

SOME EXTRACTS FROM THE BOOK – Engineering Science and Application Design

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Chapter 1: BULK MATERIAL HANDLING PLANTS AND CONVEYORS

1.0.0 NEEDS FOR BULK MATERIAL HANDLING PLANTS WITH CONVEYORS

Modern civilization requires handling of bulk materials in huge quantity. Our present civilization aims at producing innumerable items for utility and comfort of human race. All these items are produced from raw materials from our unique planet with appropriate process / modification.

To cite some examples, we need huge quantity of cement, which is made from mined materials. We need steel to make simple kitchen knife to steamers, which requires movement of huge quantity of iron ore at national as well as global level. Every man needs electric power at his disposal, which demands generation of electric power on mega scale. Hence requirement for movement of huge quantity of coal, from mine to thermal power station.

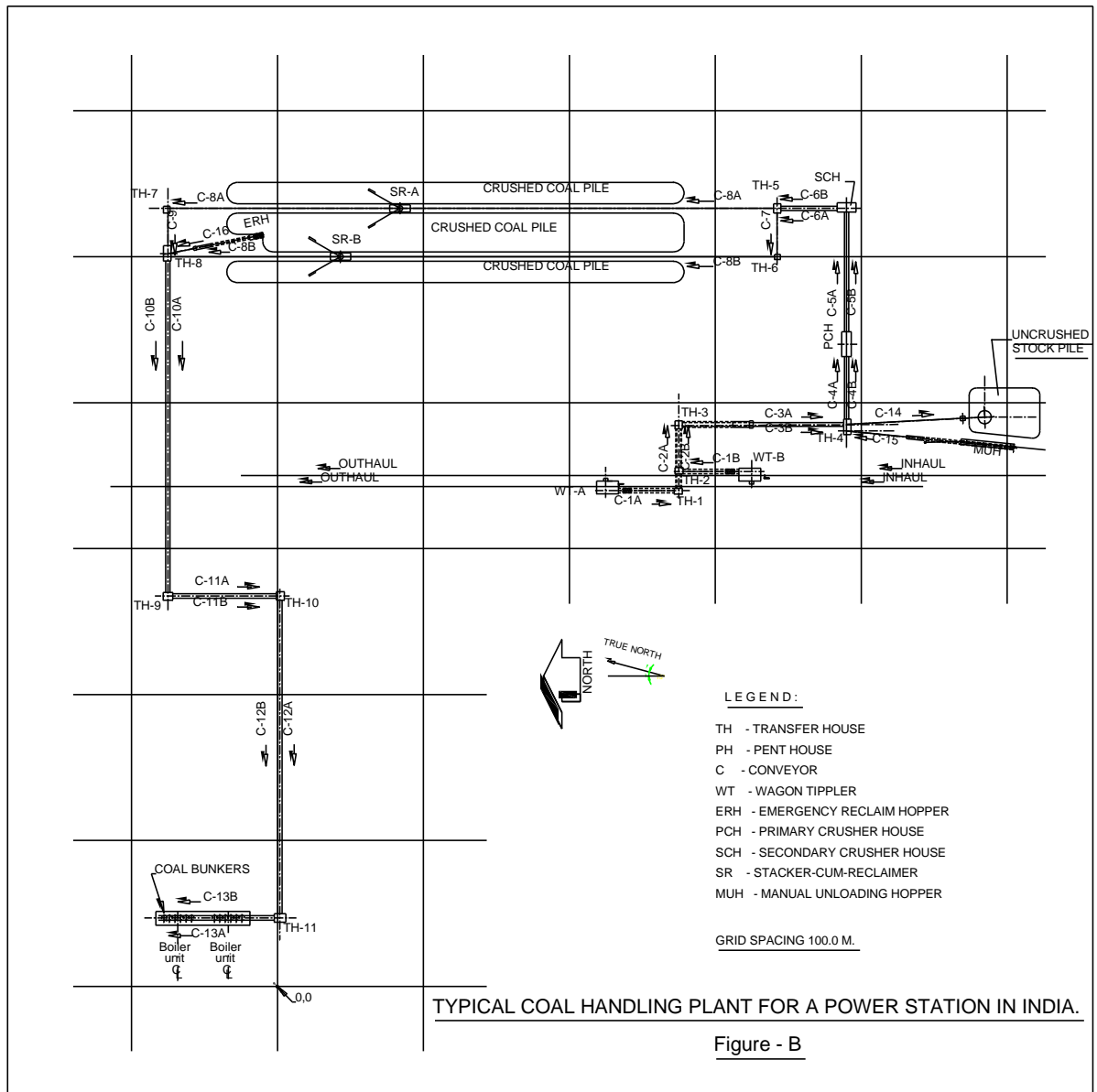
The above examples need installation of special bulk material handling system at source of the raw material, and also at process (consumption) unit. The raw materials being mined has to be temporarily stored in gigantic piles, to be retrieved on need and to be loaded into railway-wagons / trucks. When such materials arrive at cement / steel / power plant, same are to be unloaded, stored, retrieved and finally to be consumed for making end product. Again, if the end product is a new bulk material, it will require further handling for storage, reclamation and onward transport.

Not to forget food grains; often these are produced in one area, but surplus are to be transported to other area / country. This can demand bringing the food-grains to port, temporary storage, retrieval from storage and loading into ships (steamers). The reverse will happen at the port of receipt i.e. unloading from ships, temporary storage at port, retrieval from such storage and dispatch by rail / road. This needs bulk material handling system of sophisticated nature, as quantity involved could be few tens of thousand tonnes in a day.

Above are only few examples. The bulk material handling systems have innumerable areas of application. In the former time, the handling of bulk materials was manual when tonnage involved were very small. The effect of industrialization now reaching practically to every human being; the bulk materials turn over is now counted in millions of tonnes instead of hundred of tonnes in former times. This demands bulk material handling systems for varied applications.

The bulk material handling system consists of numerous equipment, which work in a coordinated preplanned manner to achieve ultimate functional need. The belt conveyors are very prominent and important equipment in such system, to ensure flow of material through various parts of the system.

Figure-B shows a typical bulk material (coal) handling plant in a thermal power station. Following clauses 1.1.1 and 1.1.2 describes the salient features of this typical plant. As mentioned before, most of the plants have certain basic similarities, and therefore, information on this plant will provide a broad outlook about the underlying principles in a bulk material handling system.



1.1.2 List of major equipment and structures for the plant

Main equipment and structures for this typical coal handling plant are as below, which provides a feel about items involved in such plants.

Mechanical:

- | | |
|---|---------|
| 1) Side discharge wagon tippler with integral weigh-bridge. | 2 Nos. |
| 2) Side arm charger with it's rail-track. | 2 Nos. |
| 3) Apron feeder with it's dribble conveyor. | 2 Nos. |
| 4) Belt conveyor including belt, pulleys, idlers, scrapers, drive unit and take-up mechanical. For conveyor Nos. C-1A, C-1B, C-2A/2B, C-3A/3B, C-4A/4B, C-5A/5B, C-6A/6B, C-7, C-8A C-8B, C-9, C-10A/10B, C-11A/11B, C-12A/12B, C-13A/13B, C-14, C-15 and C-16. | 27 Nos. |
| 5) Rotary plow feeder in conjunction with conveyor C-15. | 2 Nos. |
| 6) Primary crusher of rotary breaker type, with drive. | 2 Nos. |
| 7) Secondary crusher of ring granulator type, with drive. | 2 Nos. |
| 8) Vibrating screen. | 2 nos. |

9) Stacker-cum-reclaimer with all accessories and drives.	2 Nos.
10) Belt weigher for conveyors for conveyor Nos. C-1A, C-1B, boom conveyors , C-12A/12B, C-14 and C-15.	8 Nos.
11) Inline magnetic separator for conveyor Nos. C-1A, C-1B, C-15, C-5A/5B, C-11A/11B.	7 Nos.
12) Metal detector for conveyor Nos. C-12A/12B.	2 Nos.
13) Travelling tripper on conveyor Nos. C-13A/13B.	2 Nos.
14) Bunker slot sealing system in conjunction with trippers.	2 Nos.
15) Telescopic chute at discharge end of conveyor No. C-14.	1 No.
16) Rack and pinion gate below emergency reclaim hoppers.	2 Nos.
17) Vibrating feeders below emergency reclaim hoppers.	2 Nos.
18) Motor actuator operated flap gate.	52 Nos.
19) Bunker level indicator (high and low).	24 Nos.
20) Ventilation system for tunnels of C-1A, C-1B, C-2A/2B, C-3A/3B, C-15 and C-16.	6 Nos.
21) Ventilation system for switch gear rooms.	4 Nos.
22) Air conditioning unit for control rooms.	5 Nos.
23) Water spray type dust suppression system for WT-A, WT-B, TH-1, TH-2, TH-4, TH-5, stockpiles, TH-7 and TH-8.	14 Nos.
24) Dry cyclone-cum-bag filter type dust extraction system for TH-3, PCH, SCH, TH-9, TH-10 and TH-11.	6 Nos.
25) Dry cyclone-cum-bag filter type dust extraction system for reducing level of coal dust in bunkers' air.	4 Nos.
26) Water supply system to dust suppression units.	1 No.
27) Water line for plant cleaning.	1 No.
28) Compressed air line for plant cleaning.	1 No.
29) Sump pumps for removal of seepage water in underground structures.	12 Nos.
30) Electric hoist and manual hoist including monorail, for maintenance of equipment (in houses).	65 Nos.
31) Springs and viscous dampers vibration isolation mounting for secondary crushers.	2 Nos.
32) Coal sampling system	1 No.
Electricals and controls: All electricals and controls, which are integral to equipment and from main control room to equipment.	1 lot
Structurals:	
1) Conveyor frame.	1 lot
2) Galleries and trestles for elevated length of conveyors.	1 lot
3) Transfer houses and crusher houses.	1 lot
4) Chute-work at junction of equipment.	1 lot
5) Steel grid on wagon tippler hoppers, manual unloading hoppers, emergency reclaim hoppers and bunker slots.	1 lot
6) Rail-track for mobile machines.	1 lot
7) Cross-over on conveyors.	1 lot
Civil work: For mechanicals, electricals and structurals.	1 lot
Spare parts: For mechanicals and electricals.	1 lot

Chapter 4: BELTS

This chapter has **49 pages**. Following 2 pages write-up is typical.

