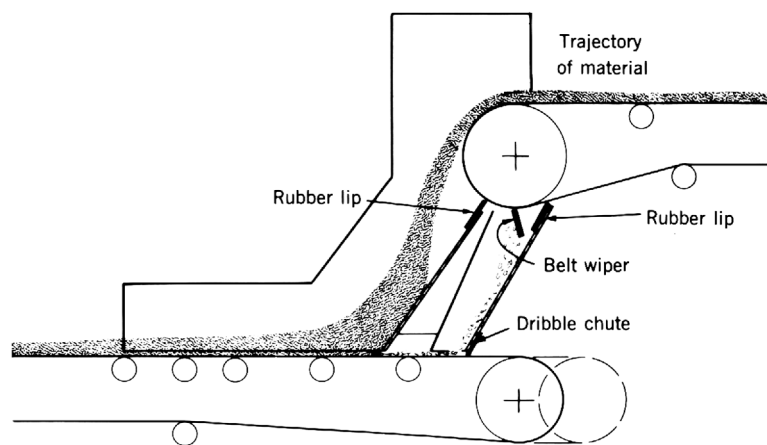


Figure 12.24 Typical simple discharge chute.

If hard, jagged lumps are contained in the material, it is good practice to provide a strip of rubber on the lip, as shown in Figure 12.24. Should the belt run to one side of the pulley, the sides of the chute should be flared where the belt enters, to minimize damage to the edges of the belt.

The chute design must be integrated with the trajectory of the material leaving the belt so that the material will fall on the sloping bottom of the chute and not on a succeeding conveyor belt.

Serious injury to a conveyor belt may result from failure of the material to fall freely at the discharge point. Such failure can be caused by overfilling a bin; by stoppage of the succeeding conveyor or apparatus receiving the discharged material; or by plugging of the discharge chute, as when sticky material refuses to slide on the sloping surfaces. Whenever such a condition is a possibility, it is advisable to provide an automatic protective device, such as a paddle switch. This should be located so that any undesirable accumulation of material will deflect the paddle to shut down the system, or signal a warning to the operator.

A correctly designed chute allows the free, smooth flow of the discharged material. It avoids abrupt changes of direction which invite material build-up and subsequent plugging.

Lowering Chutes

Where dusting and degradation are objectionable, lowering chutes are used for directing materials to storage. Four common forms of lowering chutes are illustrated and described as follows:

Spiral Lowering Chutes

Figure 12.25 shows the usual form of a spiral lowering chute used for gently lowering fragile and/or dusty bulk materials.

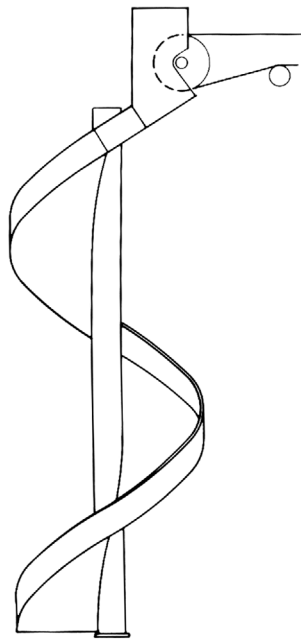


Figure 12.25 Spiral lowering chute.

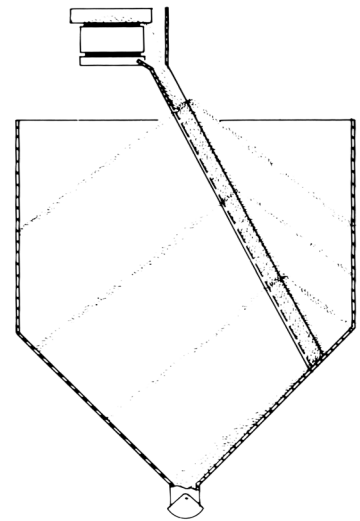


Figure 12.26 Bin lowering chute.

Bin Lowering Chutes

Straight, declined, inverted channel chutes run from the belt conveyor discharge point down into the bin, and are secured to the sloping side of the bin near the bin bottom. The slope of the chute to the horizontal must be 10 degrees to 15 degrees greater than the angle of repose of the material. Material will slide down these chutes quietly, without dusting, until it meets with the bin side or the surface of the material in the bin. There, the material will leave the chute and spread out conically. Refer to Figure 12.26.

When properly arranged, these bin lowering chutes will eliminate dusting and fill the bin to its practical capacity.

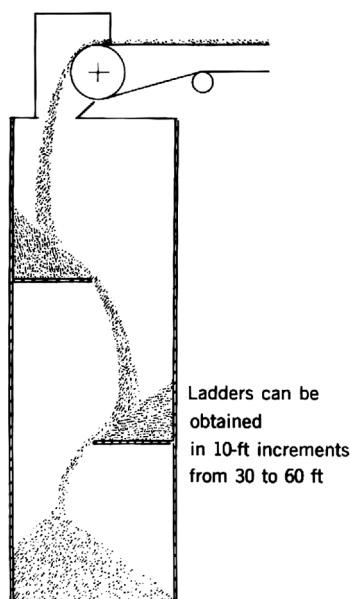


Figure 12.27 “Rock ladder” lowering chute.

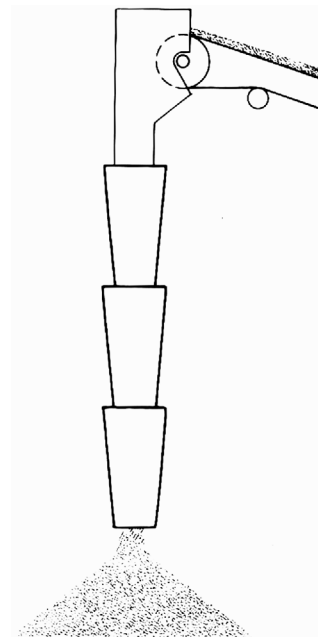


Figure 12.28 Telescopic chute.

Rock Ladders

Many specifications, particularly federal specifications, do not permit breaking, crumbling, or degrading rock as it is delivered to a stockpile by the belt conveyor. Good practice calls for the installation of a “rock ladder” under the discharge point of the belt conveyor. Figure 12.27 illustrates one form of the rock-ladder lowering chute.

The rock ladder is a structural steel (or wood) tower having a series of baffles arranged in such a manner that rock discharged from a belt conveyor never has a free drop of more than 5 feet.

If the material is heavy, abrasive, and lumpy, the baffles can be arranged in the form of rock boxes, as illustrated in Figure 12.27.

Telescopic Chutes

The telescopic discharge chute is used to minimize dusting when discharging to a pile. Telescoping sections usually are cable-connected in such a manner that a winch will successively lift the sections to keep the lower end of the chute just clear of the top of the storage pile. Coal is often delivered to storage piles through telescopic chutes. Figure 12.28 illustrates a typical arrangement.

Trippers

Trippers are devices used to discharge bulk materials from a belt conveyor at points upstream from the head pulley. Essentially, a tripper consists of a frame supporting two idling pulleys, one above and forward of the other. The conveyor belt passes over and around the upper pulley and around and under the lower pulley. The belt usually inclines to the upper pulley and may run horizontally, or it may then incline again from the lower pulley. Figure 12.29 illustrates a typical motor-driven belt conveyor tripper.



Figure 12.29 Typical motor-driven belt conveyor tripper.

With this construction, the material handled by the belt is discharged to a chute as the belt wraps around the upper pulley. The chute can be arranged to catch and divert the discharged material in any desired direction. The chute arrangement, augmented by movable gates, can be such as to discharge the material to either or both sides of the belt conveyor, or even back again onto the conveyor belt beyond the tripper.

Trippers can be stationary (fixed) or movable (Figures 12.30 and 12.31). Stationary trippers are used where the discharge of material is to occur at a specific location. More than one stationary tripper can be used on a belt conveyor, either to discharge

material from the belt at definite locations or to direct the material back again to the conveyor belt for discharge over the succeeding fixed tripper or the head pulley.

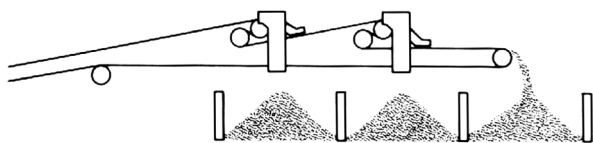


Figure 12.30 Stationary trippers.

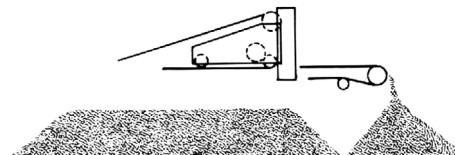


Figure 12.31 Movable tripper.

Movable trippers have the tripper frame mounted on flanged wheels which engage parallel rails supported on either side of the belt, usually from the conveyor frame. Such movable trippers can be moved by a cable and winch, by the belt itself, or by an electric motor mounted on the tripper. These movable trippers are reversible. The belt and motor-driven trippers frequently are arranged to move continuously back and forth along chosen portions of the belt conveyor, reversing automatically at the ends of their travel. All movable trippers can remain in specific locations for a short time or can be locked in position for longer periods.

Often, movable trippers have a platform to transport an operator, or a crosswalk permitting the operator to cross from one side of the conveyor belt to the other. Controls for the movement of the tripper are located for convenient access by the operator.

When handling dusty material, such as dry, fine, bituminous coal, conveyor belt and motor-driven trippers are provided with seals near the lower ends of the discharge chutes. These seals prevent the escape of dust from the covered bins or hoppers into which the chutes discharge.

There are several forms of seals, including a moving “carpet” of rubber-covered fabric belt, a pair of overlapping rubber-covered fabric belt strips, and a stationary carpet of rubber-covered fabric belt. With the stationary carpet type of seal, the tripper carries pulleys which pick up the carpet and then pass it around the discharge chute.

To increase the utility of the tripper in forming long piles or windrows of material on one or both sides of the belt conveyor, the material discharged over the upper pulley of the movable tripper is directed either to a transverse, horizontal, reversible belt conveyor carried by the tripper, or to one of two transverse, inclined belt conveyors also carried by the tripper.

All trippers absorb a certain amount of power from the belt conveyor drive, because the conveyor belt flexes over the tripper pulleys. Movable trippers that are actuated by the conveyor belt itself absorb greater amounts of power from the conveyor drive. For horsepower requirements of trippers, refer to Chapter 6. Figures 12.32 through 12.33 portray the various movable tripper arrangements and their uses.

Stationary (Fixed) Tripper

Stationary or fixed trippers pile material on either or both sides of the conveyor belt at specific locations, or return the material to the belt. See Figure 12.30.

Typical Movable Tripper

Driven by the conveyor belt itself, by an electric motor, or by a cable and winch, this tripper moves in a forward and reverse direction to fill a bin from end to end or to make a long pile on each or one side of the belt conveyor.

The movable tripper can be locked in any number of specific locations for spot discharge. Typical movable trippers are shown in Figures 12.31 and 12.32.

Figure 12.32 Two typical movable trippers.

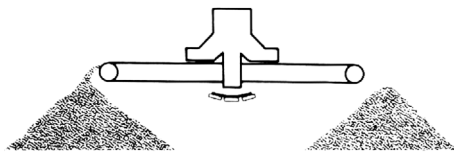
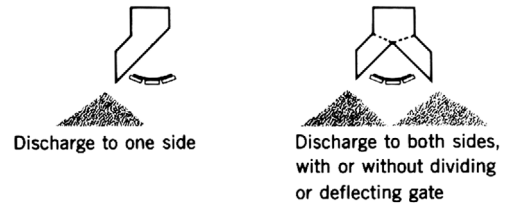


Figure 12.33 Movable tripper with reversible cross belt.



Figure 12.34 Typical tripper with two transverse stacker belts.

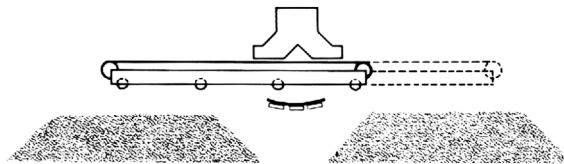


Figure 12.35 Typical movable tripper with reversible shuttle belt .

Tripper Discharge Through Auxiliary Arrangements

Three auxiliary tripper arrangements are described in the paragraphs below.

Tripper with Reversible Cross Belt

With this arrangement, a movable tripper carries a horizontal reversible cross belt. By means of a gate in the bifurcated tripper chute and a corresponding travel direction of the cross belt, material discharged from a belt conveyor by the tripper can be placed in long piles or windrows on either side of the tripper. Or, if desired, individual piles of various materials or grades of the same material can be stockpiled on either side of the tripper. See Figure 12.33.

Plows

Tripper with Two Transverse Stacker Belts

To form higher piles than is possible with a horizontal cross belt, the tripper can be outfitted with two inclined stacker belt conveyors. With this arrangement, continuous high piles can be built on either side of the tripper or spot stockpiles can be built as desired or required. See Figure 12.34.

Tripper with Reversible Shuttle Belt

This arrangement is similar to the tripper with the plain cross belt. However, here the cross belt is a shuttle or movable belt conveyor and is reversible in direction, as shown in Figure 12.35. With this arrangement, flat-topped piles with a large top area can be built as continuous windrows on either side of the tripper, or as spotted stockpiles.

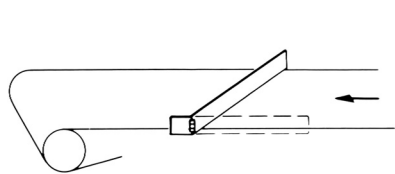


Figure 12.36 Horizontal swing plow.

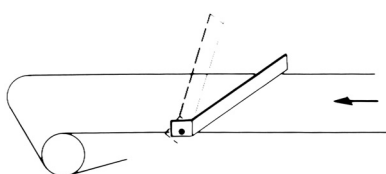


Figure 12.37 Single lift plow.

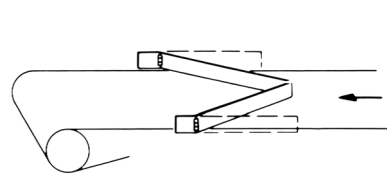


Figure 12.38 Horizontal V-plow.

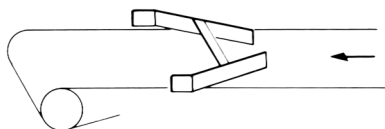


Figure 12.39 Vertical V-plow.

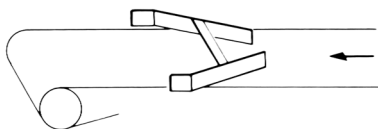


Figure 12.40 Proportioning V-plow.

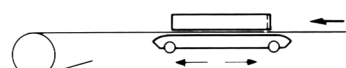


Figure 12.41 Traveling V-plow.

Plows

Discharge of fine materials from flat, horizontal belt conveyors can be accomplished by plows, either stationary or movable with cable and winch. It is also possible to plow material from flat, inclined belt conveyors, if the angle of incline is not too great and the material is not too free-flowing.

Essentially, discharge plows are metal plates set at an angle not exceeding 35° to the belt center line, fitted at the bottom edge with an adjustable, all-rubber edging. There are a number of common arrangements of plow blades to accomplish discharge to one side, to both sides simultaneously, or to both sides simultaneously but with a portion of the material left on the central part of the belt.

The portion of the belt under the plow customarily is supported on a flat metal slide plate. This keeps the belt flat so that the plow can effect a reasonably clean discharge. Plows seldom remove 100 percent of the material from a belt, so a dribble of material is likely to pass over the end pulley of the conveyor. A means to collect this dribble must be provided.

Figures 12.36 through 12.41, provided above, illustrate various plow discharge arrangements.

Plows Discharging to One Side

Plows used to discharge to one side of a flat belt conveyor can swing horizontally across the belt on a vertical pivot or lift vertically on a horizontal pivot.

Single Horizontal Swing Plow

By a horizontal movement of the plow, all or part of the material can be removed from the belt. Plows removing all of the material from the belt can be manually operated or powered by a pneumatic or hydraulic cylinder. See Figure 12.36.

Single Lift Plow

This type of plow is lowered to discharge material from the belt or raised to avoid discharge. Again, all or part of the material can be removed from the conveyor belt, and the plow can be manually or power operated. Refer to Figure 12.37.

Plows Discharging to Both Sides

Plows which discharge simultaneously to both sides of a flat belt conveyor are called “V” plows. Two blades moving horizontally on vertical pivots can form the V, or a fixed V can be raised or lowered on a horizontal pivot.

Horizontal V-Plow

Figure 12.38 shows the horizontal V-plow. The blades swing on vertical pivots and join at the center. This arrangement makes possible several combinations of discharge: part of the material to one side, part to the other side, or all of the material from the belt. The blades can be manually or power operated.

Vertical V-Plow

A fixed V can be angularly raised or lowered to remove all material. Manual or power operation can be used. See Figure 12.39.

A variation of this plow is the open V, which allows some of the material to remain on the central portion of the belt, while discharging the outer portions of the material stream at both sides of the belt. Sometimes the amount of opening at the apex of the V is adjustable. Such a plow is called a proportioning V-plow. This type of plow is illustrated in Figure 12.40.

Traveling V-Plow

Moved by cable and winch in either direction, this V-plow discharges to both sides of a flat belt conveyor in any number of specific positions. See Figure 12.41.

This plow has been adapted to troughed belts by use of a flat belt support plate that lifts the belt from the troughing idlers. Such an arrangement conserves head room.

Discharge Trajectories

The path of the material discharged over the end pulley of a belt conveyor is known as its trajectory. The curvature of this path is determined by the rotational speed and radius of the end pulley, and by the force of gravity.

The proper design of transfer chutes, including the location of chute covers and wearing plates, depends upon the shape of the trajectory. Thus, the trajectory of the discharged material must be predicted as accurately as possible.

A number of authors and most catalogs prepared by belt and conveyor manufacturers have offered methods of calculating and plotting the material trajectory. However, observations and photographs of actual material trajectories from belt conveyors are not in satisfactory agreement with any of the trajectories calculated by these methods. A careful review of the literature on calculating trajectories has led to further research on this subject. The result of this research forms the basis for the method of calculating and plotting trajectories described in this manual.

The trajectories calculated by this method are in close agreement with photographs and observations of actual trajectories of material discharged from belt conveyors.

Allowance has been made for the change in the load shape of bulk material carried on a troughed belt conveyor when the belt flattens as it approaches the discharge pulley. The material tends increasingly to slump laterally toward the belt edges as the belt flattens at the pulley. Here, the load cross-section shape becomes — for all practical purposes — a segment of a circle. The cross-sectional area of this segment is equal to the average cross-sectional area of the load on the troughed portion of the conveyor.

The forces acting on the material as it reaches the pulley must be taken at the center of mass. This practically is the center of gravity of the cross-section of the load shape. A method has been developed to determine the height of this center of gravity above the belt surface, for various capacity loads, on any width of belt and 20 degree, 35 degree, and 45 degree three-equal-roll troughing idlers. Also, the effective radius about the center of the discharge pulley can be easily determined. The median line of the trajectory is established, thus permitting a close approximation to the upper and lower limits of the path of the discharged material stream.

If the material has an apparent density of 50 pounds or more per cubic foot and is of roughly uniform particle size, the upper and lower limits of the material path will be relatively parallel to the median line for free falls of the discharged material to 7 feet below the center of the discharge pulley. If the free fall is greater than 7 feet and up to 20 feet below the center of the discharge pulley, the upper and lower limits of the material path may diverge a little.

Light, fluffy materials, very high belt speeds, and a mixture of large lumps and fines will alter the upper and lower limits of the material path. Lumps riding near the top of the material at the discharge pulley will be thrown farther from the discharge pulley. The trajectory of any such lumps can be closely approximated by calculating and plotting their individual trajectories. Air resistance will cause light, fluffy materials to spread vertically and laterally as they are being discharged over the pulley. Allowance should be made accordingly when determining the upper, lower, and lateral limits of such material trajectories.

Calculating and Plotting Normal Material Trajectories

The method of calculating and plotting the trajectories of normal materials involves careful attention to the following five considerations:

Center of Mass

The point in the mass of the material at which all forces act is the center of gravity of the load-shape cross section, at the point where the moving conveyor belt becomes tangent to the pulley.

Velocities

The material load and the belt are moving at the same linear velocity, up to the point where the belt becomes tangent to the pulley.

The tangential velocity acquired by the material taken at its cross-sectional center of gravity, as the material meets the curvature of the belt on the pulley, is proportional to the radial distance between the cross-sectional center of gravity of the material and the center of the pulley, for any given speed of discharge-pulley rotation.

Start of Trajectory

The start of the trajectory is determined by that point on the circumference of the circle about the pulley center, formed by the cross-sectional center of gravity, where the centrifugal force at that point equals the radial component of the gravity force at that point.

Load Shape

Immediately prior to the point of tangency of the belt and pulley, the material load shape of a troughed belt conveyor can be closely approximated by a segment of a circle, as shown in Figure 12.42.

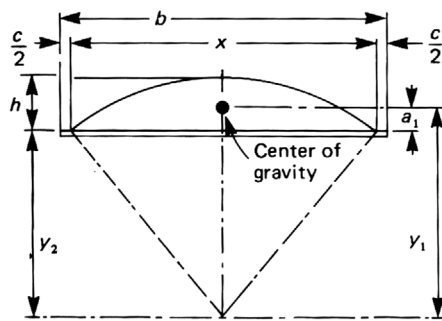


Figure 12.42 Area of circular segment is equal to the cross-sectional area of the material load on the normal troughed portion of the conveyor belt.

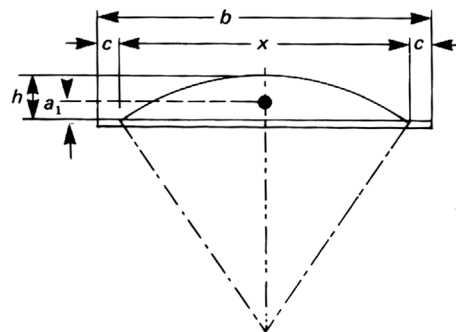


Figure 12.43 To determine the center of gravity of the cross-section of the load on a flat belt, refer to Table 12-2 for values of a_1 and h .

The area of the circular segment in Figure 12.42 is equal to the cross-sectional area of the material load on the normal troughed portion of the conveyor belt.

The center of gravity of the cross section of load on a flat belt can be determined as shown in Figure 12.43. The height of this point, a_1 , and the height of h are given in Tables 12-2A and B.

NOTE: Standard edge distance applies here.

Discharge Trajectories

Angular Tangent Direction

The angular direction of the trajectory is determined by the forces acting on the material at its center of mass (i.e., the cross-sectional center of gravity in Figures 12.42 and 12.43).

Were it not for the effect of gravity, the median line of the trajectory would be a straight line. It is this straight line tangent to a circle, the radius of which is the distance from the pulley center to the center of gravity of the material load-shape cross section, that determines what angular direction the trajectory will take. See Figures 12.41 through 12.47, which are provided below.

Fundamental Force-Velocity Relationships

Fundamentally, if the tangential velocity is V , ft per second (fps), if g is the acceleration due to gravity (32.2 ft per sec²); r is the radial distance in feet from the center of the pulley to the center of mass (i.e., the cross-sectional center of gravity of the material load shape); and W is the gravity weight force of the material acting at the center of mass, then the centrifugal force acting at the center of mass of the material is as follows:

Table 12-2A Load height and center of gravity at discharge pulley, idlers at 20° and 35°

Type of Idlers	<i>h</i> and <i>a_I</i> values, inches are based on cross section loads from Tables 4-2 through 4-5. Troughed belts are per profile shown in Figure 12.42. Flat belts are per profile shown in Figure 12.43.												
	Surcharge Angle	Belt Width, inches											
			18	24	30	36	42	48	54	60	72	84	96
20° Three-equal-length-roll troughing idlers	0°	<i>h</i>	1.1	1.7	2.1	2.7	3.1	3.7	4.1	4.7	5.7	6.9	7.9
		<i>a_I</i>	0.4	0.7	0.8	1.1	1.3	1.5	1.7	1.9	2.3	2.7	3.1
	5°	<i>h</i>	1.4	2.0	2.6	3.2	3.9	4.5	5.1	5.6	6.8	8.2	9.5
		<i>a_I</i>	0.6	0.8	1.0	1.3	1.6	1.8	2.1	2.3	2.7	3.2	3.7
	10°	<i>h</i>	1.6	2.4	3.1	3.7	4.5	5.2	6.0	6.7	8.1	9.6	11.0
		<i>a_I</i>	0.6	1.0	1.3	1.5	1.8	2.1	2.4	2.7	3.3	3.9	4.5
	20°	<i>h</i>	2.2	3.1	3.8	4.8	5.7	6.8	7.5	8.5	10.4	12.3	14.1
		<i>a_I</i>	0.9	1.3	1.6	1.9	2.3	2.7	3.0	3.4	4.2	4.9	5.6
	25°	<i>h</i>	2.5	3.5	4.5	5.4	6.4	7.5	8.5	9.6	11.6	13.7	15.7
		<i>a_I</i>	1.0	1.4	1.8	2.2	2.6	3.0	3.4	3.8	4.6	5.4	6.2
	30°	<i>h</i>	2.7	3.8	5.0	6.0	7.1	8.2	9.4	10.6	12.7	15.1	17.3
		<i>a_I</i>	1.1	1.5	2.0	2.4	2.9	3.3	3.8	4.3	5.1	6.1	7.1

Table 12-2A Load height and center of gravity at discharge pulley, idlers at 20° and 35°

Type of Idlers	<i>h</i> and <i>a_I</i> values, inches are based on cross section loads from Tables 4-2 through 4-5. Troughed belts are per profile shown in Figure 12.42. Flat belts are per profile shown in Figure 12.43.												
	Surcharge Angle	Belt Width, inches											
			18	24	30	36	42	48	54	60	72	84	96
35° Three-equal-length-roll troughing idlers	0°	<i>h</i>	1.9	2.7	3.5	4.3	5.0	6.0	6.7	7.4	9.0	10.9	12.5
		<i>a_I</i>	0.8	1.1	1.4	1.7	2.0	2.4	2.7	3.0	3.6	4.1	4.7
	5°	<i>h</i>	2.1	3.0	3.8	4.8	5.5	6.4	7.4	8.1	10.0	12.0	13.8
		<i>a_I</i>	0.8	1.2	1.5	1.9	2.2	2.6	3.0	3.3	4.0	4.7	5.4
	10°	<i>h</i>	2.3	3.2	4.2	5.2	6.1	7.2	8.2	9.0	11.0	13.1	15.1
		<i>a_I</i>	0.9	1.3	1.7	2.1	2.5	2.9	3.3	3.6	4.4	5.2	6.0
	20°	<i>h</i>	2.7	3.9	5.1	6.1	7.3	8.4	9.5	10.7	13.0	15.4	17.7
		<i>a_I</i>	1.1	1.6	2.1	2.5	2.9	3.4	3.8	4.3	5.2	6.1	7.0
	25°	<i>h</i>	3.0	4.2	5.4	6.6	7.8	9.1	10.3	11.6	13.9	16.6	19.0
		<i>a_I</i>	1.2	1.7	2.2	2.7	3.1	3.7	4.1	4.6	5.6	6.6	7.6
	30°	<i>h</i>	3.2	4.5	5.7	7.1	8.4	9.8	11.1	12.4	14.9	17.7	20.4
		<i>a_I</i>	1.3	1.8	2.3	2.9	3.4	4.0	4.5	5.0	6.0	7.0	8.0

$$\text{Centrifugal Force} = \left(\frac{W}{g}\right)\left(\frac{V_s^2}{r}\right) = \frac{WV_s^2}{gr}$$

When this centrifugal force equals the radial component of the material weight force, the material will no longer be supported by the belt and will commence its trajectory. At just what angular position around the pulley this will occur is governed by the slope of the conveyor at the discharge pulley, outlined in the following three cases.

Table 12-2B Load height and center of gravity at discharge pulley, idler at 45° and flat belt idler.