Example - 2

Design data

Find the lump velocity at chute exit for following application. Also find equivalent free fall height relative to chute exit point and relative to moving belt (Figure-23): Lump mass is 12 kg. Conveyor discharge velocity: 2.5 mps (nearly horizontal). Lump discharge level: +7.5 m,

Lump direct fall and first impact at: +6.0 m

Lump slide down at 55° inclination to level: +4.0 m

Lump direct fall and second impact at: +2.5 m

Lump slide down at 55° inclination to level: +1.5 m

Receiving belt conveyor level: +1.5 m

Receiving conveyor belt velocity: 2.5 mps (Nearly horizontal)

At each impact consider 65% loss of kinetic energy.

Solution

$$\text{Energy at +6.0m level before impact} = \left(\frac{2.5^2}{2 \times 9.81} + 1.5 \right) \times 12 \times 9.81 = (0.318 + 1.5) \times 12 \times 9.81 = 214.08 \text{ Nm}$$

$$\text{Energy after impact at +6.0 m level} = 0.35 \times 214.08 = 74.93 \text{ Nm}$$

$$\text{Energy at +4.0 m level} = 74.93 + 12 \times 9.81 \times 2 \times 0.58 = 211.48 \text{ Nm}$$

$$\text{Energy at +2.5m level before impact} = 211.48 + 12 \times 9.81 \times 1.5 = 388.06 \text{ Nm}$$

$$\text{Energy at +2.5 m level after impact} = 0.35 \times 388.06 = 135.82 \text{ Nm}$$

$$\text{Energy at +1.5 m level} = 135.82 + 12 \times 9.81 \times 1.0 \times 0.58 = 204.1 \text{ Nm}$$

$$\text{Lump velocity at chute exit} = \sqrt{\frac{2 \times 204.1}{12}} = 5.832 \text{ mps}$$

Lump kinetic energy relative to chute / ground = $0.5 \times 12 \times 5.832^2 = 204.07 \text{ Nm} \approx 204.1 \text{ Nm}$ as above
(The difference is due to digital rounding of figures.)

$$\text{Lump free fall height relative to chute exit} = \frac{204.07}{12 \times 9.81} = 1.733 \text{ m}$$

Lump kinetic energy relative to moving belt: Lump velocity perpendicular to belt = $5.832 \sin 55^\circ = 4.777 \text{ mps}$

$$\text{Lump velocity along belt} = 5.832 \cos 55^\circ = 3.34 \text{ mps}$$

$$\text{Lump velocity relative to belt} = \sqrt{4.777^2 + (3.34 - 2.5)^2} = \sqrt{22.75 + 0.7056} = 4.843 \text{ mps}$$

$$\text{Lump velocity inclination relative to perpendicular to belt} = \tan^{-1} \frac{3.34 - 2.50}{4.777} = 10^\circ < 31^\circ = (\tan^{-1} 0.6)$$

Therefore impact energy can be considered as per full relative velocity 4.843 mps.

$$\text{Impact energy} = \frac{1}{2} \times 12 \times 4.843^2 = 140.72 \text{ Nm}$$

$$\text{Lump free fall height relative to moving belt} = \frac{140.72}{12 \times 9.81} = 1.195 \text{ m}$$

The belt carcass should be suitable to withstand lump impact energy 140.72 Nm. If the material is limestone; referring to table -2e, the carcass minimum strength should be EP800/4 or st-1000.

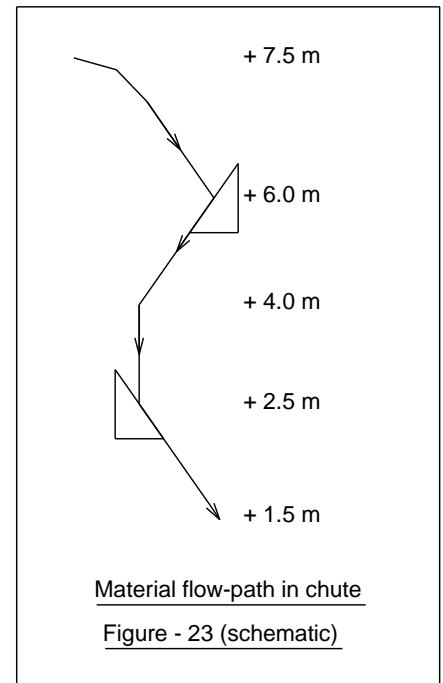


Table 2f : EP / NN conventional multi-ply carcass selection data. Typical Indian make belts (U.K. style). All belt widths are in mm.

Carcass Type	Nominal Carcass Thickness	Nominal Carcass mass	Maximum belt width (mm) for satisfactory load support (Material density in Kg/m ³)					Minimum belt width (mm) for satisfactory troughing		
	in mm	in kg/m ²	up to 375	376 to 675	676 to 1050	1051 to 1550	1551 to 2500	20°	35°	45°
	GD 200/2	2.2	2.5	1000	800	650	500	400	300	300
GD 250/2	2.4	2.6	1000	800	650	600	450	400	450	500
GD 315/3	3.1	3.3	1400	1200	1000	800	650	400	450	500
GD 400/3	3.3	3.6	1400	1200	1050	900	650	500	500	500
GD 500/3	3.6	3.8	1600	1400	1200	900	800	500	500	500
GD 630/3	3.8	4.0	1600	1400	1200	1000	800	500	500	500
MD 400/4	4.5	4.9	1600	1400	1200	1000	800	500	500	600
MD 500/4	4.7	5.2	1800	1500	1200	1050	900	500	500	650
MD 630/4	4.9	5.4	1800	1600	1400	1200	900	500	500	650
MD 800/4	5.5	5.9	2000	1800	1600	1400	1000	500	500	650
MD 1000/4	6.0	6.4	2200	1800	1600	1400	1200	500	650	800
MD 1250/4	6.3	7.0	2400	2000	1800	1600	1400	650	800	900
MD 1400/4	7.1	7.4		2200	2000	1800	1600	650	800	900
MD 1250/5	7.2	7.7		2400	2000	1800	1600	650	800	900
MD 1400/5	7.7	8.4			2200	2000	1800	650	800	1000
HD 315/3	3.7	4.1			1200	1000	800	500	500	500
HD 400/3	4.0	4.5			1200	1050	900	500	500	600
HD 500/3	4.2	4.6			1400	1200	900	500	500	650
HD 630/3	4.6	5.0			1400	1200	1000	500	500	650
HD 630/4	5.9	6.3			1600	1400	1200	500	650	800
HD 800/4	6.4	6.8			1800	1600	1400	650	800	900
HD 1000/4	6.9	7.4			1800	1600	1400	650	800	900
HD 1250/4	7.2	7.9			2000	1800	1600	800	900	1000
HD 1250/5	8.0	8.8			2200	2000	1800	800	1000	1200
HD 1400/5	8.5	9.5			2400	2200	2000	800	1000	1200
HD 1600/5	9.2	9.9				2400	2200	800	1000	1200
HD 1800/5	9.3	10.0					2400	800	1000	1200
HD 2000/5	9.5	10.4						800	1000	1200

Suffix GD (General Duty), MD (Medium Duty) & HD (Heavy Duty) are specific to belt make. Same are not universal designations.

For calculating belt mass, the mass of covers to be added into the mass of carcass. Cover mass is 1.1 Kg/m² for 1.0 mm thickness for M / N grade covers. Cover mass is 1.25 to 1.4 Kg/m² for 1.0 mm thickness for FR grade. For contractual requirement, specific data from belt supplier needs to be used.

* = RMBT means Recommended Maximum Belt Tension (allowable working tension) with respect to steady state running of conveyor. These values are for vulcanised joint - Mechanical fasteners.

** = Not recommended for regular operation.

Selected carcass to be checked for adequacy to absorb lump impact energy (page 40), particularly for wider belt (1600 mm and above) handling large and high density lumps; conversely conveyors dealing with small lumps can have a carcass of lower rating. Belt suppliers consent is essential for contractual application.

Chapter 5: IDLERS

This chapter has **61 pages + 22 pages = 83 pages**. Following 4¹/₂ pages write-up is typical only.

5.2.0 ROLLS (ROLLERS)

The rolls (rollers) are very important components of conveyor, as same are in contact with belt and affect the performance of conveyor to a large extent. The typical construction of roll is shown in following figure-28.