Type of Idlers	<i>h</i> and <i>a_I</i> values, inches are based on cross section loads from Tables 4-2 through 4-5. Troughed belts are per profile shown in Figure 12.42. Flat belts are per profile shown in Figure 12.43.												
	Surcharge Angle	Belt Width, inches											
			18	24	30	36	42	48	54	60	72	84	96
45° Three equal-length-roll troughing idlers	0°	<i>h</i>	2.2	3.1	4.0	5.0	6.0	6.8	7.7	8.6	10.5	12.7	14.7
		<i>a_I</i>	0.9	1.3	1.8	2.0	2.4	2.7	3.1	3.5	4.2	5.0	5.8
	5°	<i>h</i>	2.4	3.4	4.4	5.4	6.4	7.4	8.4	9.5	11.4	13.7	15.7
		<i>a_I</i>	1.0	1.4	1.8	2.2	2.6	3.0	3.4	3.8	4.6	5.4	6.2
	10°	<i>h</i>	2.6	3.7	4.7	5.8	7.0	7.9	9.0	10.2	12.2	14.6	16.8
		<i>a_I</i>	1.0	1.5	1.9	2.3	2.8	3.2	3.6	4.1	4.9	6.8	7.7
	20°	<i>h</i>	2.9	4.1	5.4	6.6	7.8	9.1	10.3	11.5	14.0	16.5	19.0
		<i>a_I</i>	1.2	1.7	2.2	2.7	3.1	3.7	4.2	4.6	5.6	6.6	7.6
	25°	<i>h</i>	3.1	4.4	5.7	7.0	8.3	9.7	10.9	12.3	14.8	17.5	20.1
		<i>a_I</i>	1.2	1.8	2.3	2.8	3.3	3.9	4.4	4.9	5.9	6.9	7.8
	30°	<i>h</i>	3.3	4.7	6.0	7.4	8.8	10.2	11.5	13.0	15.6	18.5	21.3
		<i>a_I</i>	1.3	1.9	2.4	3.0	3.5	4.1	4.6	5.2	6.3	7.5	8.7
Flat belt idlers	5°	<i>h</i>	0.35	0.42	0.50	0.66	0.80	0.90	1.04	1.15	1.40	1.65	1.95
		<i>a_I</i>	0.14	0.17	0.20	0.27	0.32	0.36	0.42	0.46	0.57	0.69	0.71
	10°	<i>h</i>	0.65	0.85	0.95	1.35	1.60	1.80	2.10	2.30	2.80	3.30	3.80
		<i>a_I</i>	0.26	0.35	0.38	0.54	0.65	0.73	0.85	0.93	1.13	1.35	1.70
	15°	<i>h</i>	1.00	1.30	1.43	2.00	2.40	2.67	3.16	3.45	4.20	5.10	6.00
		<i>a_I</i>	0.40	0.52	0.58	0.81	0.97	1.08	1.28	1.39	1.70	2.09	2.48
	20°	<i>h</i>	1.30	1.70	1.90	2.70	3.20	3.56	4.20	4.60	5.55	6.45	7.35
		<i>a_I</i>	0.52	0.69	0.77	1.09	1.29	1.44	1.70	1.86	2.26	2.66	3.07
	25°	<i>h</i>	1.60	2.15	2.37	3.35	4.00	4.43	5.30	5.74	6.93	7.99	8.05
		<i>a_I</i>	0.65	0.87	0.96	1.35	1.61	1.79	2.14	2.32	2.80	3.29	3.78
	30°	<i>h</i>	1.94	2.57	2.83	4.03	4.80	5.30	6.30	6.86	8.30	9.78	11.26
		<i>a_I</i>	0.78	1.04	1.14	1.63	1.94	2.14	2.54	2.77	3.35	3.95	4.55
	35°	<i>h</i>	2.25	3.00	3.30	4.70	5.60	6.20	7.40	8.00	9.70	11.50	13.30
		<i>a_I</i>	0.91	1.21	1.33	1.90	2.26	2.50	2.98	3.23	3.91	4.65	5.76

Horizontal Belt Conveyor Trajectories

If the belt conveyor is horizontal to the discharge pulley, there are two conditions to consider:

1. If the tangential speed is sufficiently high (that is, when the centrifugal force is equal to or greater than W), the material will leave the belt at the initial point of tangency of the belt with the pulley, as shown in Figure 12.44, or:

$$\frac{V_s^2}{gr} \geq 1$$

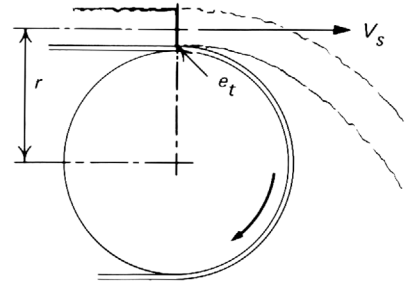


Figure 12.44 When the belt speed is sufficiently high, the material leaves the belt at the point of tangency of the belt with the pulley.

Where:

e_t = the point where the material leaves the belt

V_s = tangential velocity, fps, of the cross-sectional center of gravity of load shape

g = acceleration of gravity in feet per second per second, or ft per sec²

r = radius in feet from the center of pulley to the cross-sectional center of gravity of the load shape

2. If the tangential speed is not high enough for the material to leave the belt at the initial point of tangency (that is, when V_s^2/gr is less than 1), then the material will follow part way around the pulley for an angular distance, γ :

$$\frac{V_s^2}{gr} = \cos \gamma$$

Where:

e_t = the point where the material leaves the belt

γ = angle, in degrees, between the vertical centering, through the pulley, to the point, e_t , where the material starts its trajectory

This is illustrated clearly in Figure 12.45.

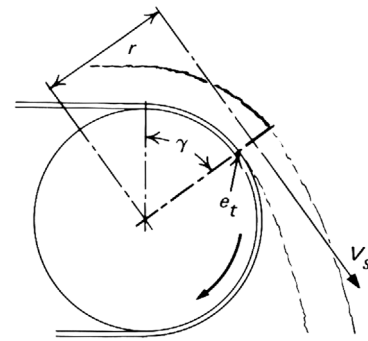


Figure 12.45 When the belt speed is not high enough, the material will follow part way around the conveyor.

Inclined Belt Conveyor Trajectories

For a belt inclined to the discharge pulley, there are four conditions to consider:

1. If the tangential speed is sufficiently high, or when $V_s^2/gr > 1$, the material leaves the belt at the initial point of tangency of the belt and pulley, as shown in Figure 12.46,

Where:

e = point where the material leaves the belt

ϕ = angle, in degrees, of incline of the belt conveyor to the horizontal

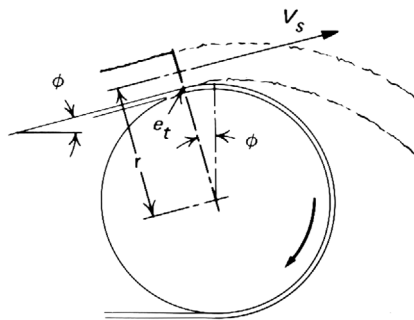


Figure 12.46 When, in an inclined conveyor, the tangential speed is high (see text), the material will leave the belt at the point of tangency of the belt and pulley.

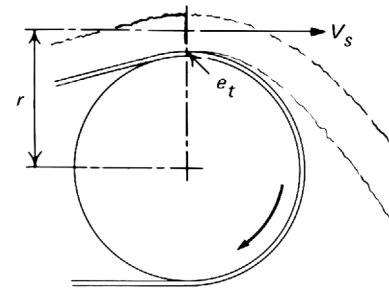


Figure 12.47 When, in an inclined conveyor, the tangential velocity is equal to a specific value (see text), the material will leave the belt at the top of the end pulley.

2. If the combination of the belt incline, the pulley diameter, the load depth, and the tangential belt speed is such that $V_s^2/gr > \cos \phi$ but is still less than 1, the material may leave the belt at the initial point of tangency of the belt and the pulley. However, the curved surface of the belt on the pulley may interfere with the theoretical

trajectory path of the material. The material may then re-engage the belt and be carried farther around the pulley before it assumes its final trajectory. See condition (4), below.

3. If the tangential speed is such that $v_s^2/gr = 1$, the material will leave the belt at the vertical center line through the pulley, as shown in Figure 12.47,

where:

e_t = the point where the material leaves the belt

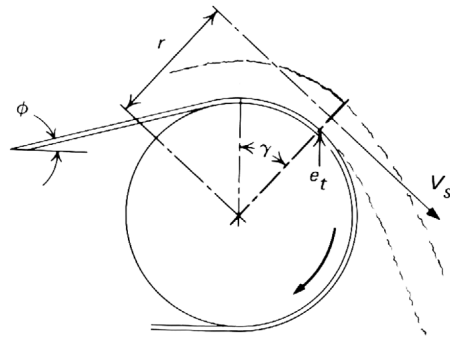


Figure 12.48 When, in an inclined conveyor, the tangential velocity is low (see text), the material will follow part way around the end pulley.

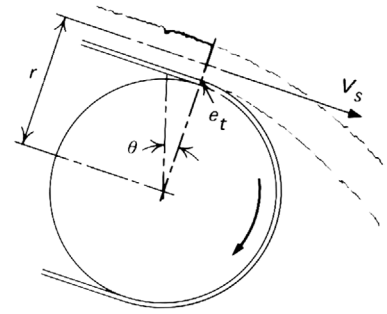


Figure 12.49 When, in a declined conveyor, the tangential velocity is high (see text), the material will leave the belt at the point of tangency of the belt and pulley.

4. If the tangential speed is sufficiently low, or when $v_s^2/gr < \cos \phi$, the material will travel partially around the pulley an angular distance, γ , beyond its top center to the point where $v_s^2/gr < \cos \gamma$. This is shown in Figure 12.48,

Where:

e_t = the point where the material leaves the belt

ϕ = angle, in degrees, of incline of the belt conveyor to the horizontal

γ = angle, in degrees, between the vertical center line, through the pulley to the point, e_t , where the material starts its trajectory

Declined Belt Conveyor Trajectories

If the belt conveyor is declined toward the discharge pulley, there are two conditions to consider:

1. If the tangential speed is sufficiently high, or when $(v_s^2)/(gr) \geq \cos \theta$, the material will leave the belt at the initial point of tangency of the belt and pulley, as shown in Figure 12.49,

Where:

e_t = the point where the material leaves the belt
 ϕ = angle, in degrees, of decline of the belt conveyor

(2) If the tangential speed is insufficient to make the material leave the belt at the initial point of tangency of the belt and pulley, the material will follow partly around the pulley until $(V_s^2)/(gr) \geq \cos \gamma$, as shown in Figure 12.50:

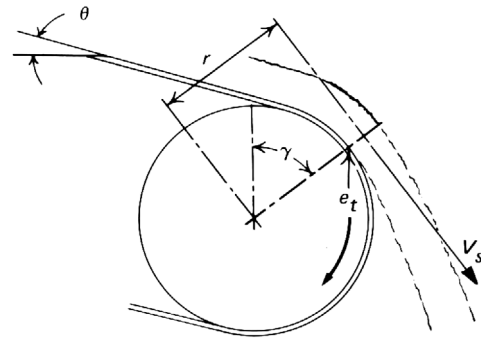


Figure 12.50 When, in a declined conveyor, the tangential velocity is low (see text), the material will follow part way around the end pulley.

Where:

e_t = the point where the material leaves the belt
 γ = angle, in degrees, between the vertical center line through the pulley to point e_t

Plotting the Trajectory

Before the trajectory of the discharged material can be plotted, it is necessary to calculate the values of V_s and r in order to solve the expression:

$$\frac{V_s^2}{gr}$$

It is also necessary to find the height of the flattened load of material on the belt, so that the upper limit of the material path can be plotted.

Referring to Figures 12.39 and 12.40,

If:

a_1 = height in inches above the belt surface of the center of gravity of the cross-section shape of the load, at the point where the pulley is tangent to the belt
 h = height in inches above the belt surface of the top of the load, at the point where the belt is tangent to the pulley
 r = radius in feet from the center of the pulley to the center of gravity of the circular segment load cross section

Then:

$$r = \frac{(a_1) + (\text{belt thickness, inches}) + (\text{pulley radius, inches})}{12}$$

The values of a_l and h have been tabulated for the various belt widths, idler end-roll angles, and surcharge angles, for troughed conveyor belts loaded to the standard edge distance ($0.055b + 0.9$ inch), as listed in Table 12-2.

The tangential velocity, V_s , should be calculated from the relation:

$$V_s = \frac{(2)(\pi)(r)(\text{rpm of end pulley})}{60}$$

V_s should *never* be calculated from the nominal speed of the belt.

Determination of Angular Position of Tangent Line

In order to determine the angular position of the straight line tangent to the circle of radius, r , it is necessary to solve the expression:

$$\frac{V_s^2}{gr} = \cos \phi \text{ or } \cos \theta \text{ or } \cos \gamma$$

Then, from the cosine, determine the angles ϕ or θ or γ in degrees.

Of course, if $\frac{V_s^2}{gr} = 1$ or more, the angle is zero.

See "Horizontal belt conveyor trajectories," pg. 330, "Inclined belt conveyor trajectories," pg. 332, and "Declined belt conveyor trajectories," pg. 333, to apply the angular values in degrees and to find the point, e_t , where the material leaves the belt and starts its trajectory.

To set up the graphical diagram, draw, to some convenient scale, the pulley rim; the belt thickness; the path of the belt, horizontal, inclined, or declined; and a circle, with radius, r , from the center of the pulley. At the determined point, e_t , draw a straight line tangent to the circle of radius r (i.e., perpendicular to the line through the pulley center fixed by the angle ϕ , θ , or γ).

Distance of Material Fall From Tangent Line

The action of gravity will prevent the material from following the tangent line, so the trajectory will lie below it. To measure and plot the amount of fall, it will be convenient to divide the time of fall of the material, resulting from the effect of gravity, into 1/20th seconds. To find the actual distance of fall from the tangent line at successive 1/20-second intervals, consult Table 12-3.

Table 12-3 Fall distance for time intervals.

Time Interval (fractions of seconds)	Fall Distance (inches)	Time Interval (fractions of seconds)	Fall Distance (inches)
1/20th	1/2	12/20ths	69 1/2
2/20ths	1 5/16	13/20ths	81 1/2
3/20ths	4 3/8	14/20ths	94 9/16
4/20ths	7 3/4	15/20ths	108 1/2
5/20ths	12 1/16	16/20ths	123 1/2
6/20ths	17 3/8	17/20ths	139 7/16

Table 12-3 Fall distance for time intervals.

Time Interval (fractions of seconds)	Fall Distance (inches)	Time Interval (fractions of seconds)	Fall Distance (inches)
7/20ths	23 5/8	18/20ths	156 7/16
8/20ths	30 7/8	19/20ths	172 1/2
9/20ths	39 1/16	20/20 (one second)	193 1/4
10/20ths	48 1/4		
11/20ths	58 3/8		

Measurement of the Time Interval

The determination of the interval of time along the tangent line depends upon the calculated tangential velocity, V_s (at radius r).

It will be helpful, in making the layout of the trajectory diagram, to recognize that the distance increments for each 1/20th of a second of time correspond to 0.6 inches for each foot per second of the tangential velocity, V_s .

For example, if the calculated tangential velocity, V_s , is 1 fps, lay out the time intervals on the tangent line from point e_t at 0.6 inch; if the tangential velocity is 2 fps, lay out the intervals at 1.2 inches; if 3 fps, lay out the intervals at 1.8 inches; etc. If the tangential velocity is some fraction of a foot per second, multiply this fraction by 0.6 inch and lay out the intervals accordingly.

1. Start the layout of time intervals on the tangent line from point e_t , the start of the line tangent to the circle of radius r . Number each interval consecutively, 0 for the point of tangency (point e_t), 1 for the first 1/20th second, 2 for the next, and so on.
2. Draw a series of parallel vertical lines downward a suitable distance from each numbered time interval and directly on the tangent line (except the zero number).
3. Lay out on these vertical lines the corresponding distance of fall from the tangent line. To do this, measure vertically downward from each numbered point on the tangent line.
4. Draw a smooth curve through the fall points. This is the median line of the trajectory of the material.

Limits of the Trajectory Path

Having established the median line of the trajectory, lay out a top-of-the-trajectory line with the distance $(h - a_1)$, using Table 12-2. Distance $(h - a_1)$ is the radius of partial circles drawn above and around each fall point. The top limit of the trajectory of normal materials will be a smooth curve drawn tangent to these partial circles. The value of h must be to the same scale as the diagram.

Similarly, the underside limit of the material path should be a smooth curve, tangent to partial circles drawn below and around each fall point. The circles will have a radius equal to the value of a_1 , which also must be to the same scale as the diagram.

See "Discharge trajectories," pg. 326, with respect to long falls below the discharge pulley; light, fluffy materials; or large lumps mixed with fines. Allowance for aberrations in the trajectory limits of such materials must be made when designing discharge chutes. For the individual trajectories of single large lumps, use r as the

distance from the center of the lump to the center of the pulley. Calculate the tangential velocity, V_s , of the lump as follows:

$$V_s = \frac{(2)(\pi)(r)(\text{rpm of end pulley})}{60}$$

The lateral dimension, or width, of the trajectory path of the material will be very close to the length of the circular segment chord x in Figures 12.39 and 12.40. This is approximately $[b - 0.055b - 0.9 \text{ inch}]$ for troughed belts, or $[b - 2(0.055b + 0.9 \text{ inch})]$ for flat belts, where b is the belt width in inches.

The lateral dimension, or width, of the material path is affected by the height of fall and the characteristics of the material; again refer to “Discharge trajectories,” pg. 326.

Examples of Trajectories

The following seven examples of plotted trajectories and trajectory limits of uniform materials discharged over the ends of troughed belt conveyors, which are shown in Figures 12.55 through 12.61, may be helpful. All these examples are based on 30-inch troughed conveyor belts operating horizontally, inclined, or declined, at various speeds (see Figures 12.55 through 12.61), on 20 degree three-equal-roll troughing idlers, and with 24-inch-diameter discharge pulleys.

Also, refer to Figures 12.51, 12.52, 12.53, and 12.54. These are actual photographs taken in the field of conveyor trajectories as materials are discharged over the end pulley of horizontal and inclined troughed belt conveyors.

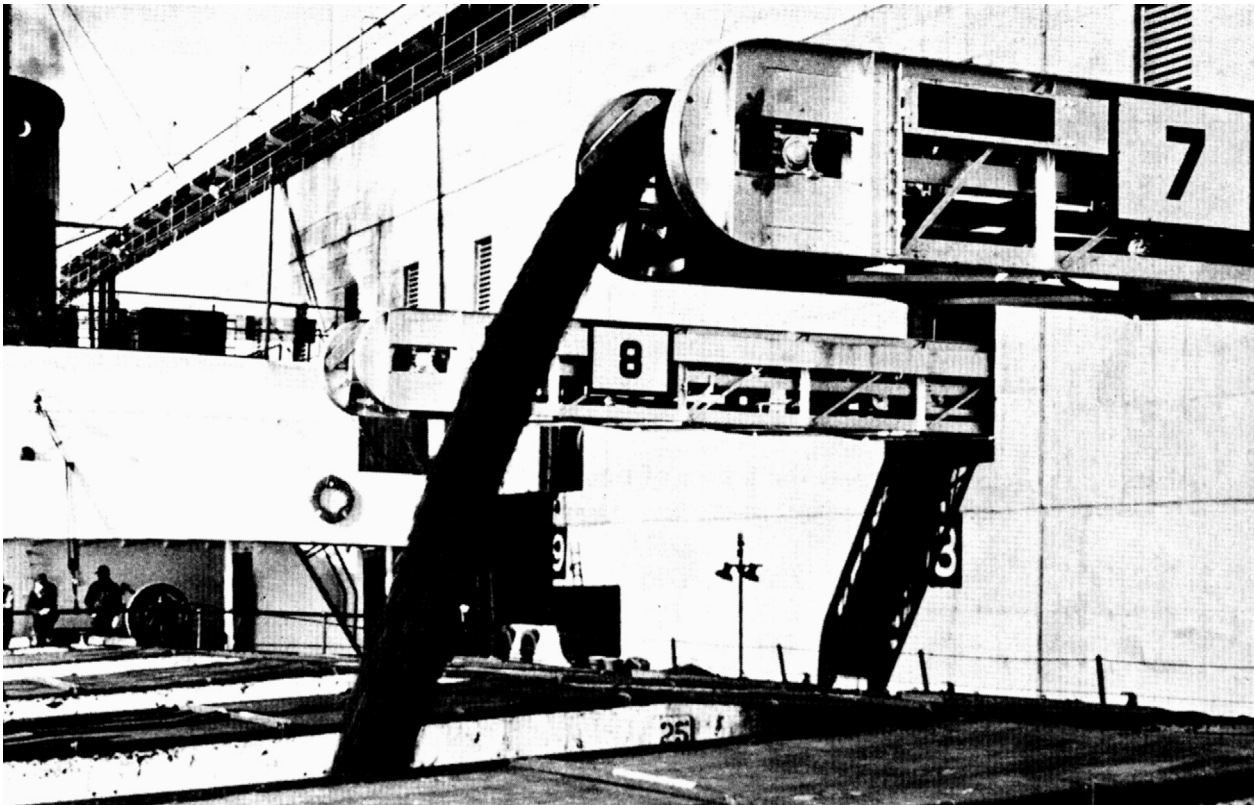


Figure 12.51 Trajectory formed as material is discharged over the end pulley of an horizontal troughed belt conveyor. Here, shuttle belt conveyors project from openings in the face of the dock to load iron ore pellets into the ship's hatches.



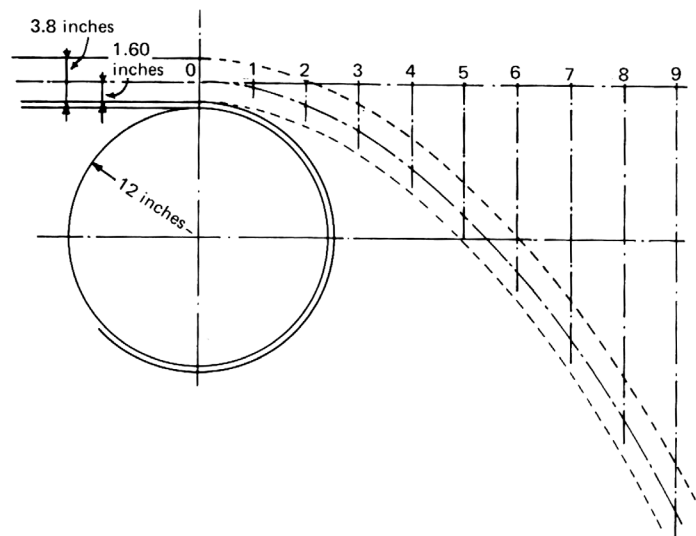
Figure 12.52 Sized aggregate pours over end pulley of this inclined conveyor.



Figure 12.53 Iron-ore pellets discharging from boom belt of Great Lakes self-unloader ship at 10,000 tph.



Figure 12.54 Close-up of trajectory from head pulley of conveyor, speed 760 fpm, handling iron-ore pellets.



Horizontal belt 30 inches wide, 7/16 inches thick, operating at 400 fpm on three-equal-roll 20° standard troughing idlers, loaded to 20° surcharge

Pulley at 61.4 rpm, 1.023 rps

From Table 11-2:

$h = 3.8$ inches

$a_t = 1.60$ inches

$r = 14.038$ inches, or 1.1697 ft

Tangential velocity, $V = (1.023)(6.28)(1.697) = 7.52$ fps

$$\frac{V^2}{gr} = 1.052 \text{ which is greater than } 1$$

See Figure 12.44, Discharge at top of pulley

Velocity spaces on tangential line

$(7.52)(0.6 \text{ inches}) = 4.52$ inches

Fall distances from Table 12-3.

Figure 12.55 Example of discharge trajectory: Drawing No. 1.

Horizontal belt 30 inches wide, 7/16 inches thick, operating at 250 fpm on three-equal-roll 20° standard troughing idlers, loaded to 20° surcharge

Pulley at 38.4 rpm, 0.64 rps
From Table 11-2:

$$\begin{aligned} h &= 3.8 \text{ inches} \\ a_t &= 1.60 \text{ inches} \\ r &= 14.038 \text{ inches, or } 1.1697 \text{ ft} \end{aligned}$$

Tangential velocity, $V = (0.64)(6.28)(1.1697)$
 $= 4.7 \text{ fps}$

$$\frac{V^2}{gr} = 0.586 = \cos \gamma$$

$$\gamma = 54^\circ 7'$$

See Figure 12.45, Discharge beyond top of pulley velocity spaces on tangent line
 $(4.7)(0.6 \text{ inches}) = 2.82 \text{ inches}$
Fall distances from Table 12-3.

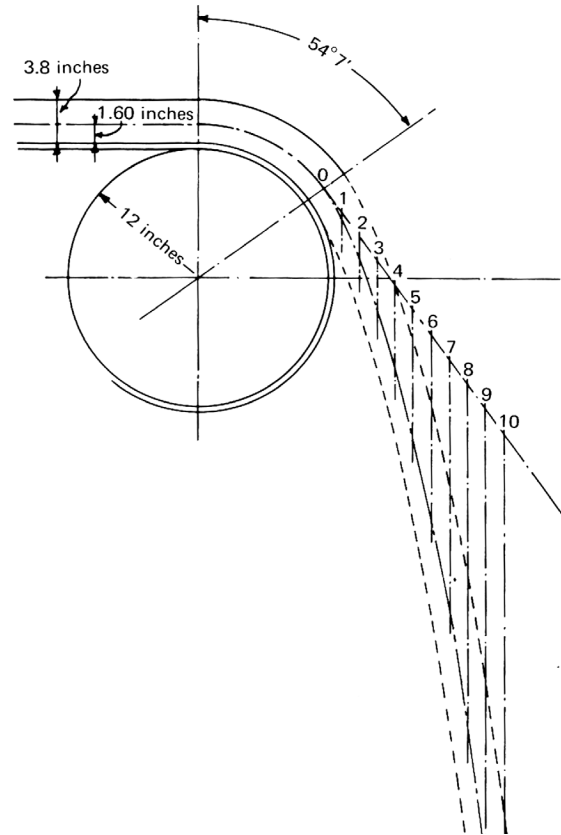


Figure 12.56 Example of discharge trajectory: Drawing No. 2.

Inclined belt 30 inches wide, 7/16 inches thick, operating at 400 fpm on three-equal-roll 20° standard troughing idlers, loaded to 20° surcharge

Pulley at 61.4 rpm, 1.023 rps
From Table 12-2:

$$\begin{aligned} h &= 3.8 \text{ inches} \\ a_t &= 1.60 \text{ inches} \\ r &= 14.038 \text{ inches, or } 1.1697 \text{ ft} \end{aligned}$$

Tangential velocity, $V = (1.023)(6.28)(1.697) = 7.52 \text{ fps}$

$$\frac{V^2}{gr} = 1.052 \text{ which is greater than } 1$$

See Figure 12.46, Discharge at top of pulley
Velocity spaces on tangent line
 $(7.52)(0.6 \text{ inches}) = 4.52 \text{ inches}$
Fall distances from Table 12-3.

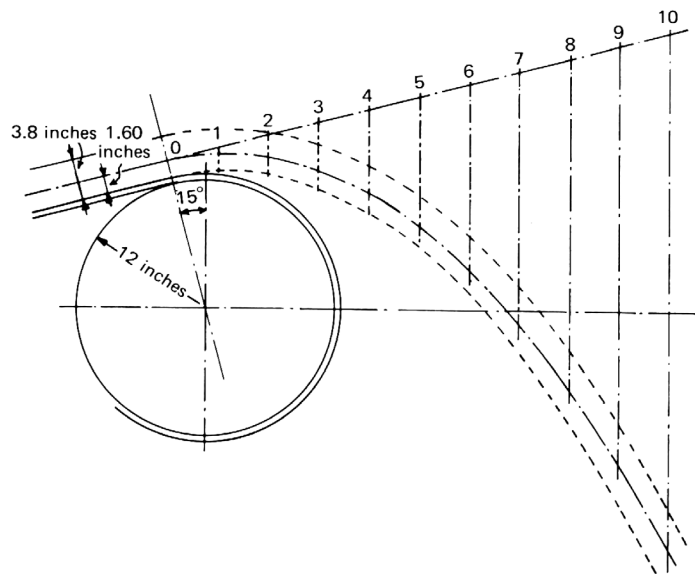


Figure 12.57 Example of discharge trajectory: Drawing No. 3.

Inclined belt 30 inches wide, 7/16 inches thick, operating at 327 fpm on three-equal-roll 20° standard troughing idlers, loaded to 20° surcharge

Pulley at 50.2 rpm, 0.837 rps
From Table 12-2:

$$\begin{aligned} h &= 3.8 \text{ inches} \\ a_t &= 1.60 \text{ inches} \\ r &= 14.038 \text{ inches, or } 1.1697 \text{ ft} \end{aligned}$$

Tangential velocity, $V = (0.837)(6.28)(1.697) = 6.15 \text{ fps}$

$$\frac{V^2}{gr} = 1$$

See Figure 12.47, Discharge at top of pulley
Velocity spaces on tangent line
 $(6.15)(0.6 \text{ inches}) = 3.690 \text{ inches}$
Fall distances from Table 12-3.

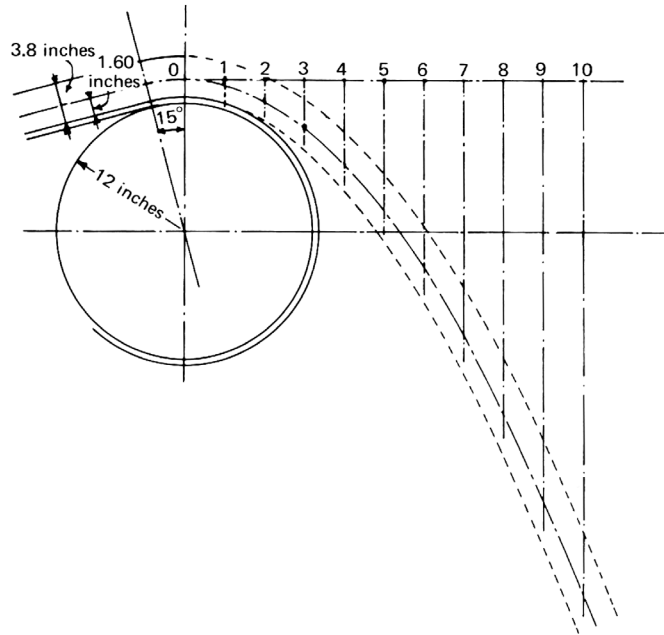


Figure 12.58 Example of discharge trajectory: Drawing No. 4.

Inclined belt 30 inches wide, 7/16 inches thick, operating at 280 fpm on three-equal-roll 20° standard troughing idlers, loaded to 20° surcharge

Pulley at 43 rpm, 0.717 rps
From Table 12-2:

$$\begin{aligned} h &= 3.8 \text{ inches} \\ a_t &= 1.60 \text{ inches} \\ r &= 14.038 \text{ inches, or } 1.1697 \text{ ft} \end{aligned}$$

Tangential velocity, $V = (0.717)(6.28)(1.697) = 5.27 \text{ fps}$

$$\frac{V^2}{gr} = 0.736 = \cos \gamma = 42^\circ 36'$$

See Figure 12.48, Discharge beyond top of pulley
Velocity spaces on tangent line
 $(5.27)(0.6 \text{ inches}) = 3.162 \text{ inches}$
Fall distances from Table 12-3.

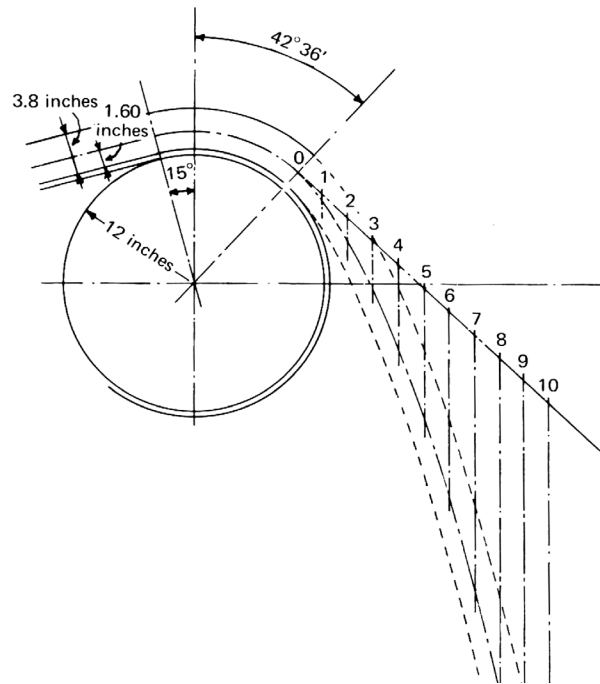


Figure 12.59 Example of discharge trajectory: Drawing No. 5.

Declined belt, 30 inches wide, 7/16 inches thick,
operating at 400 fpm on three-equal-roll 20°
standard troughing idlers, loaded to 20° sur-
charge

Pulley at 61.4 rpm, 1.023 rps

From Table 12-2:

$$h = 3.8 \text{ inches}$$

$$a_t = 1.60 \text{ inches}$$

$$r = 14.038 \text{ inches, or } 1.1697 \text{ ft}$$

Tangential velocity, $V = (1.023)(6.28)(1.697) = 7.52 \text{ fps}$

$$\frac{V^2}{gr} = 1.502, \text{ which is greater than } 1$$

See Figure 12.49, Discharge at tangency of belt
with pulley

Velocity spaces on tangent line

$$(7.52)(0.6 \text{ inches}) = 4.52 \text{ inches}$$

Fall distances from Table 12-3.

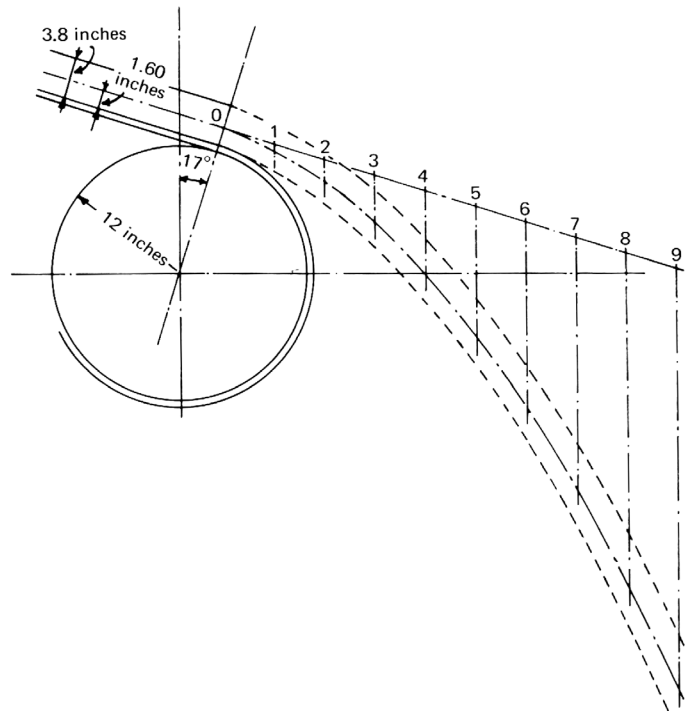


Figure 12.60 Example of discharge trajectory: Drawing No. 6.

Discharge Trajectories

Declined belt, 30 inches wide, 7/16 inches thick,
operating at 280 fpm on three-equal-roll 20° stan-
dard troughing idlers, loaded to 20° surcharge

Pulley at 43 rpm, 0.717 rps

From Table 12-2:

$$h = 3.8 \text{ inches}$$

$$a_t = 1.60 \text{ inches}$$

$$r = 14.038 \text{ inches, or } 1.1697 \text{ ft}$$

$$\text{Tangential velocity, } V = (0.717)(6.28)$$

$$(1.697) = 5.27 \text{ fps}$$

$$\frac{V^2}{gr} = 0.736 = \cos \gamma, \gamma = 42^\circ 36'$$

See Figure 12.50, Discharge beyond
tangency belt with pulley

Velocity spaces on tangent line

$$(5.27)(0.6 \text{ inches}) = 3.162 \text{ inches}$$

Fall distances from Table 12-3.

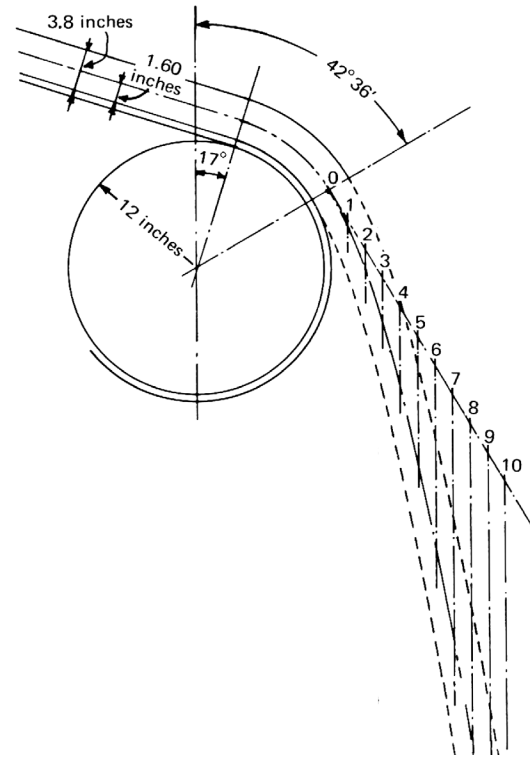


Figure 12.61 Example of discharge trajectory: Drawing No. 7.

Conveyor Motor Drives and Controls

Introduction
Conveyor Drive Evaluation Criteria
Belt Drive Attributes
Conveyor Drive Systems Overview
Other Conveyor Drives
Conveyor Belt Controls
Conclusion

Introduction

Belt conveyors are an important system for transporting bulk materials in industry. The control of the application of the starting torque from the belt drive system to the belt fabric affects the performance, life cost, and reliability of the conveyor. The force required to move a belt conveyor must be transmitted by the drive pulley via friction between the drive pulley and the belt fabric. To transmit power there must be a difference in the belt tensions as the belt fabric approaches and leaves the drive pulley. These conditions are true for steady state running, starting, and stopping.

The belt designer must control starting acceleration to prevent excessive tension in the belt fabric and forces in the belt drive system. High acceleration forces can adversely affect the belt fabric, belt splices, drive pulleys, idler pulleys, shafts, bearings, speed reducers, and couplings. Uncontrolled acceleration forces can cause belt conveyor system performance problems with vertical curves, excessive belt take-up movement, loss of drive pulley friction, spillage of materials, and festooning of the belt fabric. The belt drive system must produce an accelerating torque powerful enough to start the conveyor and controlled in such a manner that the acceleration forces are within safe limits. Smooth starting of the conveyor can be accomplished with drive starting torque control equipment, either mechanical or electrical, or a combination of the two. In some cases, it is also desirable to control the running and stopping forces of the conveyor belt system.

Conveyor Belt Evaluation Criteria

What is the best belt conveyor drive system? This is a common question for the belt designer. The answer depends on many variables. The best system is one that provides acceptable control for starting, running, and stopping at a reasonable cost and with high reliability.

Belt Drive System

For the purposes of this discussion, we will assume that belt conveyors are almost always driven by electrical prime movers. The belt “drive system” consists of multiple components including the electrical prime mover, the electrical motor starter/controller with control/regulator system, the motor coupling, the speed reducer, the low speed coupling, the belt drive pulley, the pulley brake or hold back, and the belt control system. It is important that the belt designer examine the applicability of each system component to the particular application.

Electric Motor Attributes

A conveyor system designer must specify an electrical prime mover with attributes consistent with the conveyor application. The motor attributes include electrical and mechanical specifications.

Type

The National Electrical Manufacturers Association (NEMA) defines various types of electrical motors. For conveyors, alternating current (AC) squirrel cage induction or wound rotor (WRIM), and direct current (DC) shunt or compound wound rotating enclosed machines are most common. The motor nameplate defines the motor type and rating constants.

Temperature

Electric motors are rated for nameplate full load temperature rise above an ambient for altitudes below 3,300 feet above sea level based on classes of insulation. Unless stated otherwise, ambients are assumed to be 40°C. Higher altitudes and ambients may require motor derating. Class A insulation materials are rated for 60°C temperature rise above ambient, Class B for 80°C, Class F for 105°C, and Class H for 125°C. These temperature limits are average, and assume up to 10°C hot spot rise above this average. NEMA standards recommend that large motors above 1,500 horsepower include internal stator temperature detectors. Running motors above rated insulation temperature does not cause immediate failure, but reduces insulation life and the ability of insulation to withstand moisture. It is generally assumed that for a motor steady state load running 10°C above the insulation rating, the motor insulation life is reduced by half. Vacuum pressure impregnated (VPI) insulation systems use epoxy fills to seal motor windings and insulation from moisture penetration. However, VPI insulations degrade the heat convection rate from the windings to the ambient.

Enclosures

Electric motors are available in various enclosures. NEMA publishes common descriptive terms for enclosure types. Different motor suppliers often offer variations or adaptations on motor enclosure types. Motor enclosure types are commonly referenced for AC induction motors. Wound rotor and DC motors may be available in more restrictive enclosure designations. Most motor enclosure types are designed for cooling by ambient air. Some motors enclosures are available for liquid or gas cooling.

Open motors are uncommon on conveyors. Open drip proof (ODP) motors provide environmental protection from falling water and particles approaching the motor within 15° of vertical while allowing ambient air to ventilate the motor electrical windings. Adaptations of ODP motors are weather protected (WPI or WPPI), splash proof, fully guarded, and separately force ventilated designs that add tortuous ventilation paths via baffles, insect screens, moisture resistant form wound sealed insulation, and/or dust filters. ODP motors offer economy and optimum heat dissipation. Totally enclosed fan cooled (TEFC) designs offer a sealed motor enclosure with no free exchange of ambient air between the inside and outside of the motor enclosure. Heat is dissipated through the air over the motor enclosure by integral motor fans, air to air heat exchangers, forced air cooling by separate blowers, water jackets, or gas filled enclosures. Adaptations of TEFC are totally enclosed non-ventilated (TENV), totally enclosed water-to-air cooled (TEWAC), totally enclosed air-to-air cooled (TEAAC), and totally enclosed air over (TEAO). Since TEFC designs involve a heat exchanger, motor frames are larger than ODP designs for the same horsepower. TEFC motors are commonly used with conveyors that must function in adverse environments.

Hazardous rated enclosures are used in conveyor applications in National Electrical Code Article 500 Hazardous Areas. NEC defines hazardous areas by class and division according to the type and occurrence of hazard. Class I hazards refer to combustible gases. Class II hazards refer to combustible dusts. Class III hazards refer to combustible fibers. Explosion proof or permissible enclosures are used in underground mining environments subject to regulation by federal and state mining regulations. International Electrical Commission (IEC) enclosure designations specify motor enclosures for worldwide applications. An explosion-proof machine is designed to contain an internal ignition of a specified gas, vapor, or dust and prevent the propagation of the explosion outside the enclosure.

Speed

The AC induction motor utilizes current applied to stator windings arranged in pole groups which induces power into the rotor windings. The rotor windings are arranged in pole groups and shorted at the end by conducting end rings. When the stator is supplied by a three-phase source, it produces a magnetic field that rotates near to a synchronous speed as determined by the number of poles and the applied frequency. For a 60Hz source, the common synchronous speeds available for motors are 3,600 rpm for two pole, 1,800 rpm for 4 pole, 1,200 rpm for 6 pole, 900 rpm for 8 pole, and so on. For belts, the 1,800 rpm and 1,200 rpm speeds are most common. The rotor then rotates at the synchronous rpm for no load and slows to the full load rpm at full load torque. The lagging in speed below the rotating magnetic field produces a mechanical torque called slip.

DC shunt motors are available in standard speeds similar to AC motors. AC synchronous speeds correspond with standard DC speeds of 3,500, 1,750, 1,150, and 850 rpm. At increasing horsepower, wound armature designs limit DC motor speeds. For example, depending on the manufacturer, at above 800 horsepower, 1,750 rpm motors may not be available. The base speed of DC motors is controlled by the shunt field current. The base speed can be lowered by increasing the field current and raised by decreasing the field current, within limits. The motor base speed is reached at full armature voltage.

$$HP = \frac{SPEED_{rpm} \times TORQUE_{FT-LB}}{5252}$$

Horsepower remains the product of torque and speed. For a given horsepower, a higher speed motor operates at lower torque in a smaller frame.

Voltage

For three-phase AC motors, mechanical power is the product of $\sqrt{3}$, voltage, current, power factor, and efficiency. For DC shunt motors, power is the product of voltage, current, and efficiency. For a required horsepower, a higher motor voltage results in a lower motor current. In the United States, NEMA defines common utilization voltages for motors. Common conveyor AC motor application voltages are 230, 460, 575, 950, 2,300, 4,160, and 6,600 vac. Common DC motor application voltages are 250, 500, 600, 700, and 750 vdc.

Bearings

Conveyor rotating motors are available in ball, roller, and sleeve bearings. Bearing lubrication is by grease, oil film, or oil bath. For horizontal motors, the most common bearings are standard deep groove ball bearings with grease lubrication. For severe environments, single or double shielded bearings protect the lubrication grease from contamination. Belt, chain drives, and overhung coupling loads exert additional radial forces on motor bearings. Sleeve bearings constructed of tin-based babbitted liners bonded to bronze or cast iron shells with oil film lubrication are more common on very large motors. The choice and availability of bearing and lubrication types depend on speed, size, and local practices.

Service Factor

In AC motors, service factors (S.F.) of 1.0 and 1.15 are commonly available. A 1.15 S.F. motor provides the same torque versus speed characteristics of a 1.0 S.F. motor, but allows for a 15 percent thermal overload capability regarding insulation and winding temperature. On belt conveyors, designers often choose a 1.15 S.F.

motor with the horsepower chosen for standard CEMA belt loading and the 15 percent overload capability for periodic surge loading. In DC motors, the standard S.F. is 1.0.

Efficiency

Electric motor efficiency is a measure of the ability to convert electrical input power to mechanical output power. Nameplate efficiency is listed as a percentage based on motor full load. Motors are available in standard, high, and premium efficiency models. Motor efficiency normally increases with larger motors and decreases with partial motor load. Motors are made more efficient by using more conductor material such as copper in the stator, by using thinner laminations of better grade magnetic steel in the poles and core, by increasing the copper conductor content of the rotor, by using lower friction bearings, by using more efficient cooling fans, by using larger frames to reduce operating temperature, and by reducing magnetic excitation levels to decrease core losses. The National Energy Policy Act of 1992 dictates a minimum AC motor efficiency for standard T-frame motors based on size and speed.

Torque Type

NEMA classifies low voltage AC induction motors through 200 hp into several classes with respect to locked rotor,

pull-up, breakdown torques, and running slip (see Figure 13.1). In general, NEMA Design A motors have low to moderate locked rotor torque, low pull-up torque, high breakdown torque, high starting currents, and small running slip. NEMA Design B motors have moderate locked rotor torque, moderate pull-up torque, medium to high breakdown torque, moderate starting current, and moderate running slip. NEMA Design C motors have moderate to high locked rotor torque, high pull-up torque, medium breakdown torque, medium to low starting current, and medium running slip. NEMA Design D motors have high locked rotor torque, high pull-up torque, moderate to low breakdown torque, low starting current, and high running slip. NEMA Design B and Design C motors are most commonly applied to belt conveyors. NEMA Design C motors are usually chosen for direct coupled loads.

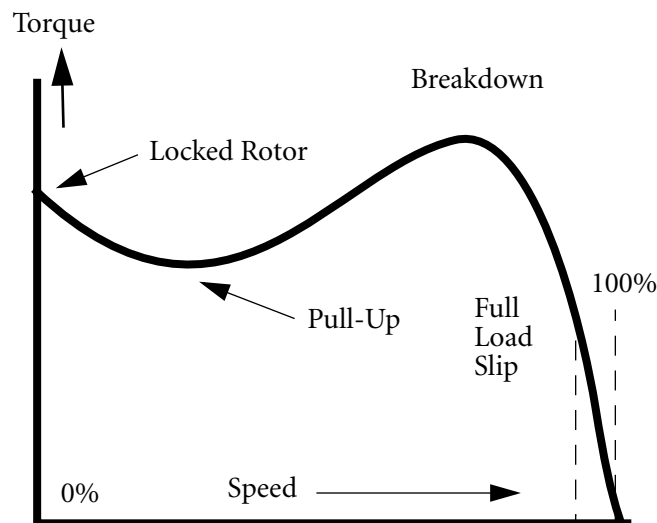


Figure 13.1 Typical AC motor torque versus speed curve.

<i>Motor Protection Devices</i>	Motors may be provided with the following optional protection features. The selection of these features depends on motor size, cost, and usage levels.
<i>Winding Temperature</i>	The motor is protected from thermal overload by an overload meeting NEC requirements and the motor ventilation system. The motor overload model may not match the installed motor characteristics or environment. The motor ventilation system is dependent on sufficient air volume and surface conditions. To augment motor protection, the user should consider motor winding temperature protection. Classically, this has involved the insertion of winding thermostats or “Klixons” into the stationary windings that operate at a preset temperature. They can trigger an alarm or shutdown. Alternatively, motor measurements of temperature can be measured by resistance temperature devices (RTDs), thermocouples, or thermistors. In conveyor environments, heavy duty 100 ohm platinum RTDs are the most common choice for temperature measurement. A 100 ohm platinum sensor measures 100 ohms at 0°C, and rises .00385 ohms per degree (European DIN standard, .003916 ohms American Standard). This resistance is monitored by a RTD transducer in the belt control system. The motor may be alarmed at high temperature and shut down at high/high temperature. The motor temperature may limit belt loading. RTD sensor wiring should be brought outside the motor enclosure in a junction box separate from the power conductors.
<i>Bearing Temperature</i>	The temperature of the bearing can reflect on the health of the lubrication system and the loading of the bearing. By embedding a RTD sensor within the bearing housing, the belt control system can monitor the bearing temperature. The absolute bearing temperature is affected by the motor ambient. A good strategy is to subtract the ambient temperature from the absolute to monitor the rise above ambient. A change in this level suggests a bearing problem. Again, 100 ohm platinum RTDs are the best choice. In bearings, some thought needs to be given to the RTD sensor choice. One strategy is to use spring-loaded, tip-sensitive RTDs inserted at 90 degrees to the shaft that contacts the outer race of the bearing. Another is to use an averaging sensor that measures the average from the tip contact with the bearing through the grease region to the motor bearing frame. RTDs can also be mounted in stainless steel wells that isolate the sensor from the grease environment, and allow for insertion and removal when running. In any event, it is preferable to have the manufacturer of the motor mount the RTD in the bearing housing, as opposed to field mounting.
<i>Vibration</i>	The vibration level of the motor is an indicator of the bearing status, motor alignment to load, or system vibrations conveyed to the motor by structure. System vibrations can be generated by large pieces of conveyed material traveling through discharge chutes at belt transfers. Vibration, by itself, is difficult to interpret. Vibration sensors can measure displacement, velocity, or acceleration. Vibration analyzers determine the vibration at each frequency. Most motor vibration sensors are commonly broadband frequency devices that add all vibration frequencies with equal weight and no filtering. This can result in different absolute vibrations for different motors in different locations. Again, the belt control system is configured to monitor the change in level of the vibration. Vibration control often generates alarms, but does not shut down the motor on belt conveyors. A vibration alarm is normally followed up by a visual inspection, and/or a vibration survey with an analyzer instrument.

Belt Drive Attributes

Motor Space Heater

In motors mounted outdoors or subjected to condensing environments, there is concern for the accumulation of water within the motor windings and enclosures. This would normally occur when the motor was not running and had cooled through the temperature of condensation. Motors can be specified with auxiliary enclosure heaters normally powered from the belt control system at 120 vac. The heater is configured to be on only after the motor is idled for some time and the motor temperature passes below a setpoint. The heater should not be allowed to run when the motor is idled and the motor is already hot, as the heater will impede cooling for the next operation duty cycle. The cost of the motor heater and its associated control must be balanced with the user's proposed environment.

Motor Tachometer

Some conveyor applications require a measurement of the prime mover speed. A motor driven tachometer provides a linear signal proportional to speed. Motors can be supplied with mounting provisions for tachometers. Analog tachometers are traditional DC generators with brushes and permanent magnet fields. Analog tachs generate a DC voltage proportional to speed at some linear rate, such as 50 volts per 1000 rpm. Digital tachs are often non-contact designs with optical or inductive coupling that generate pulses per revolution. A digital tach usually requires a companion digital to analog converter board. In the design of the D/A board, a trade-off occurs between the minimum pulse rate and the signal update time. Thus a digital tach cannot measure to zero speed. The user should obtain a tachometer of an accuracy and robustness suitable to the application.

Belt Drive Attributes

What are the requirements of a belt conveyor system? The belt designer must catalog the belt conveyor requirements in order to select and match the drive system features.

Size

Certain drive components are available and practical in different size ranges. For this discussion, we will assume that belt drive systems range from fractional horsepower to multiples of thousands of horsepower. Small drive systems are often below 50 horsepower. Medium systems range from 50 to 1,000 horsepower. Large systems can be considered above 1,000 horsepower. Division of sizes into these groups is entirely arbitrary. One should resist the temptation to overmotor or undermotor a conveyor to enhance standardization. An overmotored drive results in poor efficiency and the potential for high torques, while an undermotored drive could result in destructive overspeeding on regeneration, the failure to start a load, or overheating on loading with shortened motor life.

Torque Control

Belt designers try to limit the starting torque to no more than 150 percent of the running torque. The limit on the applied starting torque is often the limit of rating of the belt carcass, belt splice, pulley design, or shaft deflections. On larger belts and belts with optimized sized components, torque limits of 110 percent through 125 percent are common. Besides a torque limit, the belt starter may be required to limit torque increments that would stretch belting and initiate traveling waves. An ideal starting control system would first apply a pretension torque to the belt at rest, up to the point of breakaway, or movement of the entire belt; and then a torque equal to the

movement requirements of the belt with load, plus a constant torque to accelerate the inertia of the system components from rest to final running speed. This would minimize system transient forces and belt stretch. Different drive systems exhibit varying ability to control the application of torques to the belt at rest and at different speeds. Also, the conveyor itself exhibits two extremes of loading. An empty belt normally presents the smallest required torque for breakaway and acceleration, while a fully loaded belt presents the highest required torque. A conveyor drive system must be able to scale the applied torque from a 2:1 ratio for a horizontal simple belt arrangement, to a 10:1 range for an inclined or complex belt profile. Classically, stopping the conveyor involves the removal of motive driving force, while allowing the conveyor to drift to a stop. Some complex conveyor profiles or dynamic traveling wave situations require drive torque control during stopping. The belt drive torque is reduced at a controlled rate as the conveyor is driven to stop over a period longer than the normal drift time.

Thermal Rating

During starting, running, and stopping, each drive system dissipates varying quantities of waste heat. The waste heat may be dissipated in the electrical motor, the electrical controls, the couplings, the speed reducer, or the belt braking system. The thermal load of each start is dependent on the amount of belt load and the duration of the start. The designer must fulfill the application requirements for repeated starts after running the conveyor at full load. Typical conveyor belt starting duties vary from 3 to 10 starts per hour equally spaced, or 2 to 4 starts in succession. Repeated starting may require the derating or oversizing of system components. There is a direct relationship between thermal rating for repeated starts and costs for each drive system.

Variable Speed

Some belt drive systems are suitable for controlling the starting torque and speed, but only run at constant speed. Some belt applications require a drive system that can run for extended periods at less than full speed. This is useful when the drive load must be shared with other drives, when the belt is used as a process feeder for rate control of the conveyed material, when the belt speed is optimized for the haulage rate, when the belt is used at slower speeds to transport supplies, or when the belt is run at slow inspection or inching speed for maintenance purposes. The variable speed belt drive will require a control system based on some algorithm to regulate operating speed.

Regeneration or Overhauling Load

Some belt profiles present the potential for overhauling loads where the belt system supplies energy to the drive system (see Figure 13.2). Not all drive systems can accept regenerated energy from the load. Some drives can accept energy from the load and return it to the power line for use by other loads. Other drives accept energy from the load and dissipate it into designated dynamic or mechanical braking elements. Some belt profiles switch from motoring to regeneration during operation. Issues to address include: Can the drive system accept regenerated energy of a certain magnitude for the application? Does the drive system have to control or modulate the amount of retarding force during overhauling? Does the overhauling occur when running, starting, and/or stopping?

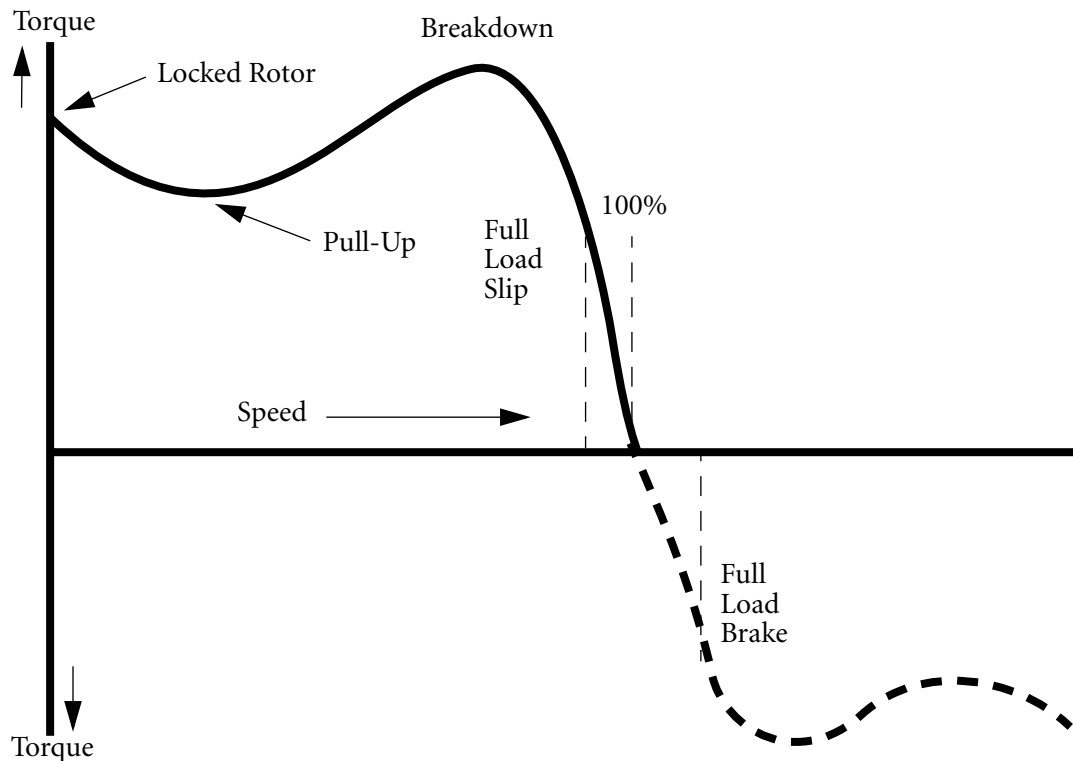


Figure 13.2 AC induction motor motoring and regeneration.

Maintenance and Supporting System

Each drive system will require periodic preventative maintenance. Replaceable items include motor brushes, bearings, brake pads, dissipation resistors, oils, and cooling water. If the drive system is conservatively engineered and operated, the lower stress on consumables will result in lower maintenance costs. Some drives require supporting systems such as circulating oil for lubrication, cooling air or water, environmental dust filtering, or computer instrumentation. The maintenance of the supporting systems can affect the reliability of the drive system.

Cost

The drive designer will examine the cost of each drive system. The total cost is the sum of the first capital cost to acquire the drive, the cost to install and commission the drive, the cost to operate the drive, and the cost to maintain the drive. The cost of power to operate the drive may vary widely with different locations. The designer strives to meet all system performance requirements at lowest total cost. Often more than one drive system may satisfy all system performance criteria at competitive costs.

Complexity

The preferred drive arrangement is the simplest, such as a single motor driving through a single head pulley. However, mechanical, economic, and functional requirements often demand the use of complex drives. The belt designer must balance the need for sophistication against the problems that accompany complex systems. Complex systems require additional design engineering for successful deployment. An often overlooked cost in a complex system is the cost of training onsite personnel, or the cost of downtime because of insufficient training.

Multiple Drives

A simple belt is often driven by one drive pulley. More complex belts are driven by multiple, independent pulleys. These pulleys may be near each other or at different belt locations. Multiple drives allow the belt designer to increase the driven horsepower while maintaining or reducing belt fabric tensions. Multiple drives require a drive starting and running system that allows for coordination between drives. Multiple drives of different sizes or different belt wrap angles may require a load proportion scheme (load sharing). Load sharing requires one of the drives to operate at a controlled, dependent torque rating during starting, or also during running conditions. Of concern is the question: Will the multiple drive belt systems operate with one or more prime movers out of service? Multiple drives at different locations will require a distributed control system.

Conveyor Jam

It is possible for a running belt conveyor to encounter a mechanical jam of the belt fabric. The drive system will continue to impart torque to the fabric up to the slip of the fabric on the drive pulley. Different drives vary in the application of breakdown torque to the stalled machine. It is important for the drive designer to examine the rotating inertia in the drive system. The prime mover motor rotor and all other rotating parts may contribute significant kinetic energy to the stalled belt. The drive response to the stall and the application of the torque limit may vary.

Belt Control System

Each drive system will require a base case permissive control system for starting and running supervision. Most belt drive systems today use some form of computer control. Computer control systems all depend on field sensors for measuring and reporting drive parameters. The belt drive designer must decide the minimum required number and location of the field devices for adequate control. The drive control system will require power switchgear and control switchgear with provisions for “lockout” for conveyor maintenance and service. Lockout applies to sources of electrical, hydraulic, pneumatic, and gravity energy.

Conveyor Drive Systems Overview

We will next examine the electrical drive and starting mechanism for the conveyor drive system. The system approach will group the electrical prime mover with control, the starting mechanism, the high and low speed couplings, the speed reducer, and the drive pulley for examination. The belt conveyor will require a force to initiate movement, termed “breakaway torque.” In movement, the conveyor will require torque to support both existing motion and to support acceleration to running speed within the time and thermal limits imposed by the system. The resultant acceleration is proportional to the applied force divided by the mass in the system according to Newton’s second law, where the mass is the system equivalent weight, or external inertia, divided by the force of gravity. More commonly, the belt designer will calculate the WK^2 , or the weight of the system times the square of the radius of gyration. It is important that the designer add the drive system external inertia to the belt system inertia to view total system inertia. System inertias are modeled at the rotating speed of the electrical prime mover, or at the linear speed of the belt fabric. All system external inertias must be referred to a common per unit base speed by a function of the square of the speed ratios.