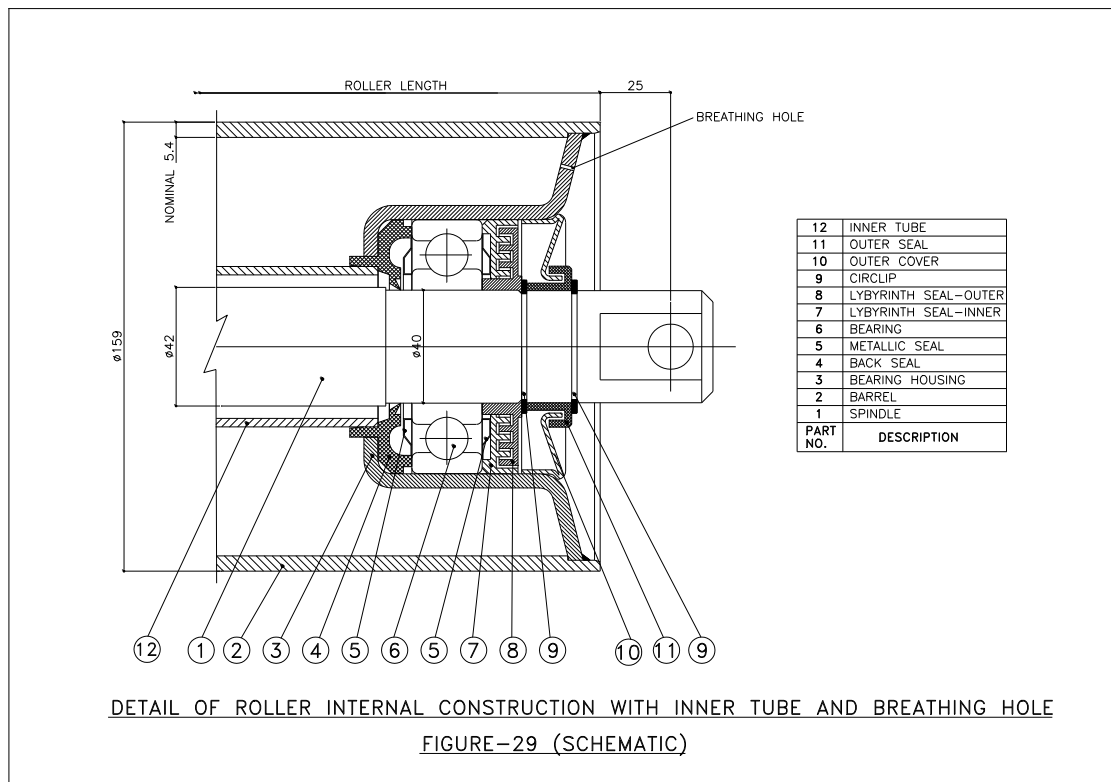
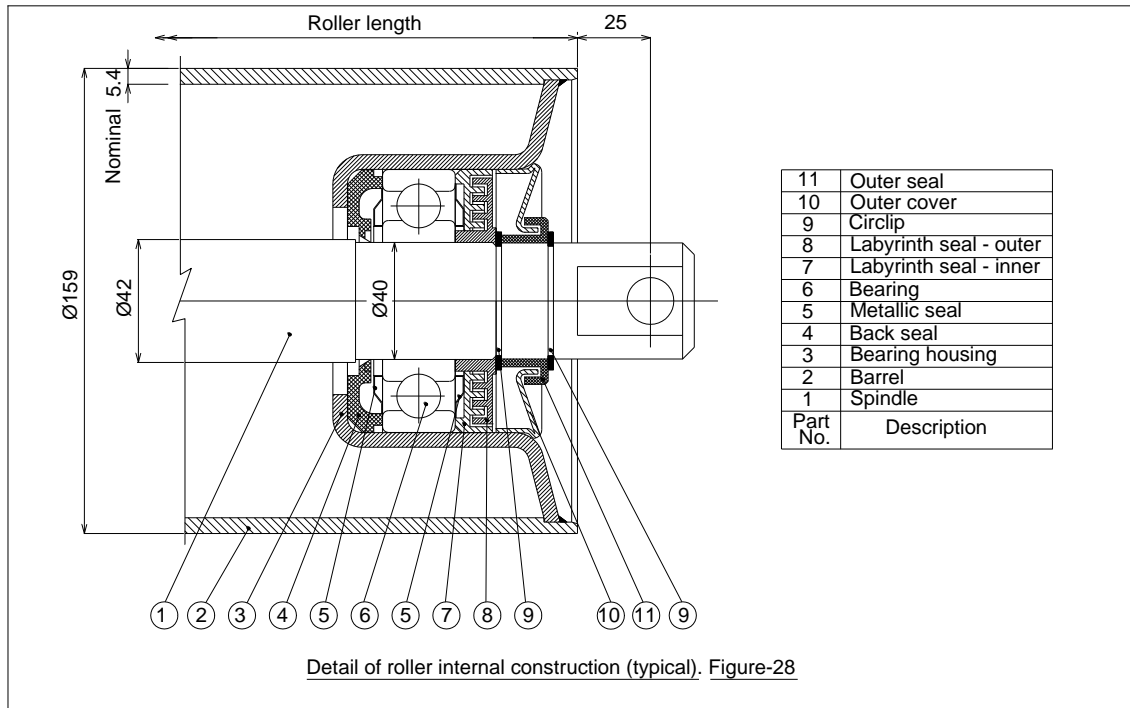


Table - 5: Selection of tube diameter and thickness for roller shell, to suit belt speed

The mentioned data are for average good quality idlers.

Sr. No.	Belt speed mps	Belt width mm	Shell O/D x thickness mm (nominal)	Roller rpm (Approx.)	Comment on thickness Normal / Extra
1	up to 1.60	up to 500	76.1 x 3.65	400	Normal
			76.1 x 4.50		Extra
2	up to 1.85	up to 1000	88.9 x 4.05	400	Normal
			88.9 x 4.85		Extra
3	up to 2.35	up to 1600	108 / 114.3 x 4.50	415 / 393	Normal
			108 / 114.3 x 4.85	415 / 393	Extra
4	up to 3.00	up to 1600	133 / 139.7 x 4.50	430 / 410	Normal
			133 / 139.7 x 4.85	430 / 410	Extra
5	up to 3.60	up to 1600	152.4 / 159 x 4.50	450 / 433	Normal
			152.4 / 159 x 4.85	450 / 433	Extra
		up to 2000	152.4 / 159 x 4.85	450 / 433	Normal
			152.4 / 159 x 5.40	450 / 433	Extra
		up to 2400	152.4 / 159 x 5.40	450 / 433	Normal
			152.4 / 159 x 6.30	450 / 433	Extra
6	up to 4.20	up to 1600	159 / 165.1 x 4.50	500 / 485	Normal
			159 / 165.1 x 4.85	500 / 485	Extra
		up to 1800	159 / 165.1 x 4.85	500 / 485	Normal
			159 / 165.1 x 5.40	500 / 485	Extra
		up to 2400	159 / 165.1 x 5.40	500 / 485	Normal
			159 / 165.1 x 6.30	500 / 485	Extra
7	up to 4.75	up to 1800	165.1 / 168.3 x 4.85	550 / 540	Normal
			165.1 / 168.3 x 5.40	550 / 540	Extra
		up to 2400	165.1 / 168.3 x 5.40	550 / 540	Normal
			165.1 / 168.3 x 6.30	550 / 540	Extra
8	up to 5.60	up to 1800	193.7 x 4.85	550	Normal
			193.7 x 5.40	550	Extra
		up to 2400	193.7 x 5.40	550	Normal
			193.7 x 6.30	550	Extra

Example-6

Data:

Select roller diameter and thickness for 1400 mm belt running at 2.2 mps to convey very abrasive coke. Solution: Sr. No. (3) of table is closest to requirement. 114.3 mm O/D x 4.5 mm thick shell is used for normal materials. However, this being very abrasive material, 114.3 mm O/D x 4.85 mm or 5.4 mm thick shell will be more suitable.

Idler diameter and pitch to suit stress strain in belt bottom cover

Example – 5/3:

Trans Tech publication Germany mentions following information about the ore handling belt conveying system at Europort Rotterdam:

Design handling rate: 5100 mtp.

Conveyor belt: 1400 mm wide, St-1600, covers 10 mm and 5 mm. Belt speed: 2.62 mps.

Troughing idlers: 133 mm o/d, 3 equal rollers, 30 deg trough, spacing 1.25 m.

Return idlers: 133 mm o/d, 2 rollers, 15 deg trough, spacing 2.5 m.

This is the ore handling system at port and working under European ambient condition. Accordingly, following features have been assumed and considered for this example as per routine practice.

Belt cover grade M and durometer hardness 75. Belt mass $M_b = 40.75$ kg/m, for the above stated belt. Material load shared by middle roller $P = 65.6\%$, based on 30 deg trough, 3-equal rollers and surcharge angle approximately 20 degree. Roller length $L = 530$ mm, as per standards. Conveyor design norm sag up to 1%. Find the compressive stress and strain in the belt while passing on the roller and its frequency (cycle/minute).

The above mentioned data is for general information and for purpose of explaining calculation here. Refer concerned parties for specific information.

A conveyor designer is using the aforesaid conveyor features as a reference for new longer conveyor of following specifications.

Conveyor belt: 1400 mm wide, St-2000, covers 8 mm and 6 mm. grade-M, Belt speed: 3.15 mps, belt mass $M_b = 40.6$ kg/m, Troughing idlers: 152.4 mm o/d, 3 equal rollers, 35 deg trough, spacing 1.7 m.

Material load shared by middle roller is 66.9%.

Conveyor capacity is same but bulk density is somewhat less (so belt speed is somewhat more as mentioned).

Conveyor design norm sag up to 1%.

Compare the average strain in belt bottom cover, for reference conveyor and new conveyor, and decide whether new conveyor idler diameter and pitch are satisfactory.

Solution:

Calculation for existing model conveyor

Material load per meter $M_m = (\text{capacity}) \div (3.6 \cdot v) = 5100 \div (3.6 \times 2.62) = 540.712$ kgf.

Material load per idler set = $M_m \times pc \times g = 540.712 \times 1.25 \times 9.81 = 6630.4866$ N

Material load (weight) on middle roller = $6630.4866 \times 65.6 \div 100 = 4349.59923$ N

Belt load (weight) on middle roller = $M_b \cdot pc \cdot L \cdot g \div B = 40.75 \times 1.25 \times 530 \times 9.81 \div 1400 = 189.17096$ N.