AC Induction Motor with Full Voltage Starting and Direct Couplings

The most common electric motor for driving belt conveyors is the alternating current squirrel cage induction motor. The relationship of the motor torque to motor speed is nearly linear between the no load and full load operating points. Since full load slip is often less than 1-2 percent, the AC induction motor when running can often be considered a constant speed device. However, during starting we must examine other motor characteristics. In this area of motor operation, the AC induction motor torque is highly non-linear, as shown in Figure 13.3.

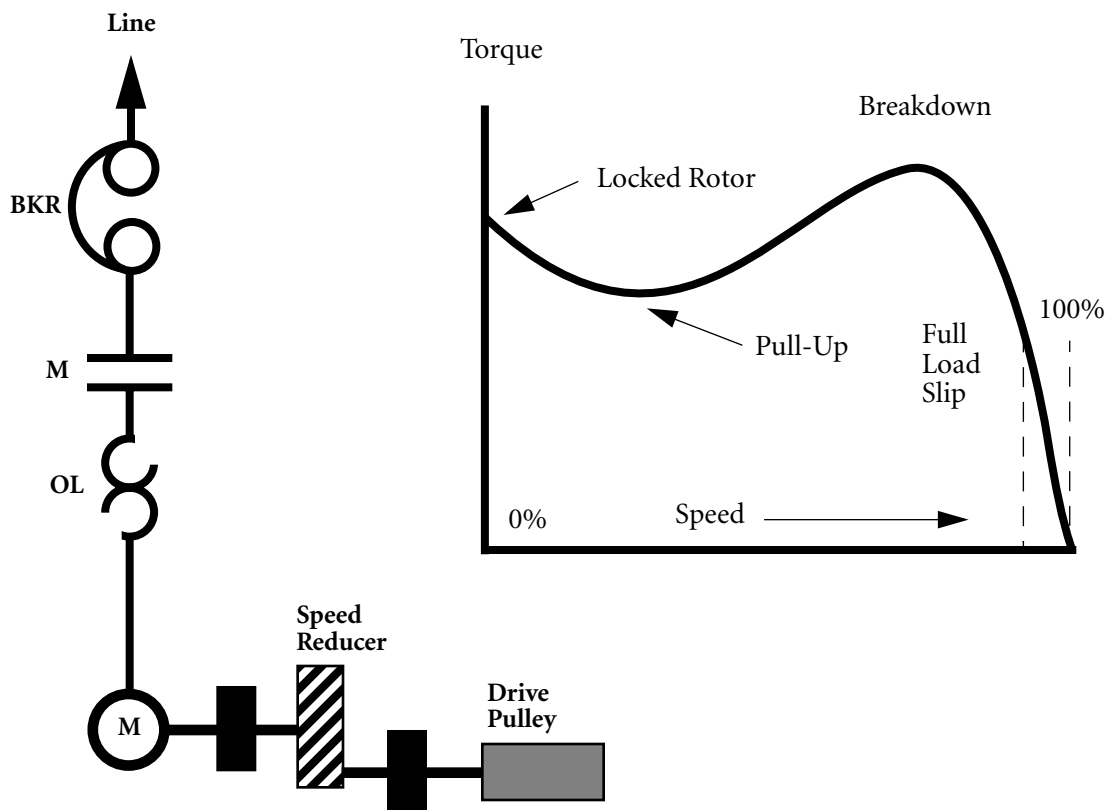


Figure 13.3 AC motor with full voltage starting and direct coupling.

At the point of full voltage application of current to the stator with the rotor at standstill, the motor is modeled at “locked rotor torque.” “Pull-up torque” is the minimum torque developed by the motor when the rotor is moving and accelerating the load toward the synchronous speed. The maximum torque attained during acceleration is termed “breakdown torque.” The motor torque at any speed varies with the stator resistance and inductance, the rotor resistance and inductance referred to the stator as a function of slip, and the applied voltage squared.

$$Time_{Accel} = \frac{WK^2 \times RPM}{308 \times T_{Accel}}$$

Where:

$Time$ = Accel Time in Seconds

T = Average Accel Torque_{lb-ft}

T_{accel} = Prime Mover Torque - Required Load Torque

Although AC induction motors represent the simplest and most economical means of driving belt conveyors, the standard NEMA motor types do not have ideal characteristics for starting control of the conveyor. AC induction motors are available from fractional horsepower through thousands of horsepower. The direct-coupled, full-voltage starting application is usually limited to small conveyors at moderate speeds and profiles due to the impact of the locked rotor, pull-up, and breakdown torque to the conveyor system during starting. This system offers no controlled torque application.

The thermal rating of the starting system is equal to the rating of the motor. Since the motor experiences high currents during starting and acceleration, the number of starts is limited as the motor size is increased. The belt operates as a constant speed device between no load and full load according to motor slip. The drive can handle belt regeneration of power up to the motor rating for extended periods. The induction motor crosses into the generator braking quadrant of operation at synchronous speed plus slip. The faster the motor is driven, the more power is generated. The power is absorbed on the power line by other motor loads.

This drive requires almost no maintenance or supporting systems. The cost and complexity of this drive type are very low. In many ways, this method represents the base case for simple small conveyors. This drive type can be configured for multiple belt drives at one or more locations. The drives must start together or in close succession from the leader to the follower on the drive side of the fabric. The drives will load share according to the slip torque versus speed curves for each motor. Upon conveyor jam, this drive system will apply full breakdown torque to belt components before stall. This torque may be more than 200 percent of conveyor rating. Jam may be sensed by measuring high motor currents, but it is important to note that currently, the high torque has been applied. The control system for this drive type consists of the base case of conveyor protective devices that simply tell the belt to move or stop.

AC Induction Motor with Reduced Voltage Starting and Direct Coupling

Using the previous base case of the AC induction motor with direct coupling, and adding an electrical starter that can apply reduced voltage to the motor stator, we have the most common type of drive system for belt conveyors. Since the AC induction motor torque at any speed is a function of the square of the applied terminal voltage, if we control the voltage, we can control the torque during starting. There are various forms of reduced voltage starters such as autotransformer, capacitor, reactor, resistor, or silicon controlled rectifier (SCR) type. The autotransformer, capacitor, reactor, and resistor type starters use switching to create step increments of applied terminal voltage, while the SCR type starters allow for continuous control of the applied root mean squared voltage. The SCR type of starter, which became economically practical in the 1970s, dominates the reduced voltage starter classification for conveyors today. The SCR starter is constructed of solid state materials with no mov-

ing or wearing parts. With conservative design in the sizing of components, the SCR starter is very reliable.

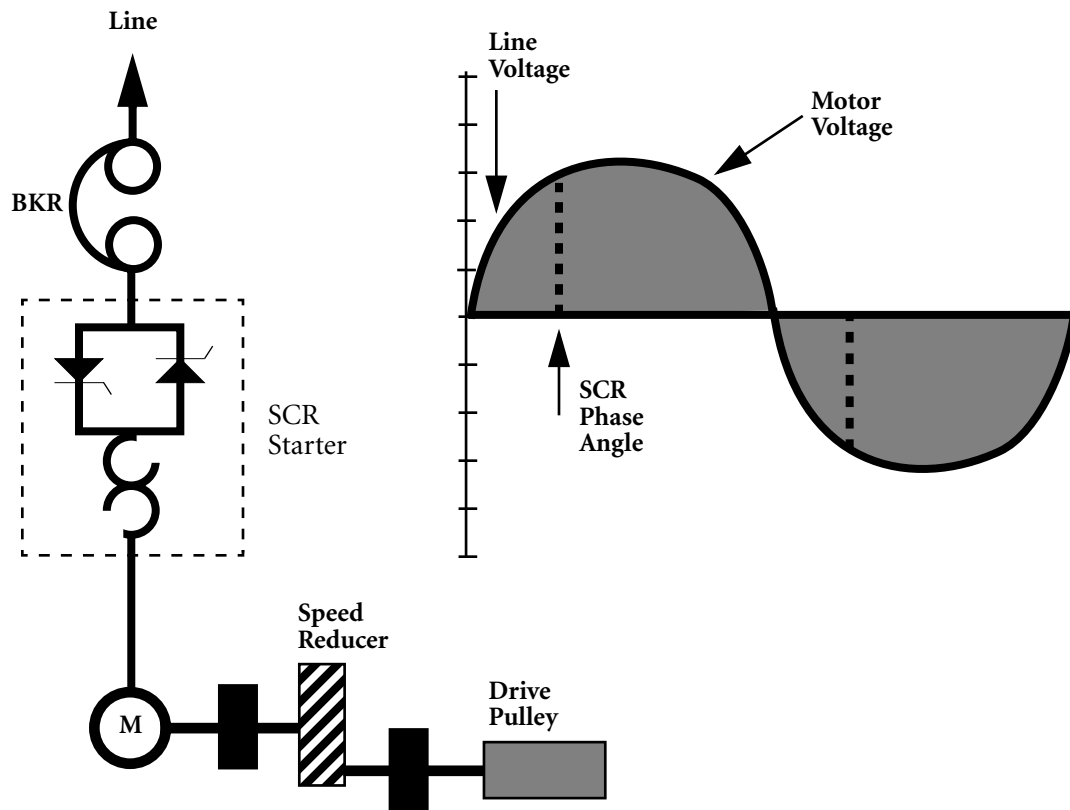


Figure 13.4 SC motor with reduced voltage starting and direct coupling.

SCR starters are commonly available from fractional horsepower to 1000 horsepower in voltages up to 1000 vac. For larger drives, there are some new, first generation, high voltage SCR starter designs for application at 2300 and 4160 vac. To provide good starting torque at locked rotor, the NEMA Design C type motor is usually matched with the SCR starter. However, since the motor torque versus speed profile is not linear, the starter must apply a non-linear progression of increasing voltage to produce good torque control. To do this, SCR starter logic offers several common modes of control operation.

The first method is open loop voltage control. An initial starting voltage is applied within a 1-3 second ramp period and adjusted to allow for slack take-up and the initial moving of the empty conveyor. Then a linear timed ramp of voltage is applied. This method will control initial torque and accelerating average torque, but will not produce a linear acceleration of the belt. Also, this method makes no allowance for the variation in required torque for the empty versus the loaded conveyor.

The second method involves the feedback of the drive load current to the drive logic. This can be used as a current limit or current ramp control. In current limit, the output torque is limited to the withstand level of the belt or drive components. This is useful in overmotored drive applications. In the current ramp approach, the current rises linearly until the locked rotor current of the motor is attained and held until the

motor accelerates above breakdown torque speed. This provides a drive that will vary the application rate of voltage between the empty versus loaded conveyor, but does not directly control belt speed. The resultant acceleration produces an S-shaped velocity profile.

The third method involves the feedback of a motor or belt speed from a tachometer. In this method, the drive controls the voltage necessary to produce a linear or piecewise linear belt acceleration. This method will produce repeatable velocity versus time profiles for empty and loaded conveyors. However, the feedback algorithm must be tuned for each conveyor mechanical response. If the belt length is changing, the mechanical response will be changing. This method is often used on belts with severe undulating profiles or optimum fabric sizing.

The drive thermal losses are a function of the forward voltage drop of the SCR devices, normally .5 to .7 volts per SCR device, multiplied by the root mean squared current through the SCR. The motor losses are a function of the square of the current over the applied acceleration period. Since motor power is a function of the product of the applied voltage and current, for low voltages the current is many multiples of the full load during most of the acceleration period. For extended starting periods on loaded conveyors, this will require derating of the motor for multiple starts. The installation of motor stator temperature detectors can aid the control system in making intelligent decisions about motor starting permissives. This system can control starting torques but makes no allowance for speed control when running. The system can handle regeneration loads up to the rating of the motor.

The SCR system requires little maintenance or supporting systems. The cost is low, with the cost of the solid state starter being small multiples of a full voltage line starter. The complexity of the system is low, with plug-in solid state regulators with integral diagnostics. The system can handle simple multiple drives with coarse coordination between starter controls during starting and uncontrolled load sharing according to motor slip curves during running. Conveyor jam torque, again, is only limited by the motor breakdown torque until stall or drive trip. The control system consists of the basic belt controls plus an SCR regulator firing package control with sequencing and diagnostics.

Wound Rotor AC Motor and Direct Coupling Drive

The wound rotor AC motor is a traditional conveyor drive for medium to large conveyors. It is often the drive of choice for conveyor starting for large conveyors, difficult conveyor profiles, or repeated start applications. The wound rotor motor is a special construction of the AC induction motor. In a wound rotor construction, the rotor windings are extended to the outside world by means of collector rings and brushes. By inserting resistance and reactance in series with the rotor windings, the motor controller can manipulate the rotor characteristics referred to the stator at slip speed. The wound rotor then has a family of torque versus speed curves for varying external impedance.

For starting, a larger resistance/reactance value is placed in series with the rotor and then slowly reduced in value as the motor speed is increased. During acceleration, motor currents are low, typically less than 100-150 percent of motor nominal rating. Since most of the rotor heating loss is dissipated in the external network, the rotor temperature increase during starting is low. Since stator currents are low, stator heating is low. At full speed, the rotor windings may be shorted for high running efficiency at low slip, or an external trim or slip resistor may be left in series with the

windings to allow load sharing or limit rate of rise of torque. Rotor voltages vary from several hundred volts on small horsepower to +3000 volts on larger drives.

Traditionally, wound rotor induction motor (WRIM) control has been by a full voltage primary contactor with multiple steps of secondary contactors that bypass progressive steps of external resistance. The resistance is stacked or arranged in steps that allow for discrete increments of acceleration torque. The wattage or thermal rating of the resistance stack determines the start duration and repeat rating of the starter. Usually, a shorting contactor shorts the rotor windings directly for running. This method is termed a “stepper” WRIM controller. To reduce belt starting torque increments, additional steps with contactors are added. Remembering that the rotor current is produced at the slip frequency, which is line frequency at rest, and lowering that frequency with increasing speed, inductors may be used to expand the range of each step. Stepper methods have been the most common type of WRIM through the 1970s.

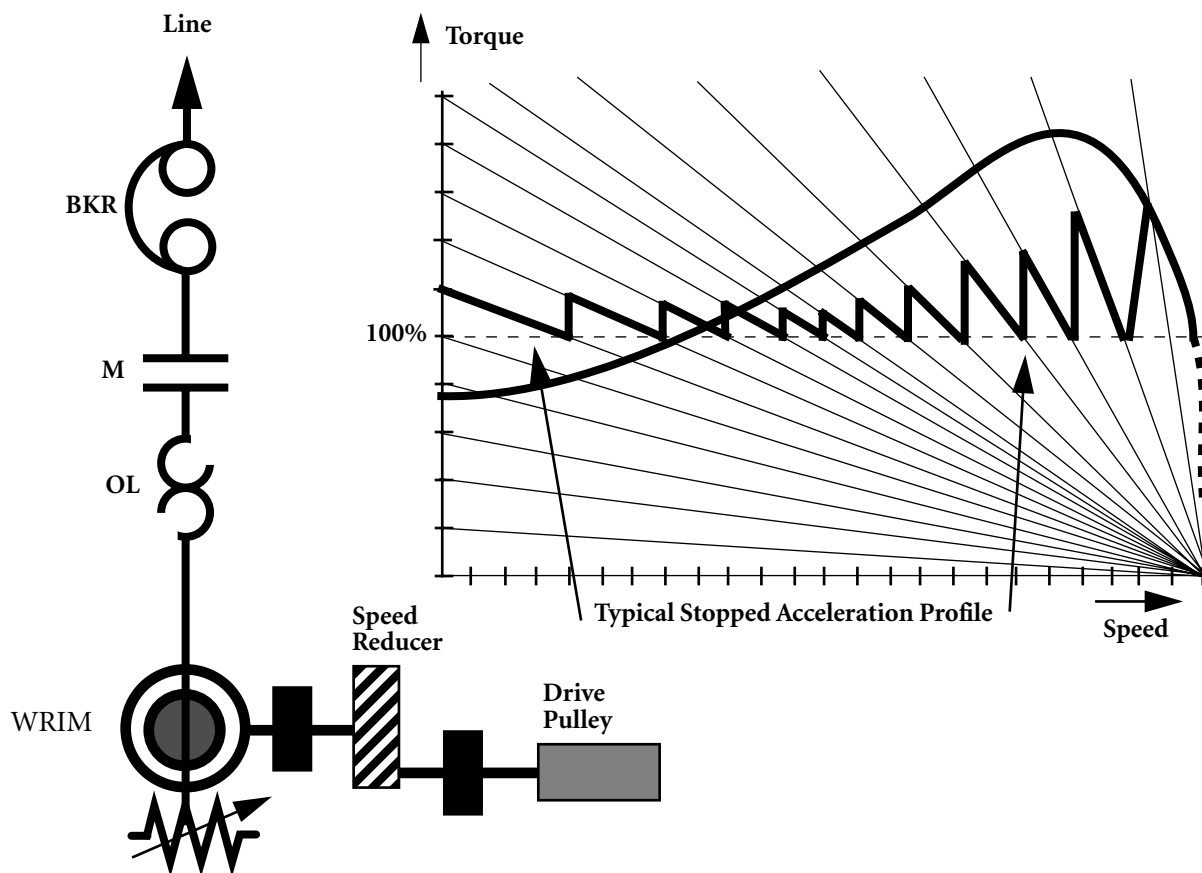


Figure 13.5 Wound rotor induction motor with a stepper secondary control and direct couplings.

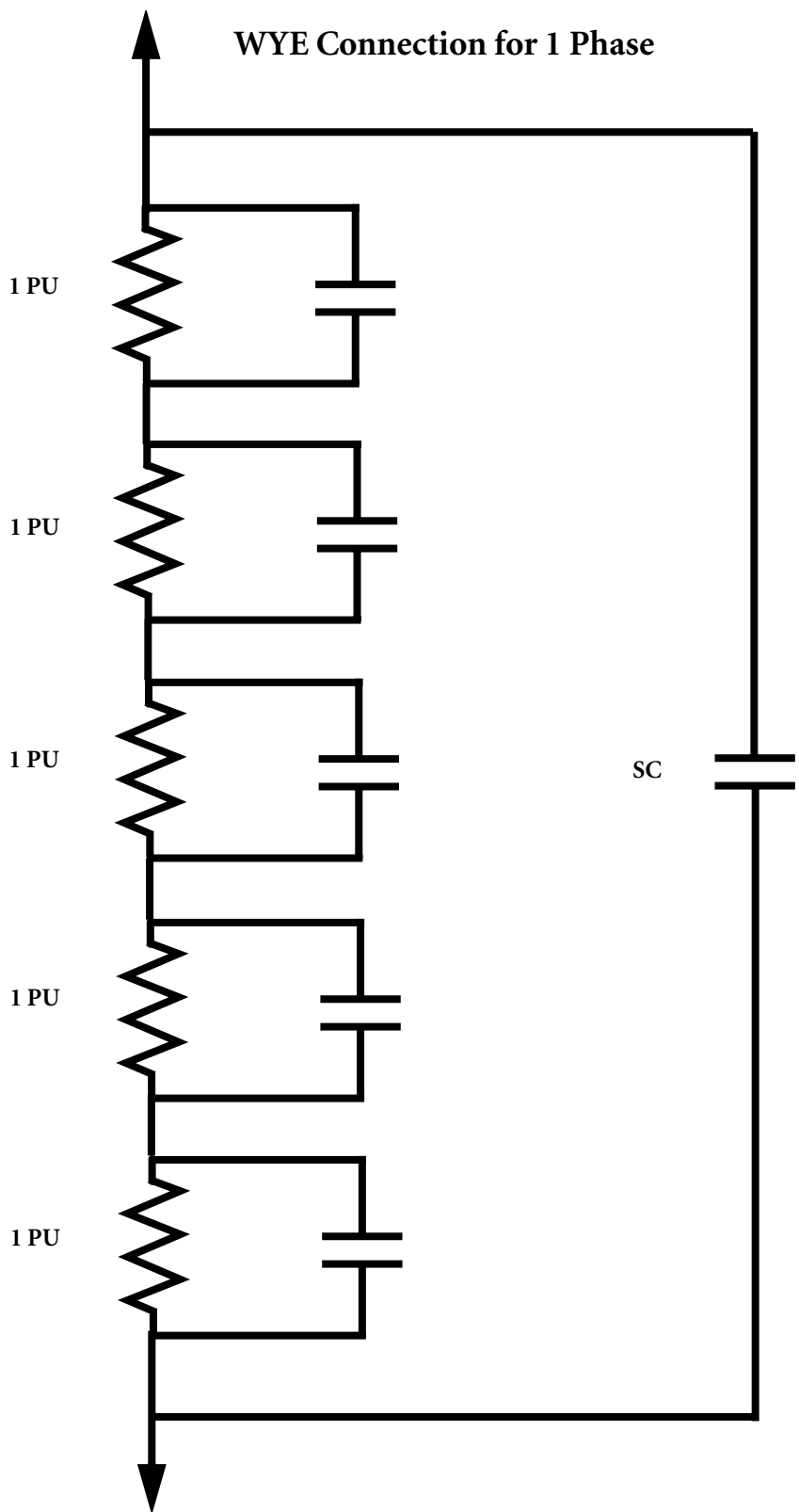


Figure 13.6 Binary stack secondary for WRIM control.

Today, there are alternatives to WRIM stepper control methods that allow for greater flexibility, thermal capacity, and control of WRIM drives:

The first method is an adaptation of the secondary resistance stack into what is termed a “binary stack.” In this method, the resistance is stacked in groups of a binary progression of per unit resistances of 1, 2, 4, 8, 16 and so on. By sequencing the step bypass contactors in a binary progression, an economical number of stepping contactors can produce an almost continuous progression of rotor control. This method is applicable to all sizes of WRIM.

The second method involves the combination of a WRIM primary SCR voltage regulator with several secondary contactor steps. By ramping the primary stator voltage from a minimum value to line voltage, the dynamic range of each secondary resistor step is increased. This method is applicable to smaller WRIM drives where space for the starter is at a premium.

The third method involves the application of SCR devices to time duty switch the secondary resistance presented to the rotor. This is termed “solid state switching of rotor resistance.” By time duty switching control, the effective RMS resistance presented to the rotor may be continuously varied. This produces a stepless control. Since the rotor frequency current is line frequency at rest and decreases with increasing speed, the time switching must synchronize with the rotor frequency. The SCRs are not phase-controlled (turned on at varying phase angles during the current sinusoid) as this would produce adverse rotor harmonics. The SCRs are turned off at zero current crossings. This limits the frequency of the time ratio to less than the rotor slip frequency. This method is most applicable to large WRIM drives where stepless acceleration is required.

The fourth method involves secondary control with “liquid resistors.” A liquid resistor consists of a tank of water/soda ash solution with a conducting probe surface. The liquid resistor is arranged with a set of fixed cylindrical probes and a movable set of probes. For a fixed solution resistivity and probe surface area, the resistor effective resistance can be controlled by the spacing of the movable probe set. On starting, the probes are positioned at maximum spacing. To accelerate the motor, the probes are slowly moved closer to the fixed probe set. The movable probes are driven by a small servo drive, DC drive, or variable frequency drive configured for closed loop computer control. After the probes are moved to a position near the fixed probes, a shorting contactor bypasses the liquid resistor for running. The liquid resistor presents a continuous changing resistance to the rotor. The turndown, or the ratio of the maximum to minimum resistance over the probe travel, for a standard liquid resistor is 30:1. Acceleration is smooth. Stepping torques are avoided. The liquid resistor has a large thermal capacity as sized by the volume of solution and auxiliary solution cooling via heat exchanger. The liquid resistor is flexible for field modification. To lower the resistivity per inch of probe travel, add soda ash to the solution. To raise the resistivity per inch, remove solution and add water. This method is most applicable to larger WRIM drives where stepless acceleration and high thermal rating are required.

Although applicable to all sizes, today WRIM availability is biased toward the medium to high horsepower. It is difficult to find suppliers of small to medium WRIM at an acceptable price. The ability to finely control the starting torque profile is a prime selling point for WRIM drives. The thermal capabilities of the WRIM drive system are large and are dependent on the secondary network rating which is outside the rotating apparatus. The WRIM is primarily a conveyor starting system, but can be configured for variable speed in a 4:1 speed range.

For variable speed, there are large energy losses in the external resistance network so efficiency is low. There is a type of WRIM secondary control where the slip energy is recovered and inserted into the primary power line using solid state inverter techniques and a coupling transformer. This provides continually varying speed with small losses. The WRIM drive can accept overhauling regeneration energy when running up to the motor breakdown rating, but has a severe problem with regeneration during starting with the secondary resistance inserted.

The maintenance of the WRIM is low. Brushes wear about 1-2 mm per 1000 hours. Supporting systems involve maintenance of the external resistor and cooling systems. The cost of the WRIM drive is 200-250 percent of base systems for medium horsepower, and 125-150 percent of base systems for larger horsepower. Still, the WRIM is the most common system on large conveyor drives.

The electrical complexity of the WRIM drive can vary from medium to highly complex. The drive layout is mechanically simple. The WRIM can be configured for multiple drives. Torques can be finely controlled and proportioned during starting. During running, the drives will share load according to the motor torque versus speed curve slopes. A permanent slip trim resistor can permanently bias the load share characteristics at the expense of energy loss. WRIMs usually have high breakdown torque ratings when running shorted. Some designers allow for permanent slip resistors to soften the torque versus speed rise during jam to allow the jam detection time to disconnect the drive. This can require as much as 3-5 percent of additional slip at the expense of energy loss.

Today, the control system for a WRIM drive is usually based on computer control of the secondary resistance switching network based on system parameters such as speed feedback, current feedback, motor temperature, resistance temperature, size of belt load, and multiple drive motor load balance. Computer control apparatuses have made such methods as binary switching simple and effective. Most control systems for WRIM conveyor drives are custom designed for the application.

DC Motor and Direct Coupling

DC motors are commonly available in series, shunt, and compound windings. The motor rotating armature is wound and brought out to the outside world through brushes. For belt conveyors, usually shunt or compound winding DC motors are used. In a DC shunt motor, the base speed is determined by a constant shunt field excitation current. The motor operating speed is a linear function of the armature voltage, while the motor torque depends on the armature current. The base speed can be lowered by increasing the field current and raised by decreasing the field current, within limits. The motor base speed is reached at full armature voltage.

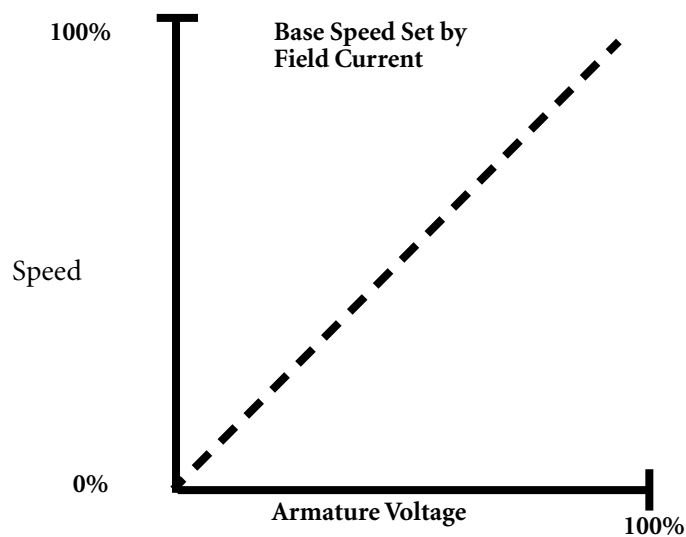


Figure 13.7 DC shunt motor speed vs. voltage.

The full range of DC motor torque versus speed is determined by the motor commutation limit curve. Below base speed, the motor operates in a constant torque region limited by armature current, and above base speed in the constant horsepower region with declining torque limits. The torque limit of DC motors can be large, between 200-300 percent of base rating. DC motors can supply very high forces to a belt conveyor under dynamic conditions unless constrained by the motor controller. In general, DC shunt motors are larger than the equivalent AC motor for the same horsepower, resulting in a higher inertia or WK^2 . The DC motor has a longer drift time on de-energization than an AC motor.

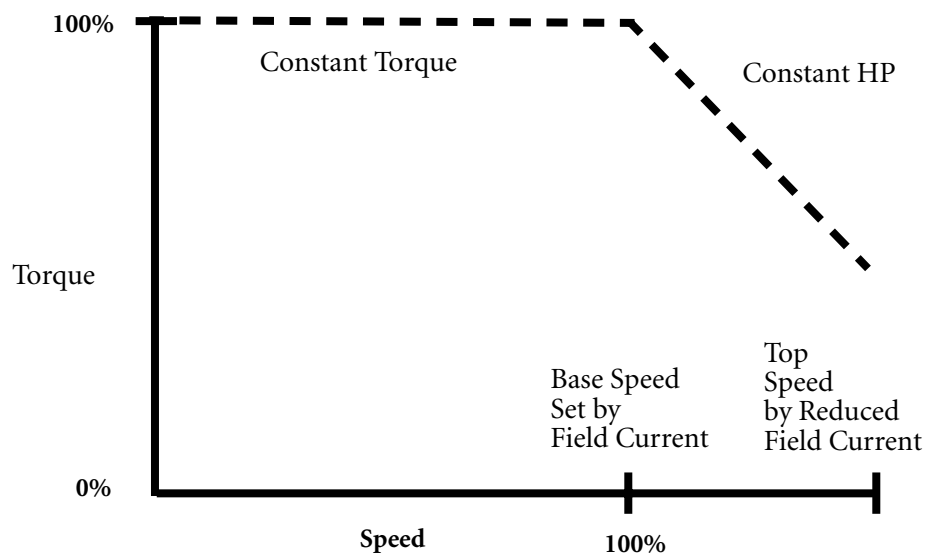


Figure 13.8 DC shunt motor torque vs. speed.