Total force by belt on contact surface of middle roller = $4349.59923 + 189.17096 = 4538.7702$ N.

The design solution is by trial and error, with the aid of computer, which has following calculation procedure itself. It shows that the strain in belt will be as per $\theta = 8.4753$ degree. Following manual calculations are for this finally applicable value of θ .

Strain as per geometric depression for assumed ($\theta = 8.4753$ degree):

Roller radius $R = 133 \times 0.5 = 66.5$ mm

$$\text{Average deflection } h_a = \frac{\pi R \theta}{288 \sin \theta} - \frac{R \cos \theta}{1.6} = \frac{3.14 \times 66.5 \times 8.4753}{288 \times \sin 8.4753} - \frac{66.5 \times \cos 8.4753}{1.6}$$

$$= 41.69331108 - 41.10861637 = 0.58469472 \text{ mm.}$$

Strain average geometric = $100 \times 0.58469472 \div (\text{bottom cover thickness } 5) = 11.694$

Strain actual: for above assumed ($\theta = 8.4753$ degree):

Chord length = $2 \cdot R \cdot \sin \theta = 2 \times 66.5 \times \sin 8.4753 = 19.60194383$ mm

Chord length considered, effective as 80% of above = $0.8 \times 19.60194383 = 15.68155506$ mm

Equivalent rubber block length (90% of roller length) = $0.9 \times 530 = 477$ mm

Equivalent rubber block width, chord length mentioned earlier = 15.68155506 mm

Equivalent rubber block depth, $H = 5$ mm.

Rubber block area for stress = $477 \times 15.68155506 = 7480.102 \text{ mm}^2$.

Stress in rubber block = Force \div area = $4538.7702 \div 7480.102 = 0.60678 \text{ N/mm}^2$

Referring to the graph for stress 0.60678 N/mm^2 and for rubber of durometer hardness 55, the percentage strain $U_{55} = 23$

From the table, for rubber of 55 durometer hardness, modulus of elasticity = 2.17715 N/mm^2 .

Also, for the rubber of 75 durometer hardness, modulus of elasticity = 4.1469 N/mm^2 .

Sf = (Rubber block length) \div (Rubber block width) = $477 \div 15.68155506 = 30.418$

Rubber block depth H = 5 mm.

Formula to calculate the actual strain for the applicable rubber / belt is as follows:

$$U_o = \frac{23 \times 2.17715}{4.1469} \times \left\{ \frac{(30.418 \times 5)^{2/3}}{\sqrt{7480.102}} \times 2.9395 \right\} = 11.694$$

$U_o = 11.694$ (this is value of strain as a percentage).

The actual strain 11.694 is same as the geometric strain 11.694. This means the assumed angle of 8.4753 deg is correct and the above result indicates the reality, i.e. model conveyor belt has 11.694 % strain.

For general information, it is not worth to make efforts to match the two values of strain beyond two-three decimals. Strain frequency = $60 \text{ v} \div \text{pc} = 60 \times 2.62 \div 1.25 = 125.76 \text{ cycle/minute}$

Calculation for proposed new conveyor (152.4 mm o/d rollers, 1.7 m pitch):

The calculation checks suitability of using 152.4 mm o/d rollers and 1.7 m pitch for carrying idlers.

Material load per meter Mm = (capacity) \div (3.6. v) = $5100 \div (3.6 \times 3.15) = 449.73545 \text{ kgf}$.

Material load per idler set = Mm \times pc \times g = $449.73545 \times 1.7 \times 9.81 = 7500.2381 \text{ N}$

Material load (weight) on middle roller = $7500.2381 \times 66.9 \div 100 = 5017.6593 \text{ N}$

Belt load (weight) on middle roller = Mb. pc. L. g \div B = $40.6 \times 1.7 \times 530 \times 9.81 \div 1400 = 256.32549 \text{ N}$.

Total force by belt on contact surface of middle roller = $5017.6593 + 256.32549 = 5273.98479$

The design solution is by trial and error, with the aid of computer. It shows that the strain in belt will be as per $\theta = 8.4986$ degree. Following manual calculations are for this finally applicable value of θ .

Strain as per geometric depression for assumed ($\theta = 8.4986$ degree):

Radius R = $152.4 \times 0.5 = 76.20$

$$\begin{aligned} \text{average deflection ha} &= \frac{\pi R \theta}{288 \sin \theta} - \frac{R \cos \theta}{1.6} = \frac{3.14 \times 76.20 \times 8.4986}{288 \times \sin 8.4986} - \frac{76.20 \times \cos 8.4986}{1.6} \\ &= 47.77585249 - 47.10205248 = 0.673800 \text{ mm.} \end{aligned}$$

Strain average geometric = $100 \times 0.673800 \div (\text{bottom cover thickness } 6) = 11.230$

Strain actual for above assumed ($\theta = 8.4986$ degree):

Chord length = $2 \cdot R \cdot \sin \theta = 2 \times 76.20 \times \sin 8.4986 = 22.52247132$.

Chord length considered, effective as 80% of above = $0.8 \times 22.52247132 = 18.01797705$

Equivalent rubber block length (90% of roller length) = $0.9 \times 530 = 477$.

Equivalent rubber block width, chord length mentioned earlier = 18.01797705.

Equivalent rubber block depth, H = 6 mm.

Rubber block area for stress = $477 \times 18.01797705 = 8594.5750 \text{ mm}^2$.

Stress in rubber block = Force \div area = $5273.98479 \div 8594.5750 = 0.61364 \text{ N/mm}^2$.

Referring to the graph for stress 0.61364 N/mm^2 and for rubber of durometer hardness 55, the percentage strain $U_{55} = 23$

From the table, for rubber of 55 durometer hardness, modulus of elasticity = 2.17715 N/ mm².

Also, for the rubber of 75 durometer hardness, modulus of elasticity = 4.1469 N/ mm².

Sf = (Rubber block length) ÷ (Rubber block width) = 477 ÷ 18.01797705 = 26.474

Rubber block depth H = 6 mm

Formula to calculate the actual strain for the applicable rubber / belt is as follows:

$$U_o = \frac{U_{55} \cdot E_{55}}{E} \times \left\{ \frac{(H \cdot Sf)^{2/3}}{\sqrt{A}} \times 2.9395 \right\} \quad U_o = \frac{23 \times 2.17715}{4.1469} \times \left\{ \frac{(6 \times 26.474)^{2/3}}{\sqrt{8594.575}} \times 2.9395 \right\} = 11.230$$

U_o = 11.230 (this is value of occurring strain as a percentage).

The geometric strain is also 11.230. Both are of same value. This means the assumed angle of 8.4986_deg is correct. This means proposed new conveyor belt bottom cover has 11.23% strain.

The strain frequency = 60 × v ÷ pc = 60 × 3.15 ÷ 1.7 = 111.17 cyc/min. For existing conveyor this was 125.76 . Thus, new conveyor has less frequency and it is on safer side.

The existing conveyor has 133 mm diameter rollers, 1.25 m pitch and belt strain is 11.694%

The proposed new conveyor has 152.4 mm diameter rollers, 1.7 m pitch and belt strain 11.230%.

Thus, existing model conveyor and new conveyor belt bottom cover has practically similar values of strain, and new conveyor chosen parameters (idler diameter and pitch) are ok.

Please note that rollers diameter being used are in steps and so it would be inconvenient to match the strain values exactly of two different conveyors (only possible by taking odd value of pitch for the idlers). Also, note that thicker bottom cover tends to moderate the occurring strain (percentage).

Chapter 6: CAPACITY EVALUATION IN SYSTEM

This chapter has **8 pages + 2 pages = 10 pages**. Following 2 pages write-up is typical only.

Example-3 (ship unloading capacity calculation)

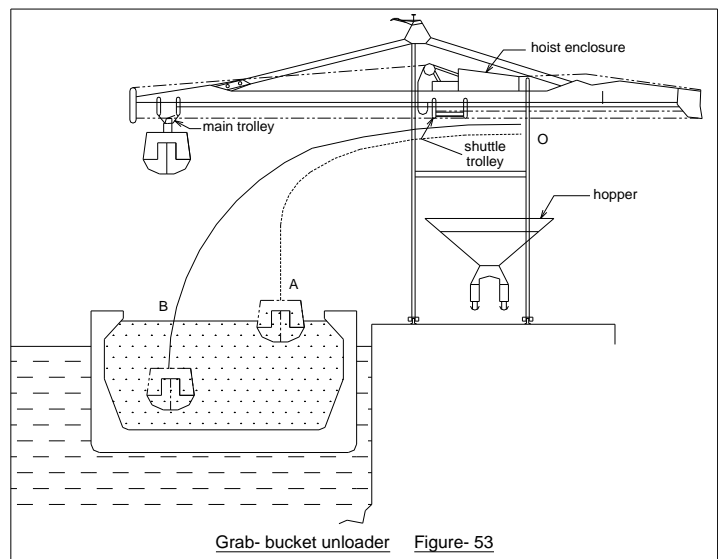
The earlier example input mentions direct relationship between average rate and cream unloading rate; and therefore its solution is accordingly.

The designer also come across the same input data in a more formal and specific terms, as mentioned here under. Then its solution will be accordingly as described here under.

Input data:

The material is arriving at port in a ship of 25000 tonne storage capacity. The ship is to

be unloaded in 35 hrs time. The ship unloading is by grab bucket type unloader. The ship unloader unloading efficiency is 51.8% and cream unloading rate is 15% more than the free digging rate. Consider system non-operational (idle) time of app 2 hrs during unloading of one ship. Feeder below the ship unloader hopper has feed rate fluctuation ±3%. Find out the design capacity for down line belt conveying system.



Solution :

The ship unloader's unloading rate (capacity) particularly for grab bucket type unloader, varies by wide margin during unloading of one hatch (compartment of ship). One unloading cycle of bucket, comprises of travel length and thereby time; starting from grab unloading hopper to actual grabbing point in ship and then back to the grab unloading hopper in ship-unloader. This travel total length and time varies due to following reason.

- The bucket's path length is different if ship is floating at high tide or at low tide, and variation of sea general level specific to the port.
- The bucket's vertical path length is shortest when material is being emptied from upper layer of fully filled compartment, but rest of compartments are empty; and sea level is highest. It will be longest when material is emptied at bottom of one compartment and all other compartments are fully filled and also the sea level is lowest.
- The ship will rise as the material is getting unloaded. So bucket vertical path average length will continue to diminish as the ship is getting unloaded, at particular time.
- The bucket travel horizontal path will be short when emptying the compartment portion closer to jetty side, and it will be longer when emptying compartment portion away from jetty.

The above variants will create confusion for demonstrating guaranteed capacity of ship unloader. The supplier will prefer to do such test during situation favourable to the supplied machine, and buyer will prefer to do it in the worst situation. To avoid such dispute the free digging rate (Test rated capacity) is generally understood as below.

Free digging rate: TransTech publication article by Dravo-USA mentions "Free digging point is generally located vertically at mean low water for the port and horizontally at the centre of vessel to be unloaded."

The meaningful interpretation/elaboration for the above statement while up-holding its purpose/spirit (real intended meaning) could be as below:

- Ship is floating on mean (or average) of low levels. The average water low level for most of the time will depend upon specific installation, and as decided by owner and agreed jointly. Suppose material unloading system is likely to work for 90% output during January to September, then average low level most of the time will have relevance to that period, rather than yearly average. The use of average of low levels is to ensure that machine is capable for guaranteed rate even for the unfavourable condition (ignoring high levels). As against this, all the following points are for average (mid-point) situation during performance test.
- Material is unloaded at mid-height of material in ship hold/hatch.
- The material is unloaded at mid width of hatch (along ship width)

The original statement does not mention whether: A) ship is fully filled (i.e. testing on fully filled ship and unloading from one hatch. Or B) Ship is empty except the last hatch being unloaded. The situation –A will require bucket vertical travel path longer. The situation-B will require bucket vertical travel path shorter. Now considering that the basic purpose is to have average unloading situation during test, we can add/clarify as below.

- Ship as if half filled.

For conducting such test it is not necessary to wait for sea level to be at certain mark, ship is half filled etc., because nobody will have such waiting time. Using the ship characteristic, one can calculate ship hatch level with respect to jetty while complying test conditions, and in that context what would be the depth (dimension) from jetty floor to the layers to be reclaimed for demonstrating test. Then test can be conducted at any time while reaching the particular layer zone during unloading operation of any hatch. In reality such situation is likely to

arise while unloading all the hatches. Total time for 5 or 10 cycles can be used to decide one cycle average time converted to resulting capacity due to material weight in grab bucket by calculation, or measurement by belt weigher on down line conveyor during this test time.

Obviously, this book cannot define testing rules as it has techno-commercial aspects. Refer reputed standards DIN/BS etc. about “Test rated capacity (free digging rate)” testing procedure and rules. Finally, the testing procedure as per contractual agreement between the seller and buyer of machine prevails.

The purpose of this explanatory write-up is only to inform reader about the intricacies of the subject.

Thus free digging rate (test rated capacity) defines the machine size in specific context. The word ‘Test rated capacity’ of machine conveys the meaning correctly, and therefore test rated capacity (TRC) has been mentioned instead of free digging rate.

The unloading efficiency is the ratio of average capacity and test rated capacity (free digging rate) as percentage.

$$\text{Therefore efficiency } \eta = \frac{\text{Average unloading rate } Q_a \text{ for the full ship}}{\text{Test rated capacity } Q_t} \times 100 \quad \therefore Q_t = \frac{Q_a \times 100}{\eta}$$

The cream unloading rate is the peak unloading rate when material is getting unloaded during most favourable situation.

Based on the foregoing information and clarity the solution is as below.

System operational time 36 hrs minus 2 hrs = 34 hrs.

System average unloading rate $Q_a = 25000 \div 34 = 735.3$ mtp

Ship unloader TRC i.e. test rated capacity or Free digging rate = $735.3 \times 100 \div 51.8 = 1419.498$ mtp

Cream digging rate = (cream load factor) \times (machine Test rated capacity i.e. TRC) = $1.15 \times 1419.498 = 1632.4$ mtp. This is the maximum capacity at which the material will be fed into the hopper of ship unloader. The feeder below this hopper feeds to belt conveyor. The feeder discharge rate fluctuation is $\pm 3\%$. Accordingly, maximum feed rate to belt conveyer will be $1632.4 \times 1.03 = 1681$ mtp. Therefore belt conveyor design capacity will be 1681 mtp (+ risk contingency). This is same as the answer in earlier example.

Chapter 7: BELT WIDTH, SPEED AND CAPACITY

This chapter has **66 pages**. Following 2 pages write-up is typical only.

TABLE - 9: Recommended maximum belt speed in m/sec (for inplant conveyors)

Sr.	Materials	Belt width in mm					
		Upto 500	600, 650	750, 800	900, 1000, 1050	1200, 1400, 1600	1800 & above
1.0	Grain and non-abrasive granular material (without dusting)	2.50	3.00	3.50	4.00	4.50	5.00
2.0	Material – X and Y with lumps upto half of maximum size						
2.1	Usual abrasive: Such as coal, lignite, salt, overburden, limestone, etc.	2.00	2.50	3.00	3.50	3.75	4.50
2.2	More abrasive: Such as soft ore, soft stone, gravel, sand, etc.	1.75	2.25	2.50	3.25	3.50	4.00
2.3	More abrasive and sharp edged jagged lumps : Such as hard heavy ore, hard stone	1.50	2.00	2.25	2.75	3.00	3.50
3.0	Material – X and Y with lumps upto maximum size						
3.1	Usual abrasive: Such as coal, lignite, overburden, limestone, etc.	2.00	2.25	2.50	3.00	3.50	4.00
3.2	More abrasive: Such as soft ore, soft stone, gravel, etc.	1.75	2.00	2.25	2.50	3.00	3.50
3.3	More abrasive and sharp edged jagged lumps : Such as hard heavy ore, hard stone	1.50	1.75	2.00	2.25	2.50	3.00
4.0	Very abrasive granular material: Such as certain coke, cullet, etc.	1.75	2.00	2.00	2.25	2.50	2.50
5.0	Material discharge by ploughs						
5.1	Usual abrasive material: Such as coal, salt, grains, etc.	1.00	1.00	1.125	1.125	1.125	---
5.2	More abrasive material : Such as sand	0.75	0.75	0.90	0.90	0.90	---
6.0	Degradable material : Such as urea prills	1.00	1.00	1.25	1.25	1.50	---
7.0	Powdery materials						
7.1	Lighter : Soda ash, pulverised coal	1.00	1.00	1.25	1.25	1.25	---
7.2	Heavier : Cement, flue dust	1.25	1.25	1.5	1.5	1.5	---
8.0	Moisture conditioned : Fly ash, bottom ash	2.00	2.25	2.50	2.50	3.00	---

'Non-abrasive' implies some materials of abrasiveness-A1, as mentioned in table-1, which are similar to grain for very low abrasiveness. Model materials are grains, wheat, rice, etc.

'Usual-abrasive' implies materials of abrasiveness-A1 (excluding grain like material) and materials of abrasiveness-A2, as mentioned in table-1 (chapter - 3). Model materials are coal and limestone.

'More-abrasive' generally implies materials of abrasiveness-A3 as mentioned in table-1 (chapter - 3). Model materials are cement-clinker, sandstone, quartz and fly ash.

'More-abrasive and sharp-edged jagged lumps' generally implies materials of abrasiveness-A4, but it can also include materials of abrasiveness-A3 if it is tough (difficult to break) and has sharp edges. Model materials is broken grey / black granite (hard igneous stone).

Picking conveyor belt speed: 0.25 m/sec to 0.50 m/sec.

The subsequent table-13 pertains to data about material cross section belt. The book includes cross-section important data tables of 19 pages, considering surcharge area shape parabolic as well as circular arc for 3 equal roller troughing idlers, 3-unequal roller troughing idlers, 2-roller carrying idlers and flat belt carrying idlers. Also for 5 roller carrying idlers basis parabolic arc.