The traditional DC motor controller for a belt conveyor consisted of a contactor controlled system that controlled armature voltage from low to high during starting to running with a constant shunt field excitation. The armature voltage regulating system usually consisted of a series of resistor elements placed in series with the DC motor armature. The system raised voltage by successively removing resistor steps until the motor armature operated on full voltage for running. With the arrival of simple AC belt drive systems, the application of DC drive motors to belt systems became unpopular. This was often due to the losses in dropping resistors, the complexity of the control system, and the maintenance of the resistors and motor brushes.

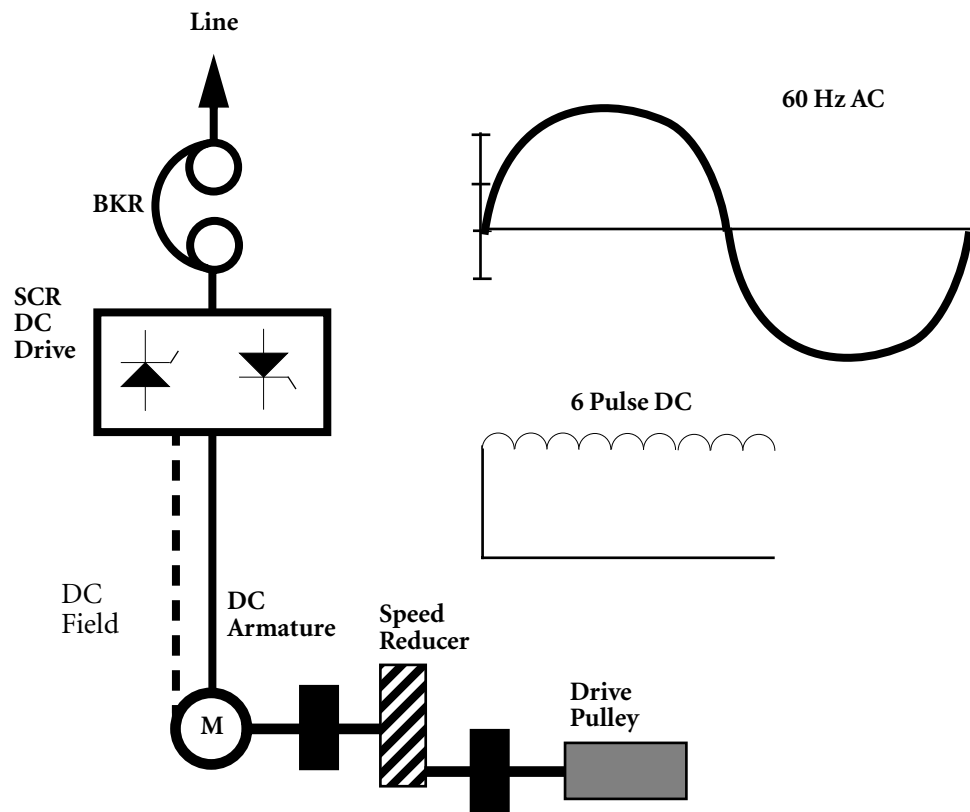


Figure 13.9 DC shunt motor with SCR armature control and direct coupling.

In the late 1960s and 1970s, the development of solid state DC shunt motor control systems, based on silicon controlled rectifiers or thyristors, made continual variable speed operation of DC shunt motors practical in many industrial applications. The thyristor drive offered low losses, fast response, repeatable control, small packaging, good speed regulation, and motor current limit. Reliability of DC drives on thyristor systems today is very good. With 20+ years of mill, mine, and plant duty, the thyristor drive is very mature. Motor brush life on current limited drives with acceleration and deceleration control is very good. The thyristor DC drive phase controls an

AC source to produce a linear variable DC supply for the motor armature while producing a filtered constant DC or a controlled DC for the shunt field. The drive can be configured as an armature voltage regulator or as a speed regulator, both with secondary current loop monitoring and regulation on high currents. The standard voltage regulator will control speed with a 3-5 percent range and the speed regulator from 1-3 percent range from no load to full load. DC drives are considered for belt conveyors requiring variable speed operation for extended periods, multiple full load repeated start capability, very large drives, controlled overhaul regeneration braking, multiple motor active load share, or speed control for process regulation.

DC drives are available in sizes ranging from fractional horsepower to thousands of horsepower. DC motors are available in different enclosures from open to totally enclosed, with self and forced ventilation systems. On very large drives, DC motors are offered in slow rpm models that eliminate speed reducers. There are wraparound motor mounting designs that also eliminate motor bearings from the design.

The DC drive offers precise and repeatable torque control at all speed ranges. The DC motor and controller are thermally rated for continuous operation at all speeds. Multiple repeated starts at reasonable accelerations are allowable. If the conveyor system is capable of overhauling, or if the drive must provide controlled braking force during stopping, the DC drive must be configured with a regenerative four quadrant controller with a second thyristor bridge capable of directing power from the DC motor to the AC power line. The braking or regeneration force can be precisely controlled.

The DC drive system requires a low level of maintenance. The DC motor and cooling system requires periodic inspection and cleaning. The cost of a DC drive system varies widely depending on the size and type of motor enclosure and the complexity of the controller. The costs for a medium to large drive system are often 125 percent to 150 percent of another system configured for starting duty. However, the cost of a DC drive may be considered the base cost for medium to large conveyors with continuous variable speed operation.

The DC drive system is very simple mechanically, but can include complex electronic controls. However, since the control types are mature and widely used in other industries, the reliability and troubleshooting diagnostics are good. DC drives are a good choice for multiple drive systems with precise load share requirements. Multiple DC drives can be configured to "follow" each other with reference to speed, torque, and other system parameters. The response of the DC drive to signal changes is only limited by the mechanical systems' ability to change. In the case of the conveyor jam, the DC drive can limit the motor output torque via a fast acting current limit minor loop. This is normally 150 percent of the nominal rating, but can be set to lower levels if desired. The control system for a DC drive conveyor consists of the normal belt controls, a thyristor firing control regulator, and possibly a belt tachometer for speed regulation of the drive. Thyristor drives are a source power system harmonics. Larger DC drive installation will require a power system study for harmonic and power quality effects on other facility equipment. These effects are greater when the DC drive is configured for regeneration.

There are new variations on the traditional DC drive such as brushless DC. In brushless DC, the motor has stator windings like an AC motor, but has a permanent magnet rotor with north and south poles and a position encoder. The brushless control synthesizes a stator waveform that generates a magnetic field that is always 90 degrees ahead of the rotor position. Brushless DC drives offer full torque at all speeds

with precise motor positioning. Brushless DC is limited presently to small to medium sized drives, at a high cost premium.

AC Induction Motor with Variable Frequency Control and Direct Coupling

In this method, a simple AC induction motor is coupled through a speed reducer to the drive pulley while an electronic controller applies a synthesized AC waveform to the motor stator. The AC motor operating speed depends on the number of stator poles and the applied line frequency.

$$RPM_{ACMotor} = \frac{120 \times \text{Line Frequency}}{\text{Number of poles}}$$

For example, a 60 Hz, 1,200 rpm AC motor will run at 600 rpm when supplied with a 30 Hz voltage source. Since horsepower is torque times speed, this motor will be running at constant torque but variable horsepower according to speed.

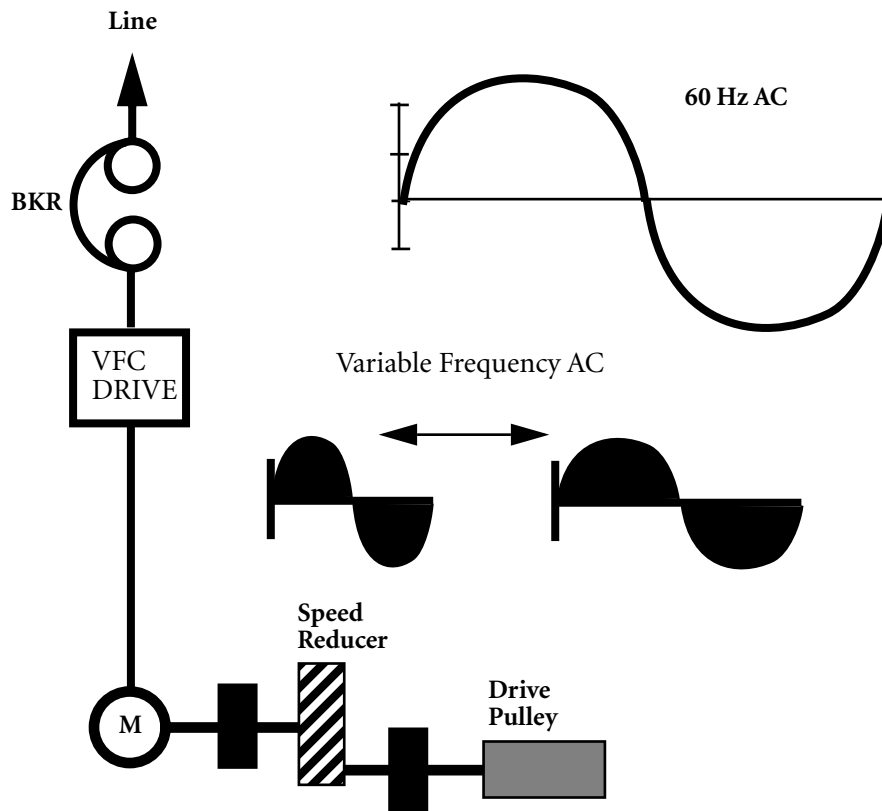


Figure 13.10 Variable frequency AC motor and direct coupling.

The controller that synthesizes the waveform is called a variable frequency inverter or converter (VFI or VFC). Generally, a VFC consists of a rectifier section that inputs line frequency and makes DC, and an inverter section that takes the DC and makes an AC source with frequency and voltage control.

There are three common types of VFC controllers. The most common is the pulse width modulated type. This consists of a static rectifier section and a switching

inverter section. The frequency and voltage are controlled by modulating the on versus off time of the DC link with high speed switching. This type is most common at voltages up to 575 vac from fractional horsepower to 300 horsepower.

The second type is variable voltage inverters. This design uses power SCRs under phase control to vary the DC supply and power SCRs or transistors to switch the output in a six or twelve-step output waveform to approximate a sinusoid current. This type is common in sizes from several hundred horsepower through 1,000 horsepower.

The third design is called a current source inverter. In this type, power SCRs are used to synthesize an AC square wave current. This type is usually used in larger sizes through several thousand horsepower. In most VFCs, the ratio of the output voltage to frequency is maintained as a constant below base speed in order to not overexcite the motor magnetic fields. Some VFCs can be configured to run above base speed, or operate with an output frequency above the input frequency. At above base speed, the VFC operates as a constant horsepower with declining torque rating.

The availability and features of these drives increase each year, while the costs decrease. In certain size ranges, the costs of AC VFC are less expensive than traditional DC. Applications for VFCs for conveyors will increase. Presently, VFC type drives are most commonly found on small belts used as part of sample or process control systems, very large belts for sophisticated variable speed operation, or retrofits where the AC direct coupled drive motors are in place.

VFC drives are available in sizes from fractional horsepower to thousands of horsepower. Since the motor is operating on a variable frequency, variable voltage controlled source at all times, torque control for starting is precise. For a given frequency output, the motor speed is proportional to frequency less slip.

The thermal rating of the drive is based on the rating of the VFC electronics and the motor cooling capability. Since the electronics are inherently current limited, the VFC will not be run overloaded. The motor rating at speed less than base speed may decrease due to low cooling from the motor fan and increased losses due to synthesized waveforms. Extended operation at low speed may require an external motor cooling fan. VFCs can normally operate as variable speed drives over a 10:1 speed range. Operation at very low speeds may result in torque pulsations to the conveyor.

VFCs vary in their ability to accept and conduct regenerated overhauling energy. If the belt profile can result in overhauling, the VFC must be equipped with regeneration or dynamic braking options. The belt designer must quantify the overhauling forces to equip the VFC with adequate capacity.

The reliability of early generation VFCs was poor. The reliability of new VFCs is good if properly installed and protected. Most VFC installation will require a voltage adjusted power supply transformer and extensive surge suppression. The major supporting systems required for VFC installations are training of maintenance and spare electronic parts. In general, the present costs of VFC systems are higher than other drive systems.

VFC drive systems are mechanically simple, but electrically complex. VFC drives can be configured for multiple drives with active controlled load sharing. In conveyor jam, the VFC responsive current limit will set an upper limit for output motor torque. This is nominally 150 percent on smaller VFCs and as low as 110 percent on larger units. The control system for a VFC drive system consists of the normal belt permissive controls, a microprocessor-based electronic regulator, a power electronic converter with protection, and sometimes a belt speed transducer. There are some large

synchronous AC motors in a wraparound design with cycloconverter type VFC drives operating at very low rpms.

AC Induction Motor with Full Voltage Starting and Fixed Fill Fluid Coupling

This method of conveyor drive has become the most common base case for medium to large conveyors configured for starting duty and simple belt profiles. The fixed fill fluid coupling is normally mounted mechanically between a normal AC induction motor and a speed reducer. The motor is started across the line. The fluid coupling allows the AC motor to start unloaded. The load accelerates slowly based on the transmitted torque from the coupling, while the coupling absorbs slip energy.

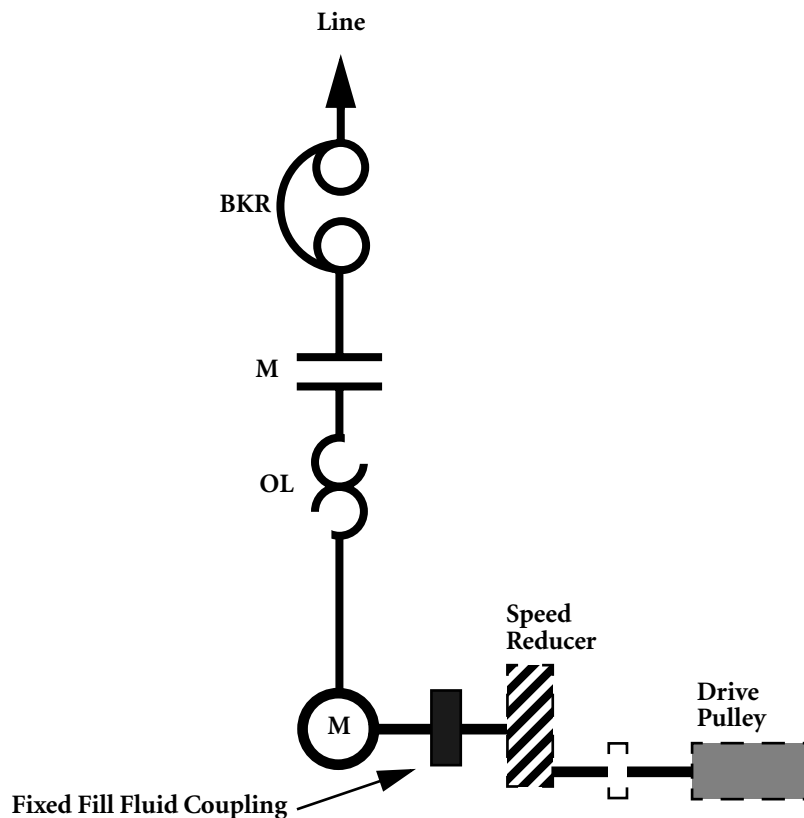


Figure 13.11 AC motor with fixed fill fluid coupling.

In simple forms, a fluid coupling consists of an input impeller attached to the prime mover, an output impeller connected to the driven machinery, and a shell casing that contains a fixed volume of fluid positioned between the impellers. Within this closed volume, the operating fluid circulates in a continuous vortex between the input impeller and the output impeller (runner). The combination is known as the working circuit and the measurement across the outside diameter of the vane toroidal impellers as the profile diameter.

The torque transmitted by the coupling is proportional to the difference in moment of the fluid as it enters and leaves each member. The speed difference, or slip, creates a net difference in opposing centrifugal heads of the impeller and runner to circulate fluid against friction and shock losses within the working circuit. The

torque required from the prime mover will be only that required by the runner plus that required to sustain the losses. Because fluid couplings are centrifugal machines, the laws of geometry apply. Therefore, for the same percent slip and filling of the working circuit, the torque transmitted is proportional to the fifth power of the working circuit profile, the cube of the input speed, and the density of the working fluid. For a given frame size, the torque versus speed slip characteristics are altered by changing the fluid fill. The contained fluid is usually thin mineral oil, but synthetics with good constant viscosity characteristics are available. By virtue of the viscous coupling, the transmission of torque transients from motor to belt, or belt to motor, are damped.

The advantage of fixed fill fluid couplings is simplicity. For belts, the standard fluid coupling has three problems. The starting torque transmitted after motor starting may be too high, above 200 percent. This results in rapid belt acceleration and possible shock waves. To extend the start period, the fluid fill must be lowered, resulting in coupling fluid heating during starting and running. The coupling transmission of torque is uncontrolled and uncoordinated with belt loading.

To this end, modifications have been made to most standard couplings now used for belts. First, the coupling reserve of fluid has been expanded outside the working chamber of the circuit into delay chamber areas. At rest, these areas fill with fluid, lowering the working circuit level. At start, the working circuit lower level transmits an initial lower level of torque, 125 percent to 140 percent. As the coupling accelerates, centrifugal force causes fluid to flow from the delay chamber through orifices into the working area of the coupling. This raises the transmitted torque slowly, allows for extended acceleration periods, runs the coupling at a more efficient fluid level, and expands the thermal capability of the coupling. The coupling start time may be adjusted by orifice selection and the final transmitted torque level by fluid fill. Couplings with delay chambers require a period at rest between starts to allow gravity flow of the fluid to the delay chamber. On some couplings, there are multiple orifices opened by centrifugal switches as the coupling accelerates the load. To increase thermal capacity for multiple starts, couplings can circulate working oil through expanded coupling storage chambers while running to increase coupling radiated heat rates. Most couplings have a fast heat dissipation rate when running and a slow rate when at rest.

In applications on conveyors, couplings have to be oversized to obtain adequate thermal capacity for multiple starts. If the coupling overheats, a temperature sensor can shut down and hold off the drive or, at extreme overheat, the working fluid dumps through a melting alloy fusible soft plug as an orifice relief.

The fluid coupling drive attributes are impressive. Fluid couplings are available in all sizes from very small to very large. The coupling is rated as a torque device employed on the high speed motor shaft. Thus the coupling size for the same horsepower is smaller at higher speeds. For fluid coupling drives, 1800 rpm motors are often chosen. The fluid couplings method of torque control is an engineering decision on the desired coupling type, size, orifice, and fluid fill. The orifice and fluid fill may be adjusted in the field with care.

The effect of fluid fill on torque and thermal rating is extremely non-linear. The coupling offers no controlled torque by the control system. The thermal rating of the coupling is limited. To increase coupling rating, the designer must increase coupling reservoir volume to a larger size, increase fill, decrease delay fill time, or provide forced ventilation. The fluid coupling is not capable of operating a conveyor at vari-

able running speeds. The fluid coupling can transmit overhauling regeneration forces to the motor when running up to the slip limit of the coupling. However, the holding force during starting is limited. The properly applied fluid coupling requires little maintenance and no supporting systems.

The cost of fluid couplings is considered the base case for medium to larger drives. The cost of couplings for small to medium drives is 125 percent of reduced voltage SCR starters. Fluid coupling applications are mechanically and electrically simple. The coupling mounting may impose some overhung loading on motor or speed reducer bearings. Fluid couplings can be used on multiple drive applications for starting. During running, the drives will share load according to the torque versus speed curves of the drive motors and the slip characteristics of the couplings. No active control is available. For conveyor jam, the fluid coupling will limit the rate of rise and amount of output torque of the motor to the slip level of the coupling, usually 125-175 percent. The control system for fluid coupling drives consists of the base case of belt permissives and sensors for fluid coupling overtemperature and fluid dump. The control system should provide for a minimum idle time for drain back and excessive re-start prevention.

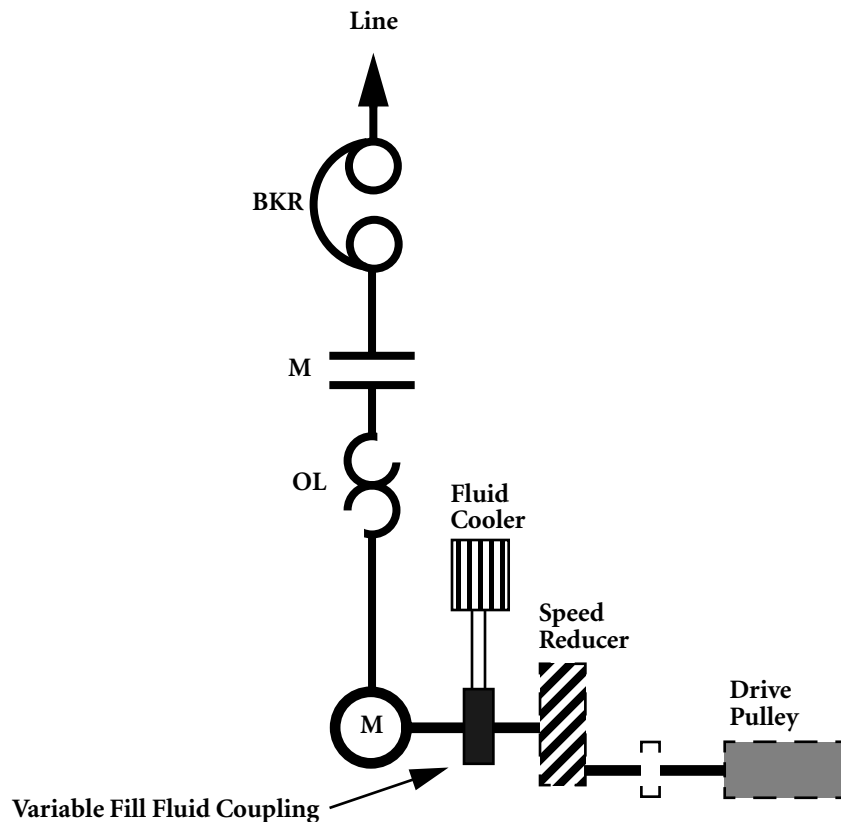


Figure 13.12 AC motor with variable fill hydrokinetic coupling.

AC Induction Motor with Full Voltage Starting and Variable Fill Hydrokinetic Coupling

By adding a system for controlling the fill of the fluid coupling during starting and running, the variable speed and controlled torque requirements of some complex conveyors with a hydraulic drive system are addressed. Again, the AC induction motor with line starting is the prime mover. There are many commercial designs of variable fill hydraulic drives. The type most commonly applied to conveyors is the scoop tube or scoop trim control drive. In this system, the hydraulic working circuit is contained in an inner casing surrounded by a rotating reservoir casing. A sliding scoop tube inserted in a stationary axial position to the reservoir determines the suction level for a working circuit supply of oil. By mechanically moving the scoop tube actuator, the fill level of the working circuit may be controlled. The fluid passes to a sump where a separate pump returns fluid to the working circuit. This system is more complex than fixed fill couplings, but allows for greater fluid capacity for thermal rating. There are also the less common types of leak-off and put-and-take couplings.

In the variable fill hydraulic drive, an external control positions the scoop tube actuator for transmitted torque control. This type of drive has a torque versus speed curve for each fill level. The drive can control starting torques and vary running speed over a 6:1 ratio. While slipping, the drive loses energy in waste heat in the coupling fluid. An auxiliary cooling system is designed to remove the waste heat. This drive is often used on conveyors that require precise torque control and the ability to run at variable speed for limited periods. The variable fill hydraulic drive is usually applied on medium to very large conveyors.

The variable fill hydraulic drive can be found in all sizes. It offers precise torque control, but the speed of response can be slowed by the scoop tube positioner and fluid fill delay time. Also, since no unique position of the scoop tube delivers the same torque, a feedback regulator must adjust the drive speed for different belt loadings. The thermal rating of the drive system depends on the volume of the fluid and the auxiliary cooling system. The drive can run at variable speeds at the expense of heat losses. The drive can accept regeneration during running up to the limit of the coupling slip at each speed. The drive response to overspeeding during overhauling can be slow. The drive requires regular maintenance of the supporting hydraulic fill and cooling equipment. The cost of this drive type is medium to high depending on size. Positioned between the motor and speed reducer, the drive can be mechanically complex with supporting system piping. The drive can be electrically complex due to feedback regulation and scoop tube position actuators. This drive can be used on multiple drive belts with starting and running torque control between drives. With conveyor jam, the drive can limit the output of motor torque to the slip level of the coupling at any speed. The control system for this drive consists of the base belt control permissives, the control of the coupling level actuator, and control interlock to the supporting systems. Most hydraulic drives require a belt speed feedback for proper regulation.

AC Induction Motor and Variable Mechanical Transmission Coupling

Horsepower is the product of torque multiplied by speed. For an AC induction motor prime mover with constant speed, the driven belt speed and torque is a function of the speed reducer ratio and drive pulley diameter.

A variable mechanical transmission drive is a class of equipment that attempts to vary the effective speed reducer ratio continuously while running. The method of varying mechanical ratio is different and sometimes proprietary with respect to transmission manufacturer. In a typical type of drive, the AC motor starts across the line and accelerates to full speed at no load. The drive input shaft is connected to a sun

gear planetary speed reducer. The sun gear rotates the three planet gears that spin an outside ring gear. The ring gear is coupled to the output shaft by a hydroviscous clutch disk pack called a reaction brake. By controlling the hydraulic pressure on the clutch pack, the running slip may be varied. The clutch pressure is slowly ramped to apply a retarding force to the ring gear. The output shaft speed increases in reverse relationship to the speed of the ring gear. At full speed, the reaction brake is fully locked. The hydraulic oil circulates from the hydroviscous clutch brake to an external cooler for heat dissipation.

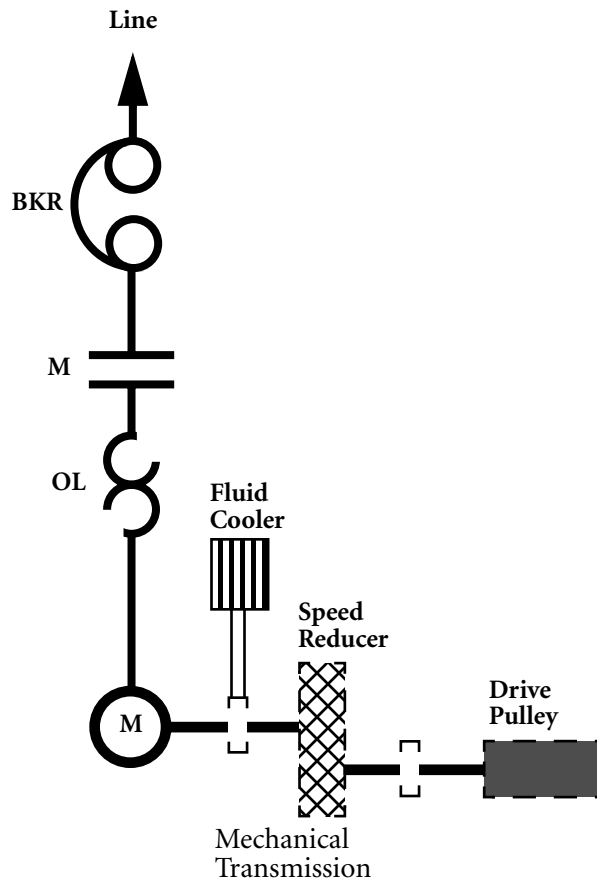


Figure 13.13 AC motor and variable mechanical transmission coupling.

The variable transmission can provide a starting and running speed control system over an extended range. The variable transmission replaces the starter and speed reducer in the system. The variable transmission drive is commonly used on medium to large sized conveyor drives where precise starting control and some variable speed operation are required. There are other commercial mechanical speed ratio type drives that can be used for conveyor starting systems.

Variable speed mechanical drives are available from fractional horsepower through large sized applications. In some systems, variable transmission component ratings limit the size of the horsepower and the gear reduction ratio. The variable type drive offers continuous and precise torque control with response determined by the hydraulic pressure response time.

The thermal rating of the drive is determined by the operation time at variable speed and the rating of the external oil cooling system. The drive can operate over a variable speed range for continuous and extended periods over a limited speed ratio depending on size. The variable transmission drive can handle regeneration overhauling force up to the limit of the clutch pack slipping force when running at less than full speed, and up to motor rating at full speed.

The variable transmission system requires a medium level of supporting systems for operation related to the hydraulic power unit with oil cooling and oil circulation. The life and strength rating of the gear units can be equal to or less than traditional speed reducer units. The cost of the variable transmission drive type varies from 125-175 percent of the base fixed fill fluid coupling drive in medium sized applications. The layout feature of the combined coupling speed reducer unit may reduce overall capital and installation costs.

The system is medium in mechanical and electrical complexity. The variable transmission system can be configured with multiple drives with active load sharing during starting and running. With conveyor jam, the variable transmission system will provide a slip torque limit when running at less than lock up speed, but will pass full motor breakdown torque when running at lock up. The control system for the variable transmission consists of the base case of conveyor permissives, a regulator for the hydraulic pressure to the clutch packs, the control of the auxiliary cooling, and protective devices for transmission overheating.

Other Conveyor Drives

There are other less common types of conveyor drives that may be encountered in use. An eddy current coupling is a magnetic coupling, non-contact, smooth rotor, and stator that couples a constant speed AC motor to a load. By varying the excitation magnetic field on the coupling, the slip may be controlled. Waste heat is dissipated in the coupling frame. Eddy current couplings were used historically on larger conveyor systems, but are not in common use today.

There are hydroviscous drives that couple between the AC motor and the load. The hydroviscous drive is a fluid drive that depends on the amount of pressure between impeller units to determine the coupling torque. Slip energy is dissipated in the fluid. These units are less common on conveyors than hydrokinetic fluid couplings.

Hydrostatic drives are adaptations of fluid couplings where a reaction impeller is placed in the working circuit between the drive impeller and the output runner. By controlling the reaction impeller slip, the output runner torque may be manipulated. This is the principle of many automotive type torque converters.

Creep or inspection drives are auxiliary pony drives coupled to the prime mover drive high speed shaft by overruning clutches, roller chains, or belts that allow the conveyor to operate at a highly reduced speed for maintenance purposes. Most creep

drives are not back-driven in normal operation. There are other drive types that the user may encounter that are beyond the scope of this discussion.