Design for belt width, speed and capacity with examples

Material Cross Section Area (With PARABOLIC ARC) for 3 roll idler : Table-13													
A1 = Water fill cross sectional area (which is common for all surcharge angle) in mm ²										P = % of total area resting on central roll			
A3 = Total cross sectional area in mm ²										c = 0,05 B + 25 mm for B <= 2000 mm.			
H = Material height on belt at centre line, mm										c = 125 mm for B > 2000 mm.			
3 EQUAL roll idler							3 UN-EQUAL roll idler						
Troughing Angle / Centre roll length in mm / Roll Gap in mm [(*)=means three unequal roll]													
Belt Width / Sur. Angle	Item	20° / 315 / 10	25° / 315 / 10	30° / 315 / 10	35° / 315 / 10	40° / 315 / 10	45° / 315 / 10	20° / 200 / 20*	25° / 200 / 20*	30° / 200 / 20*	35° / 200 / 20*	40° / 200 / 20*	45° / 200 / 20*
800 / 0	A1	28738	35090	40916	46137	50688	54520	33201	40310	46671	52178	56746	60314
800 / 0	A3	28738	35090	40916	46137	50688	54520	33201	40310	46671	52178	56746	60314
	P	66.72	67.52	68.51	69.70	71.09	72.71	50.99	51.90	53.03	54.41	56.07	58.03
	H	59	73	86	99	111	122	77	95	113	129	145	159
800 / 10	A3	41124	47040	52351	56987	60894	64033	45346	51894	57596	62360	66118	68827
	P	67.35	68.11	69.07	70.25	71.64	73.25	50.55	51.56	52.79	54.26	55.99	58.02
	H	88	101	114	126	137	147	105	123	139	155	170	183
800 / 15	A3	47559	53250	58293	62624	66196	68976	51657	57914	63273	67651	70988	73250
	P	67.55	68.31	69.28	70.46	71.85	73.48	50.40	51.44	52.70	54.19	55.95	58.01
	H	102	116	128	140	150	160	120	137	153	168	182	195
800 / 20	A3	54304	59757	64520	68532	71754	74157	58270	64222	69222	73196	76091	77885
	P	67.71	68.47	69.45	70.64	72.05	73.68	50.28	51.34	52.62	54.14	55.92	58.00
	H	118	131	143	154	165	174	135	152	168	183	196	208
800 / 25	A3	61492	66693	71156	74829	77676	79678	65319	70945	75563	79105	81531	82826
	P	67.84	68.61	69.60	70.80	72.22	73.86	50.17	51.25	52.55	54.09	55.89	58.00
	H	135	147	159	170	180	188	152	168	184	198	210	222
Troughing Angle / Centre roll length in mm / Roll Gap in mm [(*)=means three unequal roll]													
Belt Width / Sur. Angle	Item	20° / 350 / 10	25° / 350 / 10	30° / 350 / 10	35° / 350 / 10	40° / 350 / 10	45° / 350 / 10	20° / 225 / 20*	25° / 225 / 20*	30° / 225 / 20*	35° / 225 / 20*	40° / 225 / 20*	45° / 225 / 20*
900 / 0	A1	37481	45749	53321	60091	65977	70912	42888	52059	60255	67339	73201	77763
900 / 0	A3	37481	45749	53321	60091	65977	70912	42888	52059	60255	67339	73201	77763
	P	65.70	66.51	67.52	68.72	70.15	71.80	50.31	51.22	52.35	53.74	55.40	57.37
	H	68	85	100	115	129	141	88	109	129	148	166	182
900 / 10	A3	53395	61091	67985	73988	79028	83056	58503	66946	74287	80408	85220	88668
	P	66.25	67.02	68.01	69.22	70.64	72.30	49.83	50.85	52.09	53.56	55.29	57.33
	H	101	116	131	145	158	170	120	140	159	177	194	209
900 / 15	A3	61664	69063	75606	81209	85810	89367	66617	74682	81579	87199	91465	94334
	P	66.42	67.20	68.20	69.41	70.84	72.50	49.67	50.72	51.99	53.48	55.25	57.32
	H	118	133	147	161	173	184	137	157	175	192	208	223
900 / 20	A3	70330	77417	83592	88777	92917	95980	75121	82789	89220	94316	98011	100273
	P	66.55	67.35	68.35	69.57	71.02	72.69	49.54	50.61	51.90	53.42	55.21	57.30
	H	135	150	164	177	189	200	154	174	192	208	224	238
900 / 25	A3	79567	86321	92103	96842	100492	103028	84184	91430	97364	101901	104986	106602
	P	66.67	67.47	68.49	69.72	71.18	72.86	49.43	50.52	51.82	53.36	55.17	57.29
	H	154	169	182	195	206	216	173	192	209	225	240	253
Troughing Angle / Centre roll length in mm / Roll Gap in mm [(*)=means three unequal roll]													
Belt Width / Sur. Angle	Item	20° / 380 / 10	25° / 380 / 10	30° / 380 / 10	35° / 380 / 10	40° / 380 / 10	45° / 380 / 10	20° / 250 / 20*	25° / 250 / 20*	30° / 250 / 20*	35° / 250 / 20*	40° / 250 / 20*	45° / 250 / 20*
1000 / 0	A1	47681	58171	67756	76305	83706	89877	53809	65303	75566	84425	91741	97416
1000 / 0	A3	47681	58171	67756	76305	83706	89877	53809	65303	75566	84425	91741	97416
	P	64.34	65.17	66.19	67.43	68.88	70.57	49.77	50.67	51.81	53.20	54.86	56.83
	H	79	97	115	132	148	163	99	123	145	166	186	205
1000 / 10	A3	67550	77305	86022	93585	99903	104913	73331	83908	93094	100741	106736	111010
	P	64.77	65.59	66.61	67.85	69.31	71.02	49.26	50.29	51.53	53.00	54.74	56.78
	H	115	133	150	166	181	194	135	158	179	199	218	235
1000 / 15	A3	77875	87247	95513	102564	108319	112725	83474	93575	102202	109219	114528	118074
	P	64.91	65.73	66.76	68.01	69.49	71.20	49.10	50.15	51.42	52.92	54.69	56.76
	H	134	151	168	183	198	211	154	176	197	216	234	251
1000 / 20	A3	88695	97667	105459	111974	117139	120913	94105	103706	111747	118104	122693	125476
	P	65.02	65.85	66.89	68.15	69.64	71.37	48.96	50.03	51.32	52.85	54.64	56.74
	H	153	171	187	202	215	228	173	195	215	234	251	267
1000 / 25	A3	100227	108772	116060	122003	126539	129639	105434	114504	121920	127573	131396	133366
	P	65.11	65.95	67.01	68.28	69.79	71.53	48.84	49.93	51.24	52.78	54.60	56.72
	H	175	191	207	221	234	246	194	215	235	253	270	284

Chapter 8: MATERIAL BEHAVIOUR ON BELT

This chapter has **64 pages**. Following 7 pages write-up is typical only.

Transverse strain

The bulk material conveying is mostly by trough idlers. In such conveying, the bulk material is retained against belt sides, like a retaining wall. The bulk material exerts transverse (lateral) thrust on belt sides and thereby the side rollers. The belt is flexible and its sides move apart due to material side thrust / pressure, when not on idler proper. The material cross-section width increases as it moves away from idler, but again the width starts reducing as it approaches the next idler. Thus material is undergoing repetitive expansion/contraction in transverse direction as the belt travels. The bulk material positive strain (expansion) is primarily caused by gravity, but its negative strain (contraction) is forced by narrowing profile of belt, against the gravity force acting on material.

The aforesaid occurrence of transverse strain in bulk material causes complex effects in conveying of bulk material by belt. This is as if every idler is receiving oversize (broader) cross-section of bulk material, which it has to roll-down/press-down to idler size cross-section. Thus the rollers of a idler set bears belt and material weight, and in addition, they have to apply continuous forces to down-size the cross-section width of incoming bulk material. Therefore, the numerical sum of loads on rollers is more than the weight of belt plus material. Obviously, the vector sum of all loads acting on all rollers of an idler set is equal to belt weight plus material weight (gravity force).

The transverse strain is co-related to transverse force, which is intercepted by side rollers at effective angle. Therefore, the load on side rollers is made up of belt and material weight plus force related to transverse strain. The transverse strain has very marginal effect on middle roller load.

The fluctuating transverse strain also adds in to material agitation and flexure resistance. The material agitation increases the comparative spreading tendency of material, which is taken care in calculations as a part of reduced repose angle i.e. dynamic repose angle instead of static repose angle (dynamic repose angle accounts for agitation due to longitudinal as well as transverse strain). The transverse flexure of material also appears as a motional resistance to belt, which is taken care as a portion of artificial friction coefficient of conveying. The material longitudinal as well as transverse flexure resistance to belt motion is accounted empirically under single name as material flexure resistance.

The material transverse flexure effect on belt motional resistance can be easily understood by simple logic (conceptual explanation) of friction or hysteresis losses in bulk material, as explained in chapter-10. However one can have a question as to how the transverse force can have drag effect in right angle direction along conveyor length. The more specific explanation is that this transverse forces cause lateral deflection of belt sides. At every idler, the difference in lateral deflection of belt sides at entry and exit on idler, in conjunction with belt tension, creates motional resistance. Therefore, transverse strain and its co-related forces, although being right angle to conveyor length, are producing resistance to belt motion.

The transverse strain affects (adds into) the load on side rollers, idler tilt resistance, material flexure resistance, tractive pull and belt tensions. The belt of adequate stiffness operating under high tension will ensure more stable cross-section of material with less transverse strain and less resistance to belt motion.

The transverse strain / load is calculated considering designated parameters of surcharge angle, etc. However, if the conveyor is handling extreme size lumps with poor size distribution, the aforesaid consideration of parameters may not reflect true situation. If the lump size is too big in relation to small quantity of material, the formation of surcharge angle etc would be erratic. This is likely to cause relatively more resistance to motion, because big lumps with poor size distribution will be more resistant to adjust themselves during course of travel.