Conveyor Belt Controls

The belt conveyor system requires control systems. The control system architecture is composed of a subset of the drive type controller and the belt control system. The drive type controller provides for the starting, running, and stopping speeds and torque control of the drive type. The belt control system issues run and stop commands to the drive controller. The belt control system provides for belt protection controls, interlocking, telemetry, and man-machine interface.

Drive Type Controller

Each drive type requires a controller for starting, running, and stopping. This may be a simple line voltage contactor for the full voltage, across the line starting with direct coupling or fixed fill fluid coupling conveyor drive. This may involve several electrical contactors and devices in the case of wound rotor induction motor drives. This controller may involve complex electronic power controllers in DC drives or variable frequency drives. The controller may involve the sequencing of support systems such as the scoop tube actuator in variable fill fluid drives or fluid circulating or cooling systems. Some controllers are involved in only starting. Others must support drive functions for variable speed operation and ramped stopping. The conveyor designer should identify and separate the drive type controller from the general belt control system.

Starting, Running, and Stopping Control Algorithms

Each drive type controller carries out a control algorithm for starting the conveyor. The most common type of control used on smaller to medium sized drives with simple profiles is termed “open loop acceleration control.” In open loop, the control system is previously configured to sequence the starting mechanism in a prescribed manner, usually based on time. In open loop control, drive operating parameters such as current, torque, or speed do not influence sequence operation. This method presumes that the control designer has adequately modeled drive system performance on the conveyor.

For larger or more complex belts, “closed loop” or “feedback” control may be used. In closed loop control, during starting the control system monitors via sensors such drive-operating parameters as the current level of the motor, the speed of the belt, or the force on the belt, and modifies the starting sequence to control, limit, or optimize one or more parameters. Closed loop control systems modify the starting applied force between an empty and fully loaded conveyor. The constants in the mathematical model related to the measured variable versus the system drive response are termed the tuning constants. These constants must be properly adjusted for successful application to each conveyor. The most common schemes for closed loop control of conveyor starts are tachometer feedback for speed control and load cell force or drive force feedback for torque control.

On some complex systems, it is desirable to have the closed loop control system adjust itself for various encountered conveyor conditions. This is called “adaptive control.” These extremes can involve vast variations in loadings, temperature of the belting, location of the loading on the profile, or multiple drive options on the conveyor.

There are three common adaptive methods. The first involves decisions made before the start, or “prestart conditioning.” If the control system could know that the belt is empty, it would reduce initial force and lengthen the application of acceleration force to full speed. If the belt is loaded, the control system would apply pre-tension forces under stall for less time and supply sufficient torque to adequately accelerate the belt in a timely manner. Since the belt only was loaded during previous running by loading the drive, the average drive current can be sampled when running and retained in a first-in-first-out (FIFO) buffer memory that reflects the belt conveyance time. Then at shutdown the FIFO average may be used to precondition some open loop and closed loop setpoints for the next start.

The second method involves decisions based on drive observations that occur during initial starting or “motion proving.” This usually involves comparing the time of the drive current or force against the belt speed. If the drive current or force required early in the sequence is low and motion is initiated, the belt must be unloaded. If the drive current or force required is high and motion is slow in starting, the conveyor must be loaded. This decision can be divided into zones and used to modify the middle and finish of the start sequence control.

The third method involves a comparison of the belt speed versus time for this start against historical limits of belt acceleration, or “acceleration envelope monitoring.” At start, the belt speed is measured versus time. This is compared with two limiting belt speed curves retained in control system memory. The first curve profiles the empty belt when accelerated, and the second one profiles the fully loaded belt. Thus, if the current speed versus time is lower than the loaded profile, it may suggest that the belt is overloaded, impeded, or that there is a drive malfunction. If the current speed versus time is higher than the empty profile, it may suggest a broken belt, coupling, or a drive malfunction. In either case, the current start is aborted and an alarm is sent.

Belt Protection Controls

The belt conveyor is provided with various devices and systems for protecting the system. These devices are used as run-permissive input commands to the general belt control system.

Belt Alignment

It is important that the belt stay aligned with the drive pulleys, and the carrying and return idlers. Belt alignment sensors are typically positioned along the edges of the conveyor fabric. They are usually at the discharge and at the loading zones of the conveyor, but can be distributed along the conveyor at intervals. Switches consist of roller switches, limit switches, whisker switches, proximity switches, or photoelectric switches. When the edge of the belt trips the alignment switch for a timed period, the conveyor is halted. There is provision for running the conveyor in local “jog” to re-align the conveyor fabric. An adaptation of alignment sensors for large steel cord belts is the continuous measurement of edge displacement, termed “edge tracking.” Edge tracking in steel cable belts provides an indication of tension distribution within the carcass among the support cables. Upon installation, each steel cable belt will exhibit an edge tracking signature for a belt revolution. A deviation in the edge tracking displacement at a later time suggests a problem with the belt cable tension distribution.

Belt Overload

The belt conveyor system is protected from overload by the motor overload of the electric drive motors(s). The motor overload can be a simple bi-metallic or melting

eutectic alloy, or a complex computer-based motor thermal model. A belt loading “sail or paddle” switch can sense a belt overload at a specific point. Complex belts are sometimes protected from overload by belt weigh scales that measure the belt loading at a point. Or, a non-contact belt profile sensor, such as an ultrasonic, radar, laser, or video device can measure the belt loading depth. Based on an assumed material density, the loading tons per hour can be projected.

Belt Slip

Belt slip is the loss in traction of the drive pulley(s) to the belt cover. Belt slip can destroy a belt or drive pulley. Belt slip protection provides a belt drive speed sensor that compares the present belt speed with the belt intended speed. For constant speed belts this normally consists of a slip switch with a setpoint that trips the conveyor drive when the belt speed is below 80 percent of full speed. The belt slip switch is bypassed during starting and stopping. For variable speed conveyors, the belt slip consists of a speed sensor that measures belt speed and compares with the speed reference sent to the drive system. When the belt speed is below 80 percent of the intended speed, the drive is tripped. This type of belt slip is active during starting, running, and stopping. In multiple pulley adjacent drives, tachometers are sometimes provided for each drive motor. The tachometer signals are compared with the resultant normalized belt speed to sense slippage of one of the multiple drive pulleys. A method to adjust and test belt slip is normally provided in the belt control system.

Take-up Overtravel

Overtravel limit switches can be placed at the far extremes of the counterweight or take-up device travel. In a gravity counterweight take-up, the top overtravel switch trip may suggest a jammed conveyor fabric condition. A bottom overtravel switch may indicate belt stretch, or a broken belt fabric flight. Excessive take-up motion during starting and stopping indicates an inadequate or malfunctioning drive type control.

Transfer Chute Plug

A plugged chute device provides belt protection at the discharge of the conveyor into a transfer chute. Blocked flow can result in damage to the driven conveyor. Plugged chute switches are used in many configurations depended on the application. Actuation of the plugged chute switch with time delay normally results in the tripping of the conveyor drive.

Bin Level

When conveyors discharge into bins or hoppers, bin level sensors provide protection to the belt. These can consist of simple hanging tilt switches or analog measurement devices such as ultrasonic, radar, or laser devices.

Pullcord Stations

Pullcord stations are distributed stop switches with latching provisions. They are normally provided on large, important, or complex unguarded conveyors, or where required by local regulation. They can be located along the conveyor at intervals and interconnected with a colored pull wire. An operator activates the switch by pulling the pullcord until the switch trips, normally raising a visual indicator flag. The switch remains tripped until reset manually at the switch location. Tripping of the pullcord stops the conveyor drive. The belt does not restart on reset of the pullcord. Tripping of the pullcord is a control stop, and should not be considered a “lockout” of the conveyor power source.

Conveyor Belt Controls

Rip Detectors Rip detectors provide belt protection in the case of a rip or tear in the belt fabric. Simple rip detectors are usually spill switches located below the center of the belt near the point of belt loading. Complex belt rip detectors on larger belts involve the embedding of antennas with the belt construction. If the antenna is cut by a rip, signal sources and detectors located along the edge of the belt detect a broken antenna and shutdown the belt. Complex belt rip detection systems require periodic maintenance.

Fire Detection Some belts that carry combustible materials are provided with fire detection protection systems. These systems can consist of point or distributed thermal trip switches located above the belt fabric, smoke sensors, carbon monoxide sensors, or fiber optic temperature sensors.

Dust Sprays Some belts that carry dusty material are provided with water or chemical-based dust suppression systems. These systems spray the belt material at selected transfer and belt loading point. Some systems spray a constant amount of suppression per unit time and some vary the application with the loading of the belt. The dust sprays are turned off when the belt is idle or unloaded to prevent puddling, waste, and slippage.

Belt Control Belt control normally consists of the net sum of the belt permissives, the operator start/stop station(s), the start warning system, interlock, sequencing of individual conveyors, and other process controls. Belt control issues a run command to the drive type controller. Sometimes, the belt control issues a running reference speed to the drive type controller. For stopping, the belt control simply removes the positive run signal to the drive type controller or initiates a ramped stop command.

Stop/Start A belt conveyor system is normally provided with one or more control stations for operators. Start stations normally require a momentary operator input to initiate a start sequence. Stop stations normally monitor a maintained input for a run permissive. Jog stations are special start stations that allow the conveyor to run only when the operator stays at the jog station and continually commands the jog function. Some complex belts have many operator stations distributed at various physical locations. Again, a stop/start station is a control device and should not be considered a “lockout” of the conveyor power source.

Start Warning Some unguarded conveyors are equipped with an audible or visual system that provides a pre-start warning along the entire length of the conveyor. These normally consist of horns, sirens, flashing lights, or strobes. These are activated for a period after a start is requested but before initiating motion of the conveyor.

Interlock Classically, interlock is the run permissive for the conveyor to any other unit’s run status. Interlock is the control relationship between adjacent material transferring or interdependent machines. Interlock proceeds through a system in the reverse order of material flow. For example, a belt conveyor named “A” transfers material to another belt conveyor named “B.” A is interlocked to B. If B shuts down, A must shut down. Interlock then flows from B to A. Running a belt “out of interlock” or “in bypass” are

common terms for operation of a conveyor with the interlock system disabled or defeated. Interlock can be performed by physically wiring the conveyor control systems together, by computer coding interlock, by providing a motion sensing switch on the tail of conveyor B and sensing that as a run permissive for conveyor A, or by telemetry signaling from one conveyor to the next over a distance. A conveyor may be interlocked to other machinery and devices such as screens, breakers, crushers, or magnets.

Telemetry

Telemetry is the distribution of belt control and informational signals over significant distances. Since conveyors transport material over wide areas, some belts require signal telemetry. Signal telemetry can be simply multi-conductor cables with DC digital on/off control. Signal telemetry can involve the multiplex of more than one signal over a single wire path. Today, telemetry may involve the conversion of electrical signals to computer-based serial transmission of data, to light signals run over fiber optics, or to wireless radio transmission. Control, remote operator interface, and conveyor monitoring can be distributed geographically away from the physical conveyor location using commercial telephone networks and modem technology.

Lockout

Lockout of a belt conveyor is the physical lockout of all motive power sources to the conveyor so that people may access the conveyor equipment for service, inspection, clean-up, or maintenance. Lockout implies security supervision of the lockout elements. Lockout involves all sources of power including electrical, hydraulic, pneumatic, or stored kinetic energy. Each drive type and conveyor system will require an assessment of lockout requirements. Lockout equipment or apparatus may be required to comply with owner practices and policies, manufacturer's recommendations, and regulatory requirements. The lockout system must be interfaced with the belt control system.

Belt Control Apparatus

Each belt control system is supported by a control apparatus. Traditional industrial control depends on relays, timers, counters, and discrete input devices such as pushbuttons and selectors. Today, most belt conveyor systems use some form of computer control. Each belt control system communicates with the operators and attendants through a system of man-machine interface (MMI).

Computer-Based Control

Computer-based control implies real time control by a microprocessor-based system with an input section, a processing section, and an output section. Examples of computer-based control for belt conveyors include programmable logic controllers (PLC), distributed control systems (DCS), personal computers (PC), and single board custom microcontrollers. In industrial domains, the PLC is the most common control platform. Computer control provides a programming logic environment where individual changes and features may be set up to match each conveyor's application needs. Computer control is robust, reliable, cost effective, and flexible. The input sensors are the conveyor and equipment protection switches and transducers. Inputs are digital or analog. Examples of digital inputs are belt alignment or chute plug switches. Digital inputs have two conditions, "on" and "off." Digital inputs are normally powered by some low level power source such as 120 vac or 24 vdc. Examples of analog inputs are belt motor currents, bearing temperatures, vibration, bin levels, and hydraulic pressure. Analog transducers measure some physical property

and convert the measurement to a linear electrical input range. Common analog input signals are 4-20 ma, 1-5 vdc, and 0-10 vdc. The processing section of the computer control evaluates the inputs, applies the control logic, rules, and algorithms, and controls the output section. The processing section also supports serial communication to programming terminals, data highways, networks, and printers.

The output devices are the action elements controlled by the belt control system. Outputs are digital or analog. Digital outputs are normally powered by low voltage AC or DC of common control ranges. Examples of digital outputs devices are starters, relays, lights, horns, solenoids, and valves. Analog output devices take an electrical analog signal and provide some proportional physical action. Examples are valve positioners, gate actuators, belt speed control devices, and feeder rate controls. Computer control systems support local input/output (i/o) and remote i/o. Local i/o is located with the processing unit and remote i/o is distributed up to several miles from the processing unit. Remote i/o communicates with the processing unit over some serial multiplex telemetry method unique to each computer control. The computer control can communicate with other conveyor computer control systems via serial data links, data highways, or local area networks.

Man-Machine Interface

The man-machine interface (MMI) is the sum total of the devices that communicate information to the belt system operators, attendants, service staff, and management. Examples of low level MMI include pilot lights, horns, counters, ammeters, voltmeters. Low level MMI communicate with the control system via discrete i/o. Examples of mid-level MMI include alphanumeric text displays, voice synthesizers, keypads, and small LCD screen displays. Mid-level MMI communicate with the control system via discrete i/o, binary or BCD parallel ports, or serial data. Larger MMI stations included CRT, flatscreen TFT, or gas plasma graphic terminals with operator input via keyboard, membrane operator panels, bezel arrangements, and touchscreens. Larger MMI stations are controlled by computers and communicate with the control PLC via serial or local area network protocols. MMI terminals can provide status information, allow for operator input, display alarm messages in the order of occurrence with date and time, display trends, and log information to permanent storage. MMI terminals may be located some distance away from the belt conveyor. The cost of higher level MMI is decreasing. The packaging options are increasing. User configuration is becoming friendly.

Conclusion

The best belt drive system is one that provides acceptable performance under all belt load conditions at a reasonable cost with high reliability. No one drive system meets all needs. The belt designer must define the drive system attributes required for each belt. In general, the AC induction motor with full voltage starting is confined to small belts with simple profiles. The AC induction motor with reduced voltage SCR starting is the base case for belts from small to medium sizes. With recent improvements, the AC motor with fixed fill fluid couplings is the base case for medium to large conveyors with simple profiles. The wound rotor induction motor drive is the traditional choice for medium to large belts with repeated starting duty or complex profiles that require precise torque control. The DC motor drive, variable fill hydrokinetic drive, and the variable mechanical transmission drive compete for application on belts with extreme profiles or variable speed at running requirements. The choice

is dependent on location environment, competitive price, operating energy losses, speed response, and user familiarity. AC variable frequency drive applications are limited to conveyors that require precise speed control due to higher present costs and complexity. However, with continuing competitive and technical improvements, the use of synthesized waveform electronic drives will expand.

Operation, Maintenance, and Safety

Operation
Maintenance
Safety
Guidelines for Safe Operation and Maintenance

The preceding chapters of this book have provided engineering information that can form the basis for the proper application and design of high quality belt conveyors and conveyor systems. To acquire optimum performance, dependability, and cost economy, equal importance and consideration must be given to the installation, safe operation, and proper maintenance of the system and all related equipment.

Generally, each conveyor system is installed in a different location and facility and will have unique performance requirements, design features, and operating environment. All of these special considerations must be studied and evaluated when developing the best operating procedures and the safest work place.

Because of the number and range of these variables, this chapter can provide only a brief overview of some of the most important and generally applied practices. Conveyor manufacturers are often able to offer assistance in establishing the best operating maintenance and safety programs.

Operation

Characteristically, the operation of truck and rail haulage systems for bulk materials requires a relatively large number of trained personnel. Operators are needed for scheduling and master control of the system, as well as to dispatch individual units and drive large vehicles. Employees are also required for such additional operations as loading and discharge and weighing of the material.

Many belt conveyors, when appropriately designed, installed, and operated, will perform continuously and dependably with as few as one or two operators. One basic requisite is that the material being handled by the conveyor has the originally specified physical properties and is fed uniformly and at the design rate.

Performance of a system can be continuously monitored by a combination of modern electrical controls, built-in safety sensors and devices, closed circuit TV, and other signal systems. A central control panel can be equipped so that an operator can monitor performance even at remote areas.

Depending on the length and complexity of a system, one or perhaps two trained mechanics should patrol the system at regular intervals to detect any conditions or components that need attention. On a 5½-mile overland system, two mechanics and a supervisor can easily perform such inspections.

At the time of installation and during trial runs of a large and complex conveyor system, it is advantageous to offer a program of instruction and discussion for all personnel who will be involved in operation, safety, and maintenance of the system. Such programs should be repeated at sufficiently frequent intervals so that trained personnel can update their knowledge and new employees can become indoctrinated.

Maintenance

It is important that belt conveyor maintenance be performed only by well trained and competent personnel who are provided with proper test equipment and good tools. They should receive training in the value and conduct of a comprehensive preventive maintenance program.

Although one stuck idler roll may not appear important, maintenance personnel should realize that under a high-speed belt handling abrasive material, its shell could soon wear through, presenting a knife edge which could severely damage an expen-

sive belt. Well-trained personnel would be able to detect impending failure in such a case, and correct the malfunction before any damage could occur.

The conveyor belt often represents a high proportion of the total conveyor cost. Since the composition and construction of the belting makes it vulnerable to accidental damage and/or accelerated wear, belt operation and maintenance deserve special attention in a good training program to minimize replacement and repair cost.

Premature wear or accidental damage may result from loading improper grades, sizes, or volumes of conveyed material onto the belt. Also, foreign materials, such as tramp iron, spikes, timbers, and the like, when entrained in the flow, can cause expensive shutdowns and costly repairs.

Table 14-1 lists the causes and cures of a wide range of operational problems present in order of probably occurrence that relate to the belt. It can serve as good reference, both in a training course and as a maintenance instruction.

It is highly recommended that before initial operation of a conveyor on production runs there should be careful and detailed inspection of the conveyor and all of its components. Following such a close inspection, test runs should be made to allow careful observation of actual operation before conveying material. During these inspections and test runs, the alignment of all the mechanical components and the operating alignment of the belt on the carrying and return idlers should be checked. Refer to Chapter 5 for information on belt and idler alignment.

Inspection should also establish that there are no construction materials, tools, or projecting members that can rub, tear, or cut the belt when it starts up. Chute skirtboards should have been installed so that they do not touch the belt. The rubber edging strips on the skirts should be adjusted so that they rest only lightly on the belt surface. Belt scrapers should be observed and a final adjustment made, if necessary.

Modern electrical control systems can incorporate computers and other automatic means for measuring performance and controlling such functions as weighing, mixing, blending, and material flow paths. Sensors and other devices for indicating maintenance requirements and unsafe conditions may be an integral part of an electric control system. The electrical control system should be given a thorough check-out and dry run, or “bug out,” during the initial test runs.

Good housekeeping is essential for dependable operation and low-cost maintenance. A build-up of material on the deck can brake and eventually stop idler rolls, resulting in increased belt tension and possible damage to the belt. Spillage onto the return belt can also injure it as lumps are squeezed between the belt and pulleys. Scrapers on the return belt at the point where it enters the foot pulley may be desirable in some cases.

A build-up of sticky or frozen material on pulleys or idlers can cause belt misalignment and other malfunctions that can damage a belt. Pulley scrapers and/or the application of soft rubber as a pulley lagging may help to alleviate the condition.

Caution: No one should be allowed to attempt to remove the adhering material manually unless the conveyor has been stopped and the master electrical control locked off.

Table 14-1. Belt conveyor troubleshooting – causes and cures.

COMPLAINT	CAUSE						COMPLAINT	CAUSE					
	(See Table 14.2)							(See Table 14.2)					
	In Order of Probable Occurrence							In Order of Probable Occurrence					
Belt runs off at tail pulley	7	1	1	2	—	—	Excessive wear, including rips, gouges, ruptures, and tears	1	2	1	2	8	5
	5	4	7	1				2	5	7	1		
Entire belt runs off at all points of the line	2	1	1	2	4	1	Excessive bottom cover wear	2	1	5	1	2	2
	6	7	5	1		6		1	4		9	0	2
One belt section runs off at all points of the line	2	1	1	—	—	—	Excessive edge wear, broken edges	2	4	1	8	1	2
		1						6		7			1
Belt runs off at head pulley	1	2	2	1	—	—	Cover swells in spots or streaks	8	—	—	—	—	—
	5	2	1	6									
Belt runs to one side throughout entire length at specific idlers	1	1	2	—	—	—	Belt hardens or cracks	8	2	2	1	—	—
	5	6	1						3	2	8		
Belt slip	1	7	2	1	2	—	Covers become checked or brittle	8	1	—	—	—	—
	9		1	4	2				8				
Belt slip on starting	1	7	2	1	—	—	Longitudinal grooving or cracking of top cover	2	1	2	1	—	—
	9		2	0				7	4	1	2		
Excessive belt stretch	1	1	2	6	9	9	Longitudinal grooving or cracking of bottom cover	1	2	2	—	—	—
	3	0	1					4	1	2			
Belt breaks at or behind fasteners; fasteners tear loose	2	2	1	2	2	1	Fabric decay, carcass cracks, ruptures, gouges (soft spots in belt)	1	2	5	1	8	2
		3	3	2	0	0		2	0		0		4
Vulcanized splice separation	1	2	1	2	2	9	Ply separation	1	2	1	8	3	—
	3	3	0	0				3	3	1			

Table 14-2. Belt conveyor troubleshooting – complaints and recommended resolutions.

1 Belt bowed — Avoid telescoping belt rolls or storing them in damp locations.* A new belt should be straightened out when “broken in” or it must be replaced.

2 Belt improperly spliced or wrong fasteners — Use correct fasteners. Retighten after running for a short while. If improperly spliced, remove belt splice and make new splice.* Set up regular inspection schedule.

3 Belt speed too fast — Reduce belt speed.

4 Belt strained on one side — Allow time for new belt to “break in.” If belt does not break in properly or is not new, remove strained section and splice in a new piece.*

5 Breaker strip missing or inadequate — When service is lost, install belt with proper breaker strip.

6 Counterweight too heavy — Recalculate weight required and adjust counterweight accordingly.* Reduce takeup tension to point of slip, then tighten slightly.

7 Counterweight too light — Recalculate weight required and adjust counterweight or screw takeup accordingly.

8 Damage by abrasives, acid, chemicals, heat, mildew, oil — Use belt designed for specific condition. For abrasive materials working into cuts and between plies, make spot repairs with cold patch or with permanent repair patch. Seal metal fasteners or replace with vulcanized step splice. Enclose belt line for protection against rain, snow, or sun. Don’t over-lubricate items.

9 Differential speed wrong on dual pulleys — Make necessary adjustment.*

10 Drive underbelted — Recalculate maximum belt tensions and select correct belt. If line is over-extended, consider using two-flight system with transfer point. If carcass is not rigid enough for load, install belt with proper flexibility when service is lost.

11 Edge worn or broken — Repair belt edge. Remove badly worn or out-of-square section and splice in a new piece.

12 Excessive impact of material on belt or fasteners — Use correctly designed chutes and baffles. Make vulcanized splices. Install impact idlers. Where possible, load fines first. Where material is trapped under skirts, adjust skirtboards to minimum clearance or install cushioning idlers to hold belt against skirts.

13 Excessive tension — Recalculate and adjust tension. Use vulcanized splice within recommended limits.

14 Frozen idlers — Free idlers. Lubricate. Improve maintenance. (Don’t over-lubricate.)

15 Idlers or pulleys out-of-square with center line of conveyor — Realign. Install limit switches for greater safety.

16 Idlers improperly placed — Relocate idlers or insert additional idlers spaced to support belt.*

17 Improper loading, spillage — Feed should be in direction of belt travel and at belt speed, centered on the belt. Control flow with feeders, chutes, and skirtboards.

18 Improper storage or handling — Refer to the manufacturer for storage and handling tips.

19 Insufficient traction between belt and pulley — Increase wrap with snub pulleys. Lag drive pulley. In wet conditions, use grooved lagging. Install correct cleaning devices for safety. See item 7, above.

20 Material between belt and pulley — Use skirtboards properly. Remove accumulation. Improve maintenance.

21 Material build-up — Remove accumulation. Install cleaning devices, scrapers, and inverted “V” decking.* Improve housekeeping.

22 Pulley lagging worn — Replace worn pulley lagging. Use grooved lagging for wet conditions. Tighten loose and protruding bolts.

23 Pulleys too small — Use large-diameter pulleys.

24 Radius of convex vertical curve too small — Increase radius by vertical realignment of idlers to prevent excessive edge tension.

25 Relative loading velocity too high or too low — Adjust chutes or correct belt speed. Consider use of impact idlers.

26 Side loading — Load in direction of belt travel, in center of conveyor.

27 Skirts improperly placed — Install skirtboards so that they do not rub against belt.

*Consult supplier for additional recommendations and procedures.

As with all good machinery, a well-developed and mandatory lubrication program is essential for low maintenance cost and dependable operation. Because of the relatively large number of bearings in the idler rolls, and their influence on belt tensions and horsepower requirements, their lubrication is very important. Life expectancy will be enhanced by following the idler manufacturer's recommendations as to the type of lubricant, the amount and frequency of application, and the type of greasing equipment to be used.

To optimize the dependability and productivity of conveyors and to minimize maintenance costs, it will be advantageous to stock certain types of repair parts. A well developed maintenance program will provide for emergency repair of both mechanical and electrical types of equipment, and include provision of parts, parts lists, and appropriate drawings.

Climatic conditions may require some additional considerations. Special lubricants are sometimes necessary for subzero operation so as to avoid overload of the drives and undesirably large increases in belt tensions. On applications where the belt may periodically be covered with moisture or frost, it may be advisable to operate the belt empty for a brief period at startup. Application of a belt scraper on the drive pulley side of the belt, just ahead of the point where the belt engages the drive pulley, may be a great help.

It has been shown that modern belt conveyor electrical control systems employ relatively sophisticated units. This book explains and illustrates some of the modern automation that has been employed, including programmable controllers, computers, multiplexing, and solid-state control. Although these types of equipment are highly durable, the proper maintenance and servicing of large and complex systems can only be accomplished by specialists in this field.

CEMA member companies can be of assistance in solving special problems.

Safety

The many years of experience of leading conveyor engineers and manufacturers have shown that the development and maintenance of a safe work place requires the combined effort and cooperation of the several organizations that may become involved in a belt conveyor installation and operation.

Belt conveyor safety generally begins with sound design that, as far as is practical, avoids foreseeable dangers and hazards. Diligence in safety considerations must be applied during the course of manufacture, installation, and establishment of operating and maintenance policies and procedures.

Generally, the accidents that cause personal injury in connection with a belt conveyor are not the result of faulty design or component failure. It has been found that most accidents are caused by human carelessness, negligence, or lack of training in operations and awareness of possible hazards. In as many instances, the conveyor equipment will have a perfect safety record for some years before an improperly trained or careless worker will cause, and likely become involved in, an accident.

After the design and installation are developed and supervised by qualified engineers familiar with recognized safety features and requirements, the next priority should be personnel training. Operating and maintenance personnel and their supervisors should be initially, and then subsequently, instructed in safe operating proce-

dures, recognizable hazards, precautions, and the maintenance of a safe work place. They should also be provided with the proper tools and equipment to operate and maintain the conveyor in an adequately safe condition. Employees who do not have proper training should be made aware of, and forbidden to enter, hazardous areas.

Since World War II, the Conveyor Equipment Manufacturers Association has taken an active role in the development of the most widely recognized and accepted safety standard for conveyors, entitled "Safety Standard for Conveyors and Related Equipment." The eighth edition was approved through the American National Standards Institute for issuance as ASME Standard B20.1-1996. B20.1 suggests that ASME Standard B15.1, "Safety Standard for Mechanical Power Transmission Apparatus," should be used in conjunction with it.

The stated purpose of Standard B20.1 is to present certain guidelines for the design, construction, installation, operation, and maintenance of conveyors and related equipment. These guidelines and recommended safety practices will be of assistance in establishing an appropriately safe work place. It is important to realize that the best design and safety features can be useless in conjunction with faulty maintenance and operating practices.

The broad scope and fine detail of ASME Standard B20.1 preclude its inclusion in this book. However, it is highly recommended that those responsible for assuring safety in the engineering, manufacture, installation, operation, and/or maintenance of belt conveyor systems and equipment acquire and use Standard B20.1 as a reference and guide.

Guidelines for Safe Operation and Maintenance

The following general safety guidelines are not extracted directly from, and cannot take the place of, the more complete and detailed information available in the ASME Standards B20.1 and B15.1. A brief listing is presented here simply to illustrate the type of safety considerations generally applicable to belt conveyor installations and equipment.

1. At a time close to completion of installation, all personnel and supervisors should be given a complete indoctrination in the use of the system and all of its equipment. Field inspection and classroom techniques are two valuable types of training.

2. A formal safety training program for operations, maintenance, and supervisory personnel will go a long way toward establishing and maintaining the highest standards of safety in the work place.

3. Concurrent with completion of the installation and the trial runs of all belt conveyors and associated equipment, a "Safety Checkup" is recommended. The checkup should include all mechanical and electrical operating equipment, plus the structures, walkways, ladders, stairs, headrooms, and access ways. It is at this time that a detailed physical inspection of the facility and the installed conveyor equipment will often reveal the need for additional guarding, safety devices, and warning signs.