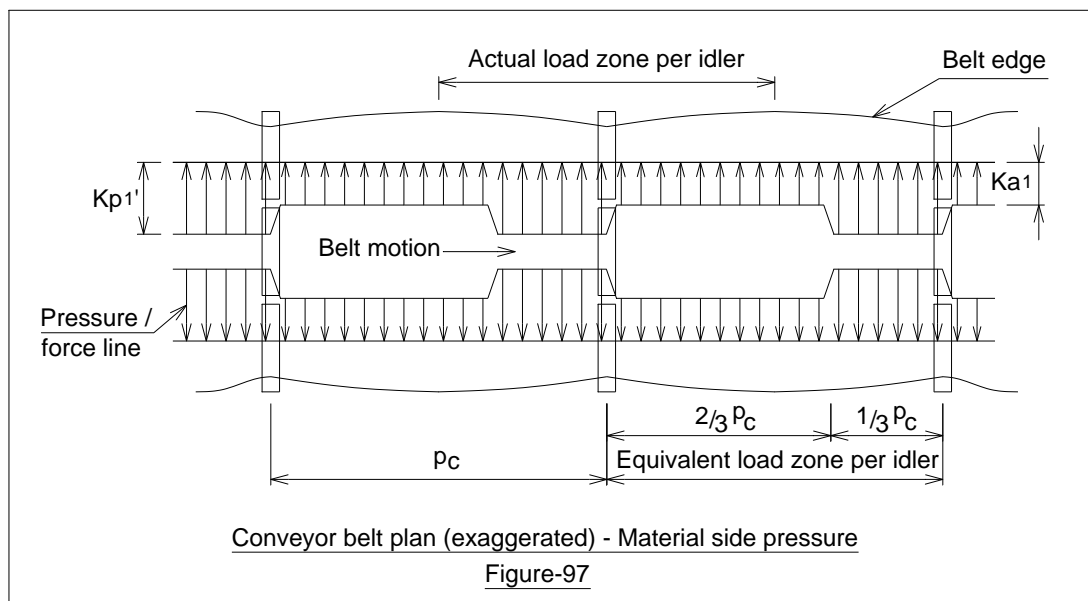
Also, the transverse strain fluctuation is not relevant to flat belt conveyor, because the material is not retained against belt side and hence phenomenon is absent. The material can spread freely, but cannot result into stress-strain situation without opposition.

The theme of this chapter is to understand the implications of \pm transverse strain occurring in bulk material on belt, and various loads arising from the same. Firstly, this chapter provides the comprehensive understanding of many subtle actions taking place in the bulk material, as it is being conveyed and its wide-ranging implications. Secondly, the chapter outlines the procedures to calculate forces on rollers due to transverse strain. These forces are needed in calculation of idler set tilt resistance and thereby the tractive pull. These forces due to transverse strain are also required in design calculation for rollers / idler sets. Thus, this chapter directly provides design procedure for aforesaid needs. The chapter deals with complex subject, however utmost care has been taken for easy explanation, to the extent feasible.

The chapter does not further discuss the longitudinal strain, because it simply acts along conveyor length and is empirically accounted in belt motional resistance, without complex repercussions.

Rankine Theory Application to material on belt:

As stated earlier, the belt / material cross-section transverse dimension tend to expand on leaving the idler and therefore the material side-pressure / force is according to active state, during this travel part. The belt (material) cross-section transverse dimensions starts shrinking as it approaches to idler. This condition is passive state of pressure / force on belt sides, during this travel part. Thus, in one pitch length



of conveyor the material on part length is under active state, whereas material on balance part length is under passive state. The apparent assumption could be to consider active state for half pitch and passive state for balance half pitch, which is also being used in calculation by designers. However, as per trend in Germany, it is preferred to consider active state for 2/3 pitch and passive state for 1/3 pitch. Figure-97 shows the bulk material side pressure on belt, as seen from top. In this calculation model with Rankine formula, K_{p1} value is **assumed** to be reaching to 85%. This consideration for this calculation model provides results close to other methods as would be clear from actual examples.

If Ke_1 is overall pressure factor for pitch length i.e. also for conveyor length; then its value in accordance to figure-97 will be-

$$K_{e1} = \frac{2}{3} K_{a1} + \frac{1}{3} K_{p1}$$

$$= \frac{2}{3} K_{a1} + \frac{1}{3} (0.85 K_{p1})$$

For very free flowing materials which have surcharge angle upto 10° , K_a and K_p values does not vary much due to flat nature of graph. Therefore, $Ke = 1.0$ can be safely considered for

bulk materials having surcharge angle less than 10° . The values of K_{a1} , K_{p1} and Ke_1 for other surcharge angles are as per following table-21. This considers material top face slope angle $\beta = 0$, due to shallow convex curve on central part, which is applicable to Rankine pressure formula.

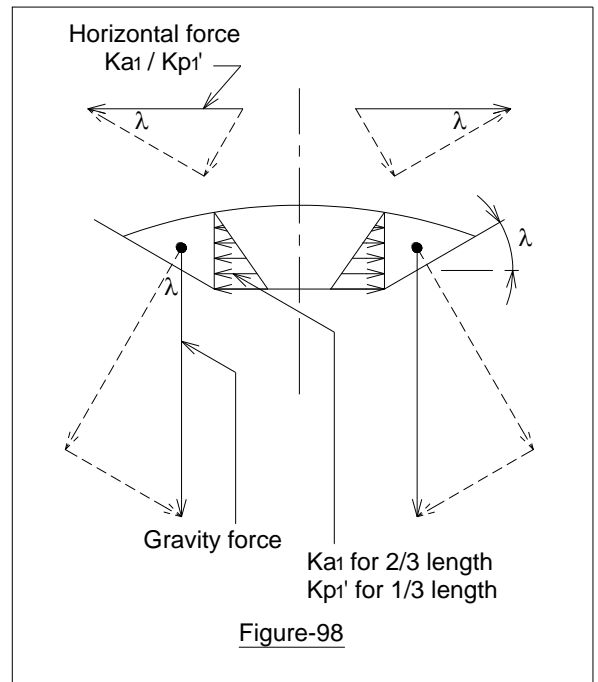
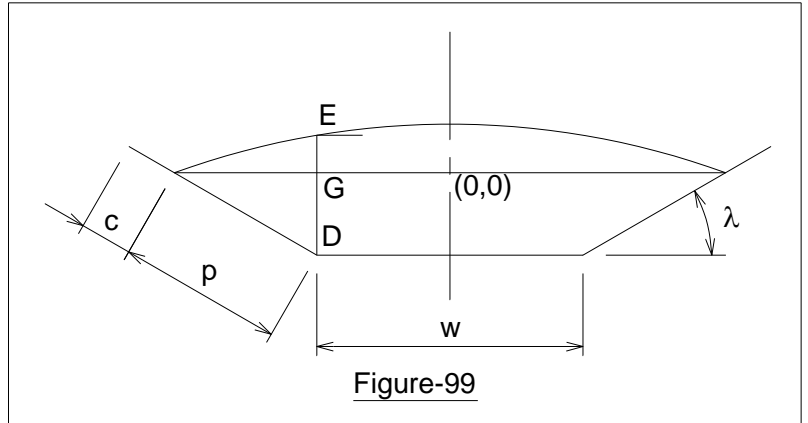


Table - 21			
Surcharge Angle	$K_{a1} =$	$K_{p1} =$	$Ke_1 =$
α	$\frac{1 - \sin \alpha}{1 + \sin \alpha}$	$\frac{1 + \sin \alpha}{1 - \sin \alpha}$	$\frac{2}{3} K_{a1} + \frac{1}{3} (0.85 K_{p1})$
$<10^\circ$	-	-	1.0000
10°	0.704	1.420	0.8716
12.5°	0.6442	1.552	0.8690
15°	0.5887	1.698	0.8730
17.5°	0.5385	1.857	0.8850
20°	0.4900	2.040	0.9040
22.5°	0.4465	2.240	0.9320
25°	0.4058	2.464	0.9695

Material horizontal force on each side:

As explained earlier, the material horizontal thrust is due to simultaneous effect of active pressure in 2/3 length and passive pressure in 1/3 length. The combine effect of active and passive pressure is accounted in pressure factor Ke_1 . Therefore,



$$\begin{aligned} \text{Horizontal side thrust on each side} &= (\text{Ave pressure}) \times (\text{Area}) = \left(\frac{1}{2} Ke_1 \cdot \gamma \cdot g \cdot H\right) \times (H \cdot Lc) \\ &= \frac{1}{2} Ke_1 \cdot \gamma \cdot g \cdot H^2 \cdot Lc \end{aligned}$$

For parabolic arc the value of H to be derived as under (refer figure –99)

$$\begin{aligned} H &= ED = DG + GE = p \sin \lambda + GE \\ &= \frac{1}{2} (\text{Belt width} - \text{central flat belt} - 2 \times \text{Edge clearance}) \sin \lambda + GE \\ &= \frac{1}{2} (B - w - 2c) \sin \lambda + GE \end{aligned}$$

Referring to clause 7.5.2, the parabola equation is as below :

$$y = \frac{L \cdot \tan \alpha}{4} - \frac{\tan \alpha}{L} x^2 \quad \text{where } \alpha \text{ surcharge angle } L \text{ is arc base length}$$

$$\begin{aligned} \text{Now } L &= w + 2p \cos \lambda \\ &= w + 2 \cdot \frac{1}{2} (B - w - 2c) \cos \lambda \\ &= w + (B - w - 2c) \cos \lambda \end{aligned}$$

GE is equal to y when $x = -\frac{1}{2} w$

$$GE = \frac{[w + (B - w - 2c) \cos \lambda] \tan \alpha}{4} - \frac{\tan \alpha}{w + (B - w - 2c) \cos \lambda} \cdot \frac{w^2}{4}$$

$$GE = \frac{[w + (B - w - 2c) \cos \lambda] \tan \alpha}{4} - \frac{w^2 \tan \alpha}{4 [w + (B - w - 2c) \cos \lambda]}$$

$$H = \frac{1}{2} (B - w - 2c) \sin \lambda + \frac{[w + (B - w - 2c) \cos \lambda] \tan \alpha}{4} - \frac{w^2 \tan \alpha}{4 [w + (B - w - 2c) \cos \lambda]} \quad \text{in m}$$

Force on side roller perpendicular to axis.

δ : Conveyor inclination at the location of analysis.

L_c : Conveyor length carrying side at the location of analysis.

M_b : Belt mass per meter, kg / m.

M_m : Material mass per meter of conveying length, kg / m.