4. At no time should the conveyors be used to handle material other than that originally specified. Capacity and belt speed design ratings should not be exceeded.

5. Only trained personnel should be allowed to operate the conveyor system. They should have complete knowledge of conveyor operation, electrical controls, safety and warning devices, and the capacity and performance limitations of the system.

6. The location and operation of all emergency control and safety devices should be made known to all personnel. Surrounding areas should be kept free of obstructions or materials that could impede ready access and a clear view of such safety equipment at all times.

7. A program should be established to provide frequent inspections of all equipment. Guards, safety devices, and warning signs should be maintained in their proper positions and in good working order. Only competent and properly trained and authorized persons should adjust or work on safety devices.

8. A “walking inspection” of a belt conveyor system is a good means by which well-trained maintenance personnel can often detect potential problems from any unusual sounds made by such components as idlers, pulleys, shafts, bearings, drives, belts, and belt splices.

9. Hands and feet should never come in contact with any conveyor component, and no one should be allowed to ride on a moving or operable conveyor. Poking at or prodding material on the belt or any component of a moving conveyor should be prohibited. Contact with, or work on, a conveyor must occur only while the equipment is stopped, with the electrical control locked off.

10. No person should be allowed to ride on, step on, or cross over a moving conveyor, nor to walk or climb on conveyor structures, without using the walkways, stairs, ladders, and crossovers provided.

11. Good housekeeping is a prerequisite for safe conditions. All areas around a conveyor, and particularly those surrounding drives, walkways, safety devices, and control stations, should be kept free of debris and obstacles, including inactive or unused equipment, components, wiring, and obsolete or nonapplicable warning signs or posted instructions.

12. Any conveyor found to be in an unsafe condition for operation, or one that does not have all guards and safety devices in excellent condition, should not be used unless adequate supplementary safety devices are installed.

13. All persons should be barred by appropriate means from entering an area where falling material may present a hazard. Warning signs and barricades can be used.

14. First-class maintenance is a prerequisite for the safest operation of conveyors. Maintenance, including lubrications, should be performed with the conveyor stopped and locked out. Special lubricating equipment, lube extensions, pipes, and the like can be installed so as to permit lubrication of an operating conveyor without any foreseeable hazards.

15. Good lighting contributes to a safe working environment.

16. During the life of a belt conveyor system, its operational conditions and environment may require changes. There should be a continuing effort to detect and treat promptly any new possible safety hazards associated with these changes. If such a hazard cannot be readily eliminated, warning signs, barricades, or posted instructions should be installed.

With the increasing use of belt conveyors in the transportation of bulk materials, the number and severity of accidents have been reduced. When conveyors are used as a means of transport in the place of vehicular units, such as railcars and trucks, the problem of traffic-related accidents is minimized. Also, environmentally related health problems can be easily limited by the elimination of dust hazards. Indeed, belt conveyors have substantially reduced the hazards present in practically any other method of bulk materials handling. The further reduction in number and severity of

accidents will be a direct result of applying and enforcing the safe practices of design, installation, operation, and maintenance such as have been described here and in ASME Standard B20.1.

Guide for Use of SI (Metric) Units

Table A-1 is reproduced with permission of The American Society of Mechanical Engineers from the ASME Guide SI-1, Orientation and Guide for Use of SI (Metric) Units, ninth edition, March 24, 1982.

Portions of the text and Figure A.1 are extracted from the American National Standard ANSI/IEEE 268-1992 Metric Practice, and are reprinted with permission.

In the preparation of *Belt Conveyors for Bulk Materials*, the Bulk Conveyor Engineering Committee of the Conveyor Equipment Manufacturers Association carefully studied the practicality of several possible schemes of incorporating metric practice into the book.

It was found practically impossible to integrate a soft conversion of English and SI units throughout the text including the extensive engineering data, formulas, and examples. Appendix A is therefore presented to provide assistance to the reader who wishes to develop certain basic belt conveyor calculations in SI terms.

Conversion Factors

Table A-1 offers a list of commonly used conversion factors extracted from ASME guide SI-1, *Orientation and Guide for Use of SI (Metric) Units*. Conversion factors are presented for ready adaptation to computer readout and electronic data transmission. Each factor is written as a number greater than one and less than ten, with six or fewer decimal places. This number is followed by the letter E (for exponent), a plus or minus symbol, and two digits indicating to power of 10 by which the number must be multiplied in order to obtain the correct value.

Thus: 3.523 907 E – 02 becomes $3.523\,907 \times 10^{-2}$
or 0.035 239 07

Similarly:

 3.386 389 E + 03 becomes $3.386\,389 \times 10^3$
or 3,386.389

An asterisk (*) after the last significant digit indicates that the conversion factor is exact and that all subsequent digits are zero. All other conversion factors have been rounded. Where less than six decimal places are shown, greater precision is not warranted.

Example

<i>To convert from:</i>	<i>to:</i>	<i>Multiply by:</i>	<i>Result:</i>
pound-force per square foot	Pa	4.788 026 E+01	1 lbf/ft ² = 47.880 26 Pa
inch	m	2.540 000*E – 02	1 inch = 0.0254 m (exactly)

The conversion factors for other compound units can easily be generated from numbers given in the alphabetical list by the substitution of converted units.

Example

To find the conversion factor of lb•ft/s to k•gm/s:
first convert 1 lb to 0.453 592 4 kg
and 1 ft to 0.3048 m
then substitute: (0.453 592 4 kg)(0.3048 m)/s
 = 0.138 255 kg•m/s
thus the factor is 1.382 55 E–01

Example

To find the conversion factor of oz•in² to k•gm²:
first convert 1 oz to 0.028 349 52 kg
and 1 in² to 0.000 645 16 m²
then substitute: (0.028 349 52 kg)(0.000 645 16 m²)
 = 0.000 018 289 98 kg•m²
thus the factor is 1.828 998 E–05

Table A-1. Commonly used conversion factors.

Quantity	Conversion	Factor		
Plane Angle	degree to rad	1.745	329	E-02
Length	in to m	2.54*		E-02
	ft to m	3.048*		E-01
	mile to	1.609	344*	E+03
Area	in ² to m ²	6.451	600*	E-04
	ft ² to m ²	9.290	304*	E-02
Volume	ft ³ to m ³	2.831	685	E-02
	US gallon to m ³	3.785	412	E-03
	in ³ to m ³	1.638	706	E-05
	oz (fluid, US) to m ³	2.957	353	E-05
	liter to m ³	1.000	000	E-03
Velocity	ft/min to m/s	5.08*		E-03
	ft/sec to m/s	3.048*		E-01
	km/h to m/s	2.777	778	E-01
	mile/h to m/s	4.470	4*	E-01
	mile/h to m/s	1.609	344*	E+00
Mass	oz (avoir) to kg	2.834	952	E-02
	lb (avoir) to kg	4.535	924	E-01
	slug to kg	1.459	390	E+01
Acceleration	ft/s ² to m/s ²	3.048*		E-01
	std. grav. to m/s ²	9.806	65*	E+00
Force	kgf to N	9.806	65*	E+00
	lbf to N	4.448	222	E+00
	poundal to N	1.382	550	E-01
Bending, torque	kgf-m to N•m	9.806	65*	E+00
	lbf-in to N•m	1.129	848	E-01
	lbf-ft to N•m	1.355	818	E+00

The factors are written as a number greater than one and less than ten with six or fewer decimal places. The number is followed by the letter E (for exponent), a plus or minus symbol, and two digits which indicate the power of 10 by which the number must be multiplied to obtain the correct value. For example:

1.745 329 E-02 is 1.745 329 X 10⁻² or 0.017 453 29

*Relationships that are exact in terms of the base units.

Table A-1. Commonly used conversion factors.

Quantity	Conversion	Factor		
Pressure, stress	kgf/m ² to Pa	9.806	65*	E+00
	poundal/ft ² to Pa	1.488	164	E+00
	lb/ft ² to Pa	4.788	026	E+01
	lbf/in ² to Pa	6.894	757	E+03
Energy, work	Btu (IT) to J	1.055	056	E+03
	Calorie (IT) to J	4.186	8*	E+00
	ft lbf to J	1.355	818	E+00
Power	hp (550 ft lbf/s) to W	7.456	999	E+02
Temperature**	°C to K	$^tK = ^t°C + 273.15$		
	°F to K	$^tK = (^t°F + 459.67)/1.8$		
	°F to °C	$^t°C = (^t°F - 32)/1.8$		
Temperature interval	°C to K	1.0**		E+00
	°F to K or °C	5.555	556	E-01

The factors are written as a number greater than one and less than ten with six or fewer decimal places. The number is followed by the letter E (for exponent), a plus or minus symbol, and two digits which indicate the power of 10 by which the number must be multiplied to obtain the correct value. For example:

1.745 329 E-02 is 1.745 329 X 10⁻² or 0.017 453 29

*Relationships that are exact in terms of the base units.

**°F should be converted to °C. Absolute temperatures should be converted to K.

Metric Practice and Units for Mass, Force, and Weight

The following information is very pertinent to the application of metric practice and units to the basic calculations of belt tensions and horsepower requirements as presented in Chapter 6. The information contained herein is copyrighted material of the IEEE, extracted from IEEE Std 268-1992, *American National Standard for Metric Practice*, copyright ©1992 by the Institute of Electrical and Electronics Engineers, Inc. This information was written within the context of the IEEE Std 268-1992. Information is reproduced with the permission of the IEEE.

Mass, Force, and Weight

In the gravimetric system of units, the same names (e.g., kilogram, pound, ounce) have often been used for both mass and force. This has led to serious confusion. In SI this confusion is eliminated because the unit of mass is the kilogram, and the unit of force is the newton (see Fig. A.1). The kilogram force (from which the suffix “force” in practice has often been erroneously dropped) shall not be used. Derived units that include force shall be formed using the newton. In all systems, distinct unit names shall be used; in the inch-pound system, for example, pound (lb) shall be restricted to mass, and pound-force (lbf) to force.

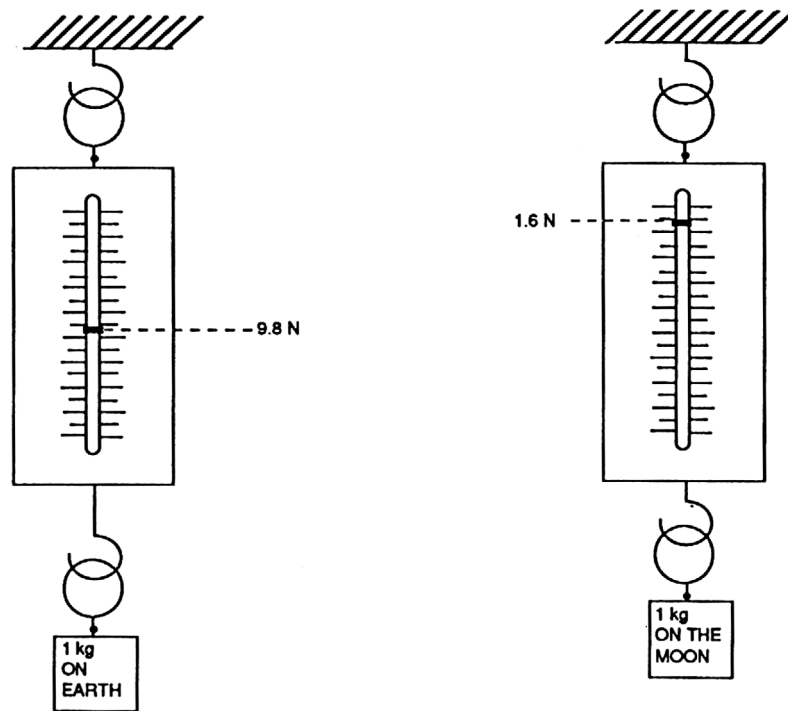


Figure A.1 Illustration of difference between mass (unit: kilogram) and force (Huntington).

Weight

The term “weight” has been used to mean either mass or the force of gravity acting on the mass, and its representation in terms of SI units depends on how it is used.

In many technical fields and in commercial and everyday use, the term “weight” is usually used as a synonym for mass. Where the term is so used, weight is expressed in kilograms in SI. The verb “to weigh” means “to determine the mass of” or “to have a mass of.”

In some fields of science and technology the term “weight” is defined as the force of gravity acting on an object, i.e., as the product of the mass of the object and the local acceleration of gravity. Where the weight is so defined, it is expressed in newtons in SI. The terms “force of gravity” or “weight-force” should be used where this meaning is intended.

Load

The term “load” means either mass or force, depending on its use. A load that produces a vertically downward force because of the influence of gravity acting on a mass may be expressed in mass units, e.g., kilograms.

A load that produces a force from anything other than the influence of gravity shall be expressed in force units, i.e., newtons.

Capacity Rating

The capacity rating of an engineered structure is intended to define the mass that can be supported safely. Such a rating should be expressed in a mass unit rather than a force unit, thus in kilograms or metric tons, as appropriate, rather than newtons.

Torque

The determination of the correct SI units for torque and for other related quantities depends on whether the radian is used or omitted in expressions for derived units.

When the radian is omitted the unit for torque is N m, the unit for moment of inertia is kg m^2 , and the unit for moment of momentum is $\text{kg m}^2/\text{s}$. These units are convenient to use in the solution of problems in statics. In this case there is no problem with dimensional consistency.

When the radian is used, the unit for torque is N m/rad, the unit for moment of inertia is $\text{kg m}^2/\text{rad}^2$, and the unit for moment of momentum is $\text{kg m}^2/(\text{rad s})$. In the solution of problems that involve rotation, the use of these units will retain all advantages of dimensional analysis.

SI (Metric Units) for Belt Conveyor, Belt Tension, and Horsepower Calculations

The following information and a sample problem will be of assistance in applying SI metric units to the calculation of belt tensions and horsepower requirements of belt conveyors.

Metric Use of CEMA Belt Tension Formulas

The basic measurement of force in the United States is pounds force, lbf. This comes from the expression:

$$F = Ma$$

Where:

$$M = \frac{W}{g}$$

And:

$$a = \text{ft/sec}^2$$
$$W = \text{lbs weight}$$

The units are:

$$F = \frac{W}{g} a = \left(\frac{\text{lbs}}{\text{ft}/(\text{sec}^2)} \right) (\text{ft}/\text{sec}^2) = \text{pounds force, lbf}$$

In the SI system the corresponding units are:

newtons, N for force

kilograms, kg for mass

meters/ sec^2 , m/sec^2 for acceleration

By definition, a newton is the force required to accelerate a mass of one kilogram by one meter per second per second:

$$N = \text{kg} \times \text{m}/\text{sec}^2$$

Any expression for force in newtons must have the units kg x m/sec². Pages 102-103 provide the summary of the components of the effective belt tension, T_e .

$$\begin{aligned}
 T_x &= \text{idler friction} &= L \times K_x \times K_t \\
 T_{yc} &= \text{belt flexure, carrying idlers} &= L \times K_y \times W_b \times K_t \\
 T_{yr} &= \text{belt flexure, return idlers} &= L \times 0.015 W_b \times K_t \\
 T_{ym} &= \text{material flexure} &= L \times K_y \times W_m \\
 T_m &= \text{material lift (+) or lower (-)} &= \pm H \times W_m \\
 T_p &= \text{pulley resistance} &= \text{See Chapter 6} \\
 T_{am} &= \text{accelerate material} &= 2.8755 \times 10^{-4} \times Q \times (V - V_o) \\
 T_{ac} &= \text{accessories} &= \text{See Chapter 6} \\
 T_e &= T_x + T_{yc} + T_{yr} + T_{ym} \pm T_m + T_p + T_{am} + T_{ac} \\
 T_e &= LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) + T_p + T_{am} + T_{ac}
 \end{aligned}$$

To calculate T_e in SI units, it is necessary to express all of the above values of tension in newtons, N , and weight in kilograms, kg , and linear measure in meters, m .

$$\begin{aligned}
 W_b &= \text{Weight of belt, kg/m} \\
 W_m &= \text{Weight of material, kg/m} \\
 L &= \text{Length, m} \\
 H &= \text{Lift or lower, m, (plus for lift, minus for lower)} \\
 K_t &= \text{Factor, same as in other units} \\
 K_y &= \text{Factor, same as in other units} \\
 A_i &= \text{Force to rotate idlers, } N \\
 S_i &= \text{Idler spacing, m}
 \end{aligned}$$

$$\begin{aligned}
 K_x &= 0.00068(W_b + W_m)9.807 + \frac{A_i}{S_i}, N/m \\
 &= 0.006669(W_b + W_m) + \frac{A_i}{S_i}, N/m
 \end{aligned}$$

(To convert A_i values in Chapter 6 to newtons, N , multiply by 4.45.)

$$\begin{aligned}
 V &= \text{Belt speed, m/sec} \\
 V_o &= \text{Initial velocity of material, m/sec} \\
 Q &= \text{Capacity, kg/sec} \\
 &\quad 1 \text{ metric tonne} = 1,000 \text{ kg}
 \end{aligned}$$

$$W_m = \frac{0.2778 \text{ metric tonnes per hour}}{V} \text{ or}$$

$$W_m = \frac{\text{kg per second}}{V}, \text{ kg/m}$$

$$kW = \frac{T_e + V}{1000}, \text{ kilowatts}$$

T_e = Effective tension, N

$$\text{Metric Horsepower} = \frac{kW}{0.735}$$

$$\text{US Horsepower} = \frac{kW}{0.7457}$$

The preceding tension formulae should be expressed in newtons, N .

$$T_x = L \times K_x \times K_t$$

$$T_{yc} = 9.807 L \times K_y \times W_b \times K_t$$

$$T_{yr} = 9.807 L \times 0.015 \times W_b \times K_t$$

$$= 0.1471 L \times W_b \times K_t$$

$$T_{ym} = 9.807 L \times K_y \times W_m$$

$$T_m = \pm 9.807 H \times W_m$$

$$T_{am} = Q \times (V - V_0)$$

T_p , T_{ac} —Calculate in pounds of tension as directed in Chapter 6 and convert to newtons, N , of tension. Pound-force, $\text{lbf} \times 4.45 = \text{newtons}, N$.

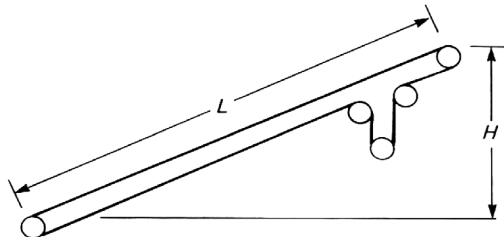
$$T_e = LK_t[K_x + 9.807 W_b(K_y + .015)]$$

$$+ 9.807 W_m(LK_y \pm H) + T_p + T_{am} + T_{ac}$$

A sample problem follows for the purpose of showing the procedure and the comparison of values. In working such a problem, either the T_e formula shown above or the individual formulas can be used. For the purpose of comparison the individual formulas were used in this sample problem. A list of frequently used conversion formulas is given on page 396 along with some comparison values of belt tension and belt velocities.

Problem

Comparing Application of US and SI Units



English or US		Metric or SI
5,760 tph	CAPACITY, Q	1,451.5 kg/sec
1,000 ft	LENGTH, L	304.80 m
240 ft	LIFT, H	73.15 m
30 lbs/ft	BELT, W_b	44.64 kg/m
320 lbs/ft	MATERIAL, W_m	476.21 kg/m
1.5 lbf	A_i	6.67 N
5 ft	S_i	1.52 m
1.2	K_t	1.2
0.018	K_y	0.018
600 fpm	VELOCITY, V	3.048 m/sec
60 in	BELT WIDTH, b	1.52 m

$$K_x = 0.00068(30 + 320) + \frac{1.5}{5}$$

$$K_x = 0.538 \text{ lbs/ft}$$

$$W_m = \frac{33.33 \times 5,760}{600} = 320 \text{ lbs/ft}$$

$$K_x = 0.006669(44.64 + 476.21) + \frac{6.67}{1.52}$$

$$K_x = 7.862 \text{ N/m}$$

$$W_m = \frac{1451.5}{3.048} = 476.21 \text{ kg/m}$$

Tensions in Pounds, lbf

T_x	=	1,000 x 0.538 x 1.2	=	646
T_{yc}	=	1,000 x 0.18 x 30 x 1.2	=	648
T_{yr}	=	1,000 x 0.15 x 30 x 1.2	=	540
T_{ym}	=	1,000 x 0.018 x 320	=	5,760
T_m	=	240 x 320	=	76,800
T_p	=	2 x 200 + 3 x 150	=	850
T_{am}	=	$2.8755 \times 10^{-4} \times 5,760 \times 600$	=	994
T_{ac}	=	3 x 60	=	180
		T_e	=	86,418 lbs

Tensions in Newtons, N

T_x	=	$304.8 \times 7.862 \times 1.2$	=	2,876
T_{yc}	=	$304.8 \times 0.18 \times 44.64 \times 1.2 \times 9.807$	=	2,882
T_{yr}	=	$304.8 \times .1471 \times 44.64 \times 1.2$	=	2,402
T_{ym}	=	$304.8 \times .018 \times 476.21 \times 9.807$	=	25,623
T_m	=	$73.15 \times 476.21 \times 9.807$	=	341,625
T_p	=	$(2 \times 200 + 3 \times 150) \times 4.45$	=	3,783
T_{am}	=	$1,451 \times 3.048$	=	4,423
T_{ac}	=	$3 \times 60 \times 4.45$	=	801
T_e				= 384,415 N

$$\text{US hp} = \frac{86,418 \times 600}{33,000} = 1,571$$

$$kW = \frac{384,415 \times 3.048}{1,000} = 1,172$$

$$\text{Metric hp} = \frac{1,172}{0.735} = 1,594$$

Frequently Used Conversion Formulas

Pound-force, lbf	x	4.4482	=	N
Mass, lbs	x	.4536	=	kg
Length, ft	x	.3048	=	m
Velocity, fpm	x	.0051	=	m/sec
Mass per length, lbs/ft	x	1.4882	=	kg/m
Acceleration, ft/sec ²	x	.3048	=	m/sec ²
Area, ft ²	x	.0929	=	m ²
Volume, ft ³	x	.0283	=	m ³
Horsepower, hp (US)	x	745.7	=	W (Watts)

Comparison of Belt Tensions

N	lbf	N	lbf	N	lbf	N	lbf
1,000	225	10,000	2,248	50,000	11,240	200,000	44,962
2,000	450	15,000	3,372	60,000	13,489	250,000	56,202
3,000	674	20,000	4,496	70,000	15,737	300,000	67,443
4,000	899	25,000	5,620	80,000	17,985	350,000	78,683
5,000	1,124	30,000	6,744	90,000	20,233	400,000	89,924
6,000	1,349	35,000	7,868	100,000	22,481	450,000	101,164
8,000	1,798	40,000	8,992	150,000	33,721	500,000	112,404

Comparison of Belt Velocities

m/sec	fpm	m/sec	fpm	m/sec	fpm	m/sec	fpm
1.00	197	2.00	394	3.00	591	4.50	886
1.25	246	2.25	443	3.25	640	5.00	984
1.50	295	2.50	492	3.50	689	5.50	1,083
1.75	344	2.75	541	4.00	787	6.00	1,181

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Nomenclature

The symbols and meanings listed in Table B.1 are those employed throughout the text for the presentation of data, equations, calculations, and definitions.

For reasons of simplification and clarity, letter symbols used only as linear dimensions in graphic illustrations as well as symbols used only for shafting calculations in Chapter 8 are omitted.

Table B-1. Nomenclature

Symbols	Definition
A	Area, ft^2 , unless otherwise specified
AL	Adjusted idler load, lbs
A_i	Force required to rotate idler, lbs
A_t	Cross sectional area of material on a troughed belt, ft^2
a	Acceleration, ft/sec^2
aa	Skirtboard height above center of belt, in.
a_1	Vertical distance, surface of belt to center of gravity of load at discharge pulley, in.
B_m	Belt modulus of elasticity, lbs/in. of width per ply
b	Width of conveyor belt, in.
C_s	Skirtboard friction factor
C_w	Wrap factor on drive pulley or pulleys
C_{wp}	Wrap factor for the primary drive pulley
C_{ws}	Wrap factor for the secondary drive pulley
c	Edge distance, edge of material to edge of belt, in.
d_m	Apparent density of material, lbs/ft^3
e	Base of Napierian logarithms, 2.718
e_t	Point on the curved belt around a discharge pulley, at which the material starts its trajectory
F	Force, lbs
F_a	Accelerating or decelerating force, lbs
F_d	Braking force at beltline, lbs
F_f	Frequency factor, minutes for belt to traverse one cycle
F_r	Resultant force on idlers at convex vertical curve, lbs
f	Coefficient of friction between pulley and belt surfaces

Table B-1. Nomenclature

Symbol	Definition
fpm	Abbreviation: feet per minute
fps	Abbreviation: feet per second
ft	Abbreviation: feet
ft ²	Abbreviation: square feet, area
ft ³	Abbreviation: cubic feet, volume
ft/sec ²	Abbreviation: feet per second per second, acceleration or deceleration
g	Acceleration due to gravity, 32.2 ft/sec ²
H	Vertical distance that material is lifted or lowered, ft
H_c	Vertical distance from tail pulley to start of concave vertical curve, ft
h	Vertical distance, surface of belt to top of load at discharge pulley, in.
h_s	Depth of material touching skirtboard, in.
hp	Abbreviation: horsepower
Hd	Life to drive pulley (Ref: see pages 124, 125, and 126)
IL	Actual idler load, lbs
in.	Abbreviation: inch
K_t	Ambient temperature correction factor
K_x	Frictional resistance of idlers and sliding resistance between belt and idler rolls, lbs/ft
K_y	Factor for resistance of belt and load to flexure while moving over idlers
K_1	Lump adjustment factor, idlers
K_2	Environmental and maintenance factor, idlers
K_3	Service factor, idlers
K_4	Belt speed correction factor, idlers
L	Length of belt conveyor, centers of terminal pulleys, ft
L_b	Length of one skirtboard, ft
L_c	Length of belt conveyor from tail pulley to start of concave vertical curve, ft
lbs	Abbreviation: pounds
lbs/ft	Abbreviation: pounds per foot of length
lbs/ft ²	Abbreviation: pounds per square foot, pressure
lbs/ft ³	Abbreviation: pounds per cubic foot, density
M	Mass, slugs

Table B-1. Nomenclature

Symbo l	Definition
M_e	Mass equivalent, slugs
NEMA	Acronym: National Electric Manufacturers Association
N_{ri}	Number of return idlers
n	Number of spaces between idlers at convex vertical curve
P	Total force against one skirtboard, lbs
p	Number of plies in belt
P_f	Pulley face width, in.
Q	Quantity of material conveyed, tph (short tons, 2,000 lbs)
R	Resultant radial load, lbs (pulleys)
R_1	Mechanical advantage ratio, takeups
RMA	Acronym: Rubber Manufacturers Association
r	Radial distance center of discharge pulley to center of gravity of load, in.
r_1	Minimum radius of concave vertical curve, ft
r_2	Minimum radius of convex vertical curve, ft
rpm	Abbreviation: revolutions per minute
rpm _b	Abbreviation: revolutions per minute of brake shaft
rpm _p	Abbreviation: revolutions per minute of drive pulley shaft
rpm	Abbreviation: revolutions per second
S_i	Troughing idler spacing, ft
S_{ic}	Maximum troughing idler spacing on convex vertical curve, ft
sec	Abbreviation: seconds
T	Tension to overcome skirtboard friction, lbs
T_a	Tension induced in belt by accelerating forces, lbs
T_{ac}	Total of the tensions from friction of conveyor accessories, lbs
T_{am}	Tension required to accelerate material, lbs
T_b	Tension required to lift or lower the belt, lbs
T_{bc}	Tension required for belt cleaning devices, lbs
T_{1b}	Tension in the tight side of the belt, under the head-end drive pulley, during braking, lbs
T_{2b}	Tension in the slack side of the belt, above the head-end drive pulley, during braking, lbs
T_c	Tension in belt at beginning of a vertical curve, lbs
T_{cx}	Tension in belt at point x on carrying run, lbs

Table B-1. Nomenclature

Symbol	Definition
T_e	Effective belt tension at drive, lbs
T_{eb}	Equivalent braking force, lbs
T_{ep}	Effective tension at primary pulley of a two-pulley drive, lbs
T_{es}	Effective tension at secondary pulley of a two-pulley drive, lbs
T_{fcx}	Tension in belt at point x on carrying run, resulting from friction, lbs
T_{frx}	Tension in belt at point x on return run, resulting from friction, lbs
T_{hp}	Tension at head or discharge pulley, lbs
T_m	Tension needed to lift or lower material, lbs
T_{max}	Maximum tension in belt, lbs
T_{min}	Minimum tension in belt, lbs
T_o	Minimum allowable sag tension in belt for a definite idler spacing, lbs
T_p	Tension from belt flexure around pulleys plus pulley bearing friction, lbs
T_{pl}	Tension from friction of plows, lbs
T_r	Rated belt tension, lbs
T_{rx}	Tension in belt at point x on return run, lbs
T_{sb}	Tension from skirtboard friction, lbs
T_t	Tension in belt at tail pulley, lbs
T_{tr}	Tension from friction of pulleys and flexure of belt on trippers and stackers, lbs
T_{wcx}	Tension in belt at point x on carrying run, resulting from the weight of the belt plus the material carried, lbs
T_{wrx}	Tension in the belt at point x on the return run, resulting from the weight of the empty return belt, lbs
T_x	Tension from friction of carrying and return idlers, lbs
T_{yb}	Tension from belt flexure as belt rides on carrying plus return idlers, lbs
T_{yc}	Tension from belt flexure as belt rides over the carrying idlers, lbs
T_{ym}	Tension from material flexure as material rides on the belt over the carrying idlers, lbs
T_{yr}	Tension from belt flexure as belt rides over the return idlers, lbs
T_1	Tension in the belt at the tight side of the driving pulley, lbs
T_2	Tension in the belt at the slack side of the driving pulley, lbs
T_3	Tension in belt between primary and secondary drive pulleys of a dual pulley drive, lbs
TU	Abbreviation: takeup
t	Time, seconds
t_d	Actual stopping time of a belt conveyor braked or coasting to a stop, seconds
t_m	Maximum permissible stopping time, seconds
tph	Abbreviation: tons per hour (ton = 2000 lbs)

Table B-1. Nomenclature

Symbol	Definition
V	Design belt speed, fpm
V_o	Initial velocity of material fed onto belt, fpm
V_s	Tangential velocity of center of gravity of material discharged over head pulley, fps
W	Weight, lbs
W_b	Weight of belt, lbs/ft of length
W_c	Total weight to be accelerated by the partially loaded belt, at the beginning of a concave vertical curve, lbs
W_e	Equivalent weight of load and moving parts of a conveyor that are accelerated or decelerated by the belts, lbs
W_f	Force to overcome friction of an automatic takeup, carriage, ropes, sheaves, and other frictional resistances, lbs
W_g	Required takeup force, lbs
W_m	Weight of material conveyed, lbs/ft of belt length
W_p	Vertical component of the weight force of an automatic takeup, carriage, wheels, pulley, shaft, and shaft bearings, etc., lbs
WK^2	Moment of inertia of rotating parts, lb-in ²
W_t	Total equivalent weight of all moving parts plus the weight of the full conveyed load that must be accelerated, excluding the drive and drive pulley, lbs
Z_b	Torque rating or setting of brake, lb-ft

Belt Tension to Rotate Pulleys

This alternate method of determining belt tension to rotate pulleys is offered to those desiring a more precise value than Table 6-5 (pg. 98). This alternate method would be especially useful for calculation of smaller, lighter duty conveyors where T_p could be a large percent of conveyor total effective tension (T_e).

Following equations are very similar to those in ISO Standard 5048 First Edition 1979-09-01.

$T_{p(1)}$ = Resistance of belt to flex over pulley, lbs:

$$T_{p(1)} = 9b \left(0.8 + 0.01 \frac{T}{b} \right) \frac{B_t}{D_p} \text{ (for fabric carcass belts)}$$

$$T_{p(1)} = 12b \left(1.142 + 0.01 \frac{T}{b} \right) \frac{B_t}{D_p} \text{ (for steel cable belts)}$$

$T_{p(2)}$ = Resistance of pulley bearings, lubricant, and seals, lbs:

(Not to be calculated for drive pulleys.)

$$T_{p(2)} = 0.01 \frac{d_s}{D_p} R$$

b = width of conveyor belt, in.

B_t = conveyor belt thickness, in.

D_p = pulley diameter, in.

d_s = pulley shaft diameter, in.

R = resultant radial load (vector sum of belt tensions and pulley weight), lbs

T = belt tension at pulley, lbs

Tables C-1 and C-2 provide graphical methods for quick estimate of T_p value. This method should be suitable for most conveyors and can also be used as comparison to Table 6-5.

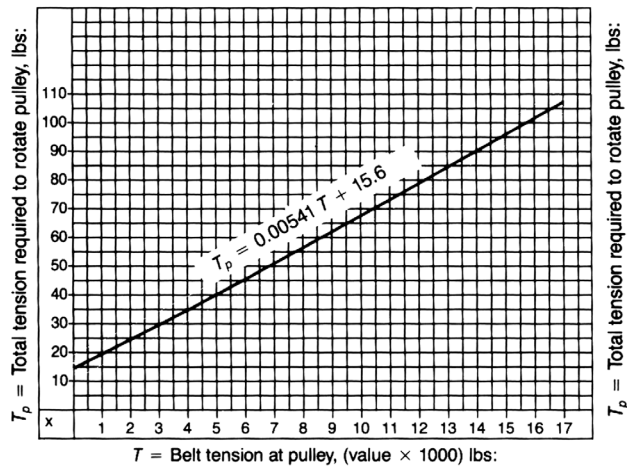


Table C-1. T_p determination — graphical method for fabric carcass belts.

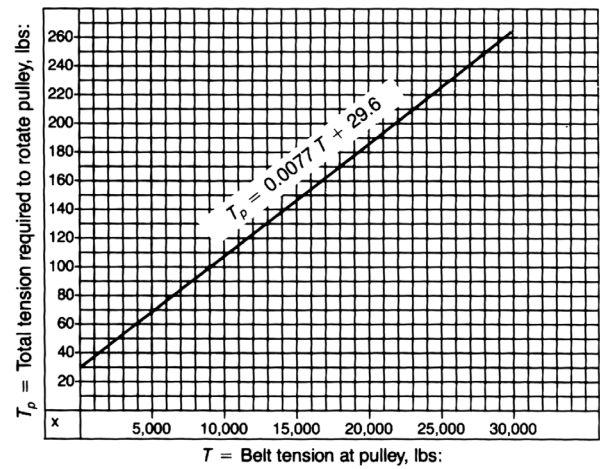


Table C-2. T_p determination — graphical method for steel cable belts.

NOTE: Tables C-1 and C-2 have been derived using belt thickness (B_t) = 1.00 in. If another belt thickness is used, multiply the tables T_p value by the actual belt thickness.

Example:

$T = 10,000$ lbs with $B_t = 0.75$ in (Ref. Table C-2 for Steel Cable Belt)

$$T_p = 107 \text{ lbs} \times \frac{0.75 \text{ in}}{1.00 \text{ in}} = 80 \text{ lbs}$$

Conveyor Installation Standards

For Belt Conveyors Handling Bulk Materials

Introduction

Conveyor stringer alignment

Pulley and shaft alignment

Reducer/motor base installation tolerances

Flexible coupling alignment

Idler alignment

Belt alignment

Skirtboard adjustment

Idler lubrication

Introduction

A trouble-free belt conveyor operation is the product of three properly executed stages of development:

1. Design
2. Manufacturing
3. Installation

followed by an effective maintenance program. Less than satisfactory performance in any of these developmental stages will negatively impact all others, resulting in unanticipated operating problems.

CEMA has already addressed many of the design and maintenance considerations critical to proper operation. It is not our intent to specify minimum levels of manufacturing quality. Indeed, it is the responsibility of each manufacturer to produce a product of which he and the user can be proud.

This document will specify minimum standards for installation of belt conveyors, i.e., acceptable tolerances for structural and mechanical erection in addition to dynamic operating tolerances. In addition, it will provide helpful suggestions that can be utilized to meet or exceed these standards. Each item will be addressed in the sequence in which it is encountered in the field.

NOTES:

- | |
|---|
| <ol style="list-style-type: none">1. It is important that ANSI lockout procedures be followed when making adjustments to bring conveyor machinery into tolerance (ref: ASME B15.1, B20.1, and ANSI Z244.1). |
|---|
2. All mechanical tolerances should be documented by millwrights.

Conveyor Stringer Alignment

Trusses and channel frame conveyor stringers must be installed parallel, straight, and level to allow proper belt training. During installation, dimensional checks shall be made to insure that the following tolerances in the idler carrying chords are not exceeded:*

Parallel

A maximum tolerance of $\pm 1/8"$ shall be allowed for the “back to back” dimension in channel frame or angle stringers. Similarly, $\pm 1/8"$ shall be allowed between webs of I-beams, wide flange beams, or tees when used as truss chords (Figure D. 1).

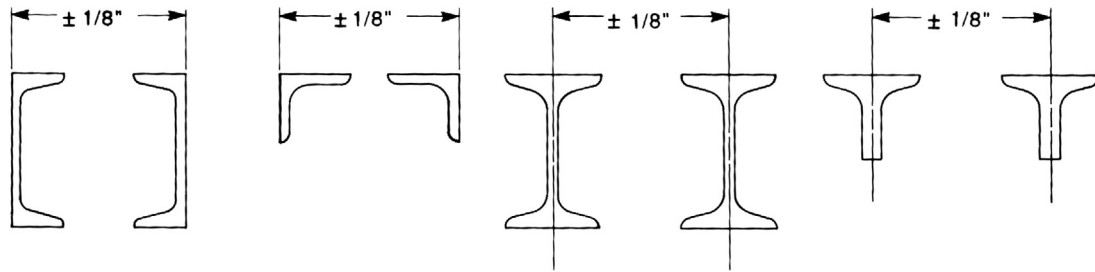


Figure D.1 Maximum tolerances for channel frame or angle stringers.

Straightness

The maximum allowable lateral offset in conveyor stringers shall be $1/8"$ in 40' of length (Figure D.2).

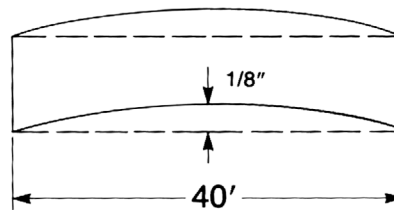


Figure D.2 Maximum allowable lateral offset in conveyor stringers.

* **NOTE:** These tolerances are guidelines for design/manufacture to facilitate proper idler and belt alignment in accordance with the Idler Alignment section of this standard. The overriding issue is idler and belt alignment as opposed to structural alignment.

Squareness

A check on squareness can be made by comparing diagonal measurements between idlers, as shown in Figure D.3.

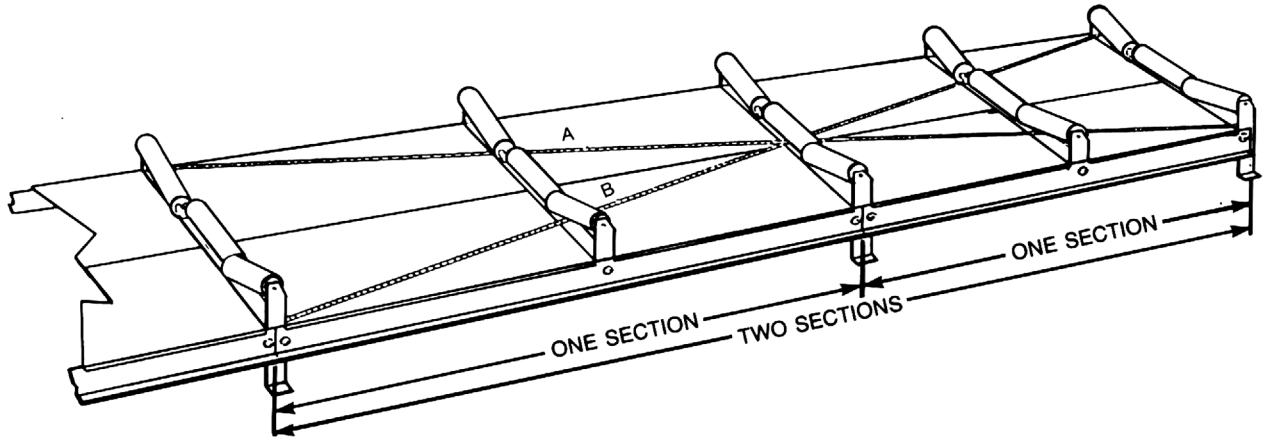


Figure D.3 Conveyor frame and idler alignment.

Lines A and B must be within $\frac{1}{8}$ " to assure conveyor frame squareness.

Return idlers should, likewise, be installed level and parallel to the head and tail pulleys.

Level

The two idler support members shall be level within $\frac{1}{8}$ " regardless of belt width. In addition, the elevation of stringer above support structure shall be held within $\pm\frac{1}{4}$ " (Figure D.4).

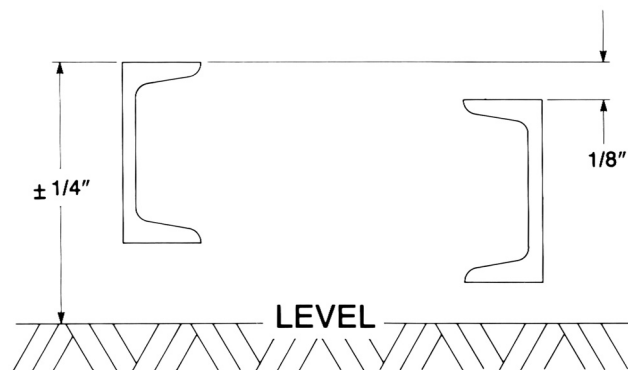


Figure D.4 Level requirements.

NOTE: Designers and fabricators must compensate for mill tolerance of members to satisfy this level requirement.

Pulley and Shaft Alignment

Except for special purpose applications (such as turnover exit), all belt conveyor pulleys should be set level, and the shaft centerline should be perpendicular to the centerline of the belt. Significant departure in alignment will result in unnecessary thrust load on bearings, accelerated and uneven pulley lagging wear, and belt training problems.

It is most convenient to field align pulleys after supporting steel is secured and before belting is installed. After alignment, it is good practice to matchmark bearing housing and supporting steel to enable realignment if disturbed for any reason. Intentional misalignment should not be used as a means to counteract other misalignment forces for training the belt.

Because of pulley manufacturing tolerances, alignment measurements should be taken on the shaft and not on pulley elements. By use of adjustable bearing stops (commonly supplied by the conveyor manufacturer) and full bearing surface shim packs (commonly supplied by the erector), the following alignment tolerances can be reasonably achieved:

Using a level and checking both sides of the pulley, shaft elevations at the bearings should be set within $1/32$ of an inch (Figure D.5).

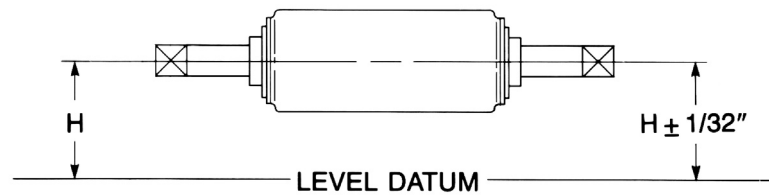


Figure D.5 Shaft elevation settings.

Measuring from a line constructed perpendicular to the conveyor centerline, the shaft centerline should not deviate greater than $\pm 1/32$ of an inch at the bearings (Figure D.6). Because of pulley locations and access thereto, it is common practice to utilize offset lines and plumb bobs to make these measurements.

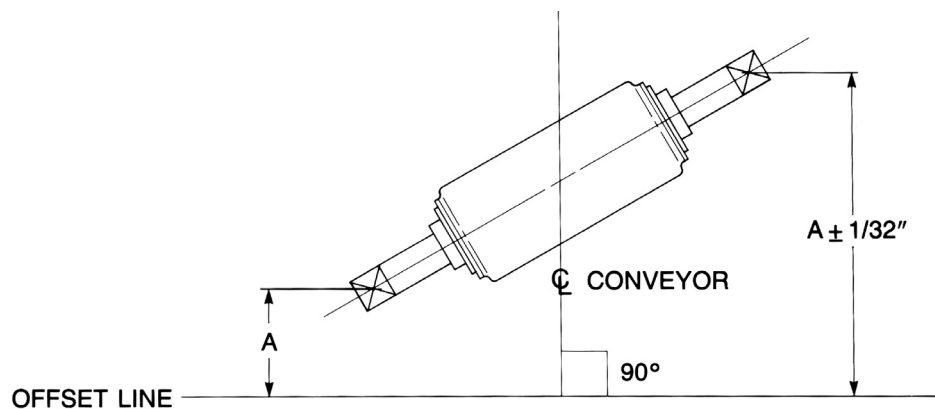


Figure D.6 Shaft elevation settings.

NOTE: See Figure D.3 for squareness check.

When pulleys are mounted on shafts by the equipment supplier, they shall be secured to the shaft in accordance with the recommended practice of the locking element manufacturer. When supplied separately, they should be located on the shaft $\pm\frac{1}{8}$ of an inch of the position shown on the drawings.

If pulleys and shafts are field assembled or repositioned, pulley manufacturer recommendations must be followed to avoid dishing of end discs and residual end disc prestress.

With some locking elements, it may be necessary to check and tighten clamping bolts several times to insure security of hub and shaft assembly. Manufacturers' recommendations should be followed with regard to checks and bolt torques.

Pulleys, shafts, bearings, and supporting steel may be shipped in various degrees of preassembly. Due to manufacturing and erection tolerances, final bearing setting should be done in the field after supporting steel has been secured.

Procedures and tolerances for bearing settings should be obtained from the bearing manufacturer's catalogs. Improper setting of both fixed and expansion type bearings may result in thrust pre-load, causing premature bearing failure.

Reducer/Motor Base Installation Tolerances

Characteristic of all operating mechanical apparatus, correct manufacturing and installation alignment tolerances are necessary to prolong operating life and prevent unnecessary breakdowns. Also, since many reducer manufacturers do not factory oil their reducers, it is imperative that oil levels be checked and maintained by field personnel. Reducer/motor base installation guidelines generally fall into two categories, concrete or structural steel. Either should be sufficiently rigid to minimize vibration and to maintain alignment between the motor and the driven equipment within the manufacturer's recommended tolerances. Base application information may apply to either motors or reducers, or to both in combination.

Fabricated Structural Bases

Structural bases require a surface which, if not flat, can be shimmed to proper position in the structure, using .003 inch to .005 inch shim packs. After the base plate is secured, flatness should be verified and shims added, if necessary.

Concrete Supports

Concrete foundation-type supports are normally the most rigid and secure. However, the installation must be correct within certain limitations to provide acceptable tolerances in the final arrangement with a minimum of shimming.

"Sole plates" can provide a means of adjusting and aligning the motors and reducers or common bedplate to an acceptable tolerance prior to final grouting. Reducer and motor mounting plate surface should be flat to tolerances acceptable to the manufacturer. The bottom of the sole plate needs no tolerance requirement inasmuch as the grout provides a fulling bearing surface.

If the output from the reducer is direct-coupled to a driven unit, the reducer shaft centerline should be held to a minus tolerance only, which would allow shimming by the erector if required. The same is true of the motor shaft in relation to the reducer input shaft.

It is mandatory that alignment checks be made after the installation is complete, but before operation. Misalignment should be no more than 50 percent of the misalignment tolerance allowed by the manufacturer of the coupling being aligned.

Structural Steel Supports

Although combination fabricated structural bases are normally built as a unit, it is not always necessary to provide continuous support in the structural steel below the units; however, it is necessary to provide an integrated and sufficiently rigid connection between reducer base and support steel to accommodate every bolt hole.

The supporting structure must minimize deflection and distortion under load, preventing warping of the motor/reducer base and subsequent unacceptable misalignment.

Flexible Coupling Alignment

Flexible couplings are used in conveyor drive trains to transmit torque from one rotating element to another. In doing so, they protect expensive driving and driven machinery from misalignment, shock loads, vibration, and thrust loads.

It is true that most couplings will operate under severe misalignment for a given period of time; however, the penalty is reduction in useful life. For this reason, manufacturers of couplings have established misalignment tolerances that should be strictly adhered to during field assembly.

A prerequisite to proper alignment is a rigid and level base on which the drive and driven elements are to be set. If they are to be mounted on a steel frame, a common base plate is necessary to prevent movement between independently supported steel members.

Prior to coupling installation, the installer must thoroughly familiarize himself with all of the manufacturer's published requirements of the motor, speed reducer, and couplings. Field breakdown and checking of alignment during assembly to the support structure shall be accomplished using dial indicators, machine test levels, or other means, with only commercial shim stock used for final alignment.

Three types of alignment must be checked: (1) Angular; (2) Parallel; (3) Axial.

Angular Alignment

“Angular alignment” is defined as the movement of the input and output halves of the coupling in such a manner as to permit a rocking and/or sliding action of the element that connects the coupling halves. See Figure D.7.

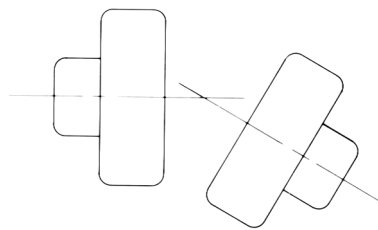


Figure D.7 Angular alignment.

After the coupling halves have been mounted to the driver and driven shafts, the two units should be positioned so that the distance between the coupling faces is equal to the “normal” coupling gap.

The coupling halves are then aligned by positioning a spacer block of thickness equal to the required gap between the faces. The spacer block should check gap at a minimum of 90-degree intervals. Once this is accomplished, the gap should be measured with a feeler gauge.

Parallel Alignment

“Parallel Alignment” is defined as the movement of the input and output halves of the coupling in a way that maintains parallelism between the faces of each coupling half but allows the shafts to occupy separate center lines. The driving and driven equipment should be aligned so that a straight edge can be placed on both coupling flanges at 90-degree intervals and remain parallel to the equipment shafts. Care should be taken when tightening set screws and bolts to insure the proper torque is achieved.

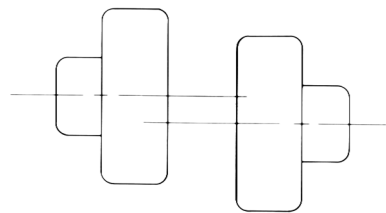


Figure D.8 Parallel alignment.

Axial Alignment

End float for both the driving and driven shafts is sometimes required for expansion and various other reasons. End float on most couplings can be limited to any required distance by the use of limited end float kits; however, these kits should only be used as recommended by the manufacturer.

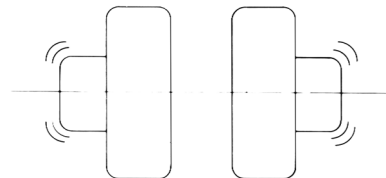


Figure D.9 Axial alignment.

After the couplings have been aligned (cold) and the drive run in to obtain normal operating temperature, it shall be shut down and alignment re-checked using the following procedure:

Break all couplings and re-check “hot” alignment. These readings shall not exceed 75 percent of the manufacturer’s maximum allowable tolerances. If they are not within these limits, the procedure should be repeated as often as needed to achieve necessary “hot” readings.

Idler Alignment

The objective in aligning idlers is to achieve settings which are square and in line to the conveyor centerline and parallel to each other, as shown in Figure D.3.

Idlers shall be set from a previously squared and leveled terminal pulley (preferably the discharge pulley, although this is not essential). A suggested method is to use a tight wire on the conveyor centerline or offset to it, stretched to form a true centerline reference. This line should be at least 100 feet long and referenced to the “squared” starting pulley.

Idlers shall be placed at design spacing and square to the tight wire. After a span of about 50 feet has been filled with idlers, the 100-foot line shall be relocated so that there is a 50-foot overlap on the first position. Repositioning of the tight wire should continue until the entire conveyor length is filled with idlers.

Belt Alignment

Belt training is usually minimized if:

- Conveyor has been installed straight and level with tolerances stated herein
- All pulleys and idlers are square with the conveyor centerline
- Belt splices are correct and square
- There are no defects in the conveyor belting
- All idler rolls turn freely

However, all these conditions are rarely present concurrently, and some training of the belt is usually required.

NOTE: *Belt training should be supervised by one person.*

Training the belt is a process of adjusting idlers and the method of loading in a manner that will correct any tendency of the belt to run off to one side or the other.

When the belt continues to run off consistently over a fixed length of conveyor, the cause is probably in the alignment or leveling of the conveyor structures, idlers, or pulleys in that area.

If one or more segments of the belt run off at all points along the conveyor, the cause is more likely the belt itself, a splice, or the method of loading. When a belt is loaded off center, the lightly loaded edge will rise on the inclined idler roll that it contacts.

All pulleys should be level and at 90 degrees to the centerline of the conveyor. They should be maintained this way and never shifted to train the belt.

Training the belt with the troughing idlers is accomplished by shifting the idler axis with respect to the path of the belt, commonly known as “knocking idlers.”

This method is effective when the entire belt runs to one side along a fixed length of the conveyor. The belt can be centered by “knocking” ahead (in the direction of belt travel) the end of the idler to which the belt runs (See Figure D.10).

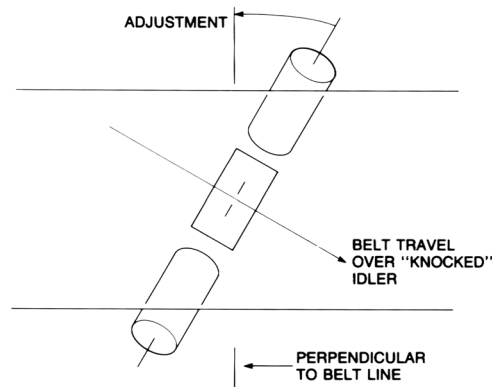


Figure D.10 Correcting belt alignment.

Shifting idlers in this manner should be spread over a length of conveyor preceding the region of trouble. In no event shall any idler be shifted more than $\frac{1}{4}$ inch in any direction from its squared position.

NOTE: Compensation by idler “knocking” may have adverse effects on reversible belts; therefore, avoid “knocking” on reversible belts. Instead, use extreme care in initial alignment.

A belt might be made to run straight with half the idlers “knocked” one way and half the other, but this would increase rolling friction between the belt and idlers. For this reason, all idlers must be initially squared with the path of the belt in accordance with the Idler Alignment section of this standard, and minimum shifting should be used for training. If the belt is over-corrected by shifting idlers, it should be restored by moving the “knocked” idlers, not by shifting additional idlers in the other direction.

Empty Run-in

The conveyor belt should first be aligned (trained) without material load. Most alignment problems, with the exception of off-center loading, will be detected during this empty run-in period.

After ensuring that all elements have been aligned true as stipulated in the Idler Alignment section of this standard, the belt should be slowly “inched” along to provide indication of where corrections of the types described may be required. The first corrections must be those at points where the belt is in danger of being damaged. Once the belt is clear of all danger points, a sequence of training operations can be followed.

The best procedure to use in the training sequence is to start with the return run and work toward the tail pulley. This ensures early centering of the belt on the tail pulley so that it can be centrally loaded. Normally the belt can be trained properly onto the tail pulley by manipulation of the return idlers.

With the empty belt trained satisfactorily, good operation with load is usually assured. Disturbances that appear with load are then usually due to off-center loading or to accumulation of material on snub pulleys and return idlers.

If all corrective measures have been applied, and a particular section of belt continues to run to one side along the entire conveyor length, it can be concluded that the belt may be cambered at that point or may have a splice that is not properly squared. The only corrective measure for a “non-squared” splice is to re-splice. On the other hand, slight camber in a belt may remove itself after an adequate run-in period under load.

Full Load Run-in

A belt may be considered properly aligned if after eight hours of continuous operation under full design load, the belt edges remain within the width of the pulley face. Also, when extra wide face pulleys are used, a further criterion that the belt runs within the confines of the normal troughing, or return, idler should be followed.

***NOTE:** Disc return idlers shall be exempt from the above as the standard roll faces supplied are approximately equal to the belt width. In this case, the belt shall be made to track within 1½ inches of the conveyor centerline. Idler brackets shall not restrict this movement.*

To obtain the above alignment under load, it will be necessary to ensure that the load is centered and that all belt scrapers, plows, and skirt seals exert uniform pressure on the belt. Minor adjustments can be made to these devices to improve tracking.

Carrying and return training idlers should be considered in correcting belt alignment problems before rejecting the belting or splice.

Helpful Hints:

- a. One person should be responsible for training belts. This person should supervise all adjustments.
- b. Periodically check the conveyor to be sure it is level. This requirement is apparent if the conveyor becomes misaligned with no apparent change in loading.
- c. After the belt is run in, an electrician should take readings on voltage, amperes, or wattage. This information can be used for future comparison and a quick trouble check. (Higher readings in the future may indicate excessive drag due to belt misalignment or frozen belt idler rolls.)
- d. Do not overtrain the belt. Overtraining will result in increased belt cover wear and increased power consumption.

Skirtboard Adjustment

Skirtboards are an important element of a well designed belt conveyor. They help centralize and shape the load while minimizing spillage over the belt edges and helping to contain dust.

The steel plates that form the skirting (and any chute lining) should be installed high enough to avoid the risk of contact with the belt, yet low enough to protect the rubber or urethane sealing strips from excessive load pressure and material wear. The cross-sectional area of the chute should be checked carefully to make certain that the skirting and liner do not interfere with flow. It is critical that these steel plates be installed as a smooth, seamless layer that opens in the direction of belt travel, to minimize material entrapment and the risk of belt damage. Typical dimensions for clearance above the belt start at ½ inch above the belt at the tail and open to 1 inch at the exit of the load zone.

Affixed to the bottom of the steel is the skirtboard sealing system, which provides a light, flexible contact with the conveyor belt. This low sealing pressure will minimize friction while retaining all material within its boundaries. Excessive pressure will burn grooves in the belt rubber cover and, therefore, must be avoided. Excessive adjustment or downward pressure of the skirtboard seal can lead to accelerated wear of the sealing strip, and/or belt.

The maintenance of a flat and stable belt line in the loading zone is a key to effective sealing. Efforts taken to avoid fluctuations in the belt line, including closely spaced idlers, belt support cradles, and spiral-wrapping of wing pulleys have generally proven helpful in controlling fugitive material.

Idler Lubrication

Modern conveyor idlers have evolved through better bearings, improved lubricants, and more effective seals. Likewise, the lubrication requirements have changed to accommodate current designs. The following considerations should be reviewed when establishing a lubrication program tailored to a particular installation:

Manufacturer's Recommendation

The manufacturer's data can provide a solid basis for designing the lubrication program.

Type of Lubricant

The proper lubricant will be heavily influenced by operating conditions, the operating environment, and the quality of maintenance program desired. Operating conditions such as operating speed, idler loading, type and size of material being handled, and number of operating hours per year should be considered.

The operating environment is one of the greatest contributors in determining the lubrication program. Temperature, dust, material abrasiveness, washdown techniques, and washdown frequency all affect the lubrication that flushes contaminants from the bearings. Special synthetic lubricants are available for subzero operating temperatures.

Idler Construction

The type and efficiency of the seals directly reflect on the lubrication schedule. Following are several recommendations that will contribute to establishing a successful lubrication program:

A. Remove several idlers from service every six months, disassemble them, and inspect to determine if adequate lubrication has been attained, if migratory contamination and wear have occurred, or if pressure relief plugs are fouled.

If these idlers demonstrate that adequate lubrication has been attained, they could be put back into service and the conveyor operated as normal. If contamination was found in this sampling, all idlers on the conveyor should be lubricated. Also, fouling of pressure relief plugs in sampling should encourage cleaning of all plugs.

B. Thoroughly clean all grease fittings prior to lubricating the idlers to eliminate the possibility of contamination introduced during maintenance.

C. Establish a good maintenance recordkeeping system. Record the date on which samples were taken and the condition of the samples taken and verify trends that may be occurring within the system. Also record idler failures, including location, to establish system trends and effectiveness of the lubrication program.

The establishment of a formal lubrication program as described above will increase the likelihood of many years of successful, trouble-free operation.

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