B : belt width, m

W : Flat length of belt i.e. central roller length + gap between rollers, m

P : Percentage of cross section area resting on central roller (for example P=66 for 2/3 area)

H : Material height at idler kink, m.

λ : Troughing angle

The force perpendicular to side roller is made-up of following components.

- Belt weight component perpendicular to roller.
- Belt transverse spring effect on side roller (which is completely perpendicular to side roller).
- Material weight component perpendicular to roller.
- Material horizontal thrust component perpendicular to roller.

$$\text{The belt weight vertical} = \frac{1}{2} \left(\frac{B - W}{B} \right) M_b \cdot g \quad \text{N/m}$$

$$\text{The belt weight component in idler plane} = \frac{1}{2} \left(\frac{B - W}{B} \right) M_b \cdot g \cdot \cos \delta \quad \text{N/m}$$

$$\text{The belt weight component, perpendicular to side roller axis} = \frac{1}{2} \left(\frac{B - W}{B} \right) M_b \cdot g \cdot \cos \delta \cdot \cos \lambda \quad \text{N/m}$$

To account for force due to belt transverse spring effect $\cos \lambda$ is ignored in above.

$$\therefore \text{The belt weight and spring effect combined load perpendicular to roller axis} = \frac{1}{2} \left(\frac{B - W}{B} \right) M_b \cdot g \cdot \cos \delta$$

Similarly,

$$\text{Material weight vertical} = \frac{1}{2} \left(\frac{100 - P}{100} \right) M_m \cdot g \quad \text{N/m}$$

$$\text{Material weight component in idler plane} = \frac{1}{2} \left(\frac{100 - P}{100} \right) M_m \cdot g \cdot \cos \delta \quad \text{N/m}$$

$$\text{Material weight component perpendicular to roller axis} = \frac{1}{2} \left(\frac{100 - P}{100} \right) M_m \cdot g \cdot \cos \delta \cdot \cos \lambda \quad \text{N/m}$$

$$\begin{aligned} \text{Component of horizontal force perpendicular to side roller} &= \frac{1}{2} K e_1 \cdot \gamma \cdot (g \cdot \cos \delta) \cdot H^2 \sin \lambda \quad \text{N/m} \\ &= \frac{1}{2} K e_1 \cdot \gamma \cdot g \cdot H^2 \sin \lambda \cdot \cos \delta \quad \text{N/m} \end{aligned}$$

$$\begin{aligned} \text{Total force } F_{sr1} \text{ perpendicular (radial) to roller axis} &= \frac{1}{2} \left(\frac{B - W}{B} \right) M_b \cdot g \cdot \cos \delta \\ &+ \frac{1}{2} \left(\frac{100 - P}{100} \right) M_m \cdot g \cdot \cos \delta \cdot \cos \lambda \\ &+ \frac{1}{2} K e_1 \cdot \gamma \cdot g \cdot H^2 \cdot \cos \delta \cdot \sin \lambda \quad \text{N/m} \end{aligned}$$

$$\begin{aligned} \text{Total force } F_{sx_1} \text{ along roller axis} &= \frac{1}{2} \left(\frac{B - W}{B} \right) M_b \cdot g \cdot \cos \delta \cdot \sin \lambda \\ &+ \frac{1}{2} \left(\frac{100 - P}{100} \right) M_m \cdot g \cdot \cos \delta \cdot \sin \lambda \\ &- \frac{1}{2} K_{e_1} \cdot \gamma \cdot g \cdot H^2 \cdot \cos \delta \cdot \cos \lambda \quad \text{N/m} \end{aligned}$$

Example-1

Find the radial loads on side roller of troughing idler, per metre of conveying route length for following data.

1000mm belt, supported on 30° trough equal rollers, is conveying limestone having bulk density 1300 kg/m³, surcharge angle 20° at 90% utility of cross-section. Belt mass $M_b = 13$ kg/m. Central roller length is 380mm. Conveyor is horizontal.

From above, $\lambda = 30^\circ$, $\alpha = 20^\circ$, $W = 380\text{mm} + 10\text{mm} = 0.390\text{m}$, $B = 1.0\text{m}$, $\delta_b = \alpha = 20^\circ$, $\delta = 0$

Referring to table-13 for material cross section,

$$\begin{aligned} M_m &= [(\text{Material cross section } \text{mm}^2) \div 10^6] \times 0.9 \times 1300 = [(105459) \div 10^6] \times 0.9 \times 1300 \\ &= 123.39 \text{ kg/m} \end{aligned}$$

$P = 66.89\% \cong 67\%$ as per above table, at 100% utility. The proportion will remain nearly same at 90% utility, hence $P = 67\%$.

$C = 75\text{mm}$ at 100% utility of cross section. This value is 94.5 mm at 90% utility as per calculation. One can find out its value by calculation or refer to table 24. Here $C = 94.5\text{mm} = 0.0945\text{m}$

Method-1 (Rankine)

First it is necessary to find material height at idler kink.

$$H = \frac{1}{2} (B - w - 2c) \sin \lambda + \frac{[w + (B - w - 2c) \cos \lambda] \tan \alpha}{4} - \frac{w^2 \tan \alpha}{4 [w + (B - w - 2c) \cos \lambda]} \quad \text{m}$$

$$\begin{aligned} H &= \frac{1}{2} \times (1.0 - 0.39 - 0.189) \times 0.5 + \frac{[0.39 + (1.0 - 0.39 - 0.189) \times 0.8660] \times 0.36397}{4} - \\ &\frac{0.39^2 \times 0.36397}{4 \times [0.39 + (1.0 - 0.39 - 0.189) \times 0.866]} \quad \text{m} \end{aligned}$$

$$H = 0.10525 + 0.0686616 - 0.0183411 = 0.1555705 = 0.15557 \text{ m}$$

$$F_{sr_1} = \frac{1}{2} \left(\frac{B - W}{B} \right) M_b \cdot g \cdot \cos \delta + \frac{1}{2} \left(\frac{100 - P}{100} \right) M_m \cdot g \cdot \cos \delta \cos \lambda + \frac{1}{2} K_{e_1} \cdot \gamma \cdot g \cdot H^2 \cdot \cos \delta \cdot \sin \lambda \quad \text{N/m}$$

$$\begin{aligned} F_{sr_1} &= \frac{1}{2} \times \left(\frac{1 - 0.39}{1} \right) \times 13 \times 9.81 \times 1.0 + \frac{1}{2} \times \left(\frac{100 - 67}{100} \right) \times 123.39 \times 9.81 \times 1.0 \times 0.8660 \\ &+ \frac{1}{2} \times 0.904 \times 1300 \times 9.81 \times 0.15557^2 \times 0.5 \times 1.0 \quad \text{N/m} \quad (K_{e_1} \text{ from table - 21}) \\ &= 38.89 + 172.96 + 69.75 = 281.60 \quad \text{N/m} \end{aligned}$$

Method-2 (Coulomb-Rebhann)

$$K_{a2} = \left[\frac{\sin^2(\lambda + \alpha)}{\sin(\lambda - \delta_b) \cdot \left\{ 1 + \sqrt{\frac{\sin(\alpha + \delta_b) \cdot \sin(\alpha - \beta)}{\sin(\lambda - \delta_b) \cdot \sin(\lambda + \beta)}} \right\}^2} \right]$$

$$K_{a2} = \left[\frac{\sin^2(30 + 20)}{\sin(30 - 20) \times \left\{ 1 + \sqrt{\frac{\sin(20 + 20) \times \sin(20 - 0.42 \times 20)}{\sin(30 - 20) \times \sin(30 + 0.42 \times 20)}} \right\}^2} \right] = \frac{0.58679}{0.1736(1 + 1.0595)^2} = 0.77037$$

This value of 0.77037 is same as in table - 22, which is 0.7702.

$$K_{p2} = \left[\frac{\sin^2(\lambda - \alpha)}{\sin(\lambda + \delta_b) \cdot \left\{ 1 - \sqrt{\frac{\sin(\alpha + \delta_b) \cdot \sin(\alpha + \beta)}{\sin(\lambda + \delta_b) \cdot \sin(\lambda + \beta)}} \right\}^2} \right]$$

$$K_{p2} = \left[\frac{\sin^2(30 - 20)}{\sin(30 + 20) \times \left\{ 1 - \sqrt{\frac{\sin(20 + 20) \times \sin(20 + 0.60 \times 20)}{\sin(30 + 20) \times \sin(30 + 0.6 \times 20)}} \right\}^2} \right] = \frac{0.030145}{0.7660 \times (1 - 0.81520)^2} = 1.152$$

This value of 1.152 is same as in table - 22, which is 1.1524.

$$K_{e2} = \frac{2}{3} K_{a2} + \frac{1}{3} K_{p2} = \frac{2}{3} 0.77037 + \frac{1}{3} 1.152 = 0.8975$$

This value of 0.8975 is same as in table – 22, which is 0.8976.

Now, $p = \frac{1}{2} \times (B - W - 2C) = 0.5 \times (1.0 - 0.39 - 2 \times 0.0945) = 0.2105 \text{ m}$

$$F_{sr2} = \frac{1}{2} \left(\frac{B - W}{B} \right) Mb \cdot g \cdot \cos \delta + \frac{1}{2} \gamma \cdot g \cdot p^2 \cdot K_{e2} \cdot \cos \delta \cdot \cos \delta_b$$

$$= \frac{1}{2} \times \left(\frac{1.0 - 0.39}{1.0} \right) \times 13 \times 9.81 \times 1.0 + \frac{1}{2} \times 1300 \times 9.81 \times 0.2105^2 \times 0.8975 \times 1.0 \times 0.9396$$

$$= 38.89 + 238.26 = 277.15 \quad \text{N/m}$$

Method-3 (Empirical)

$$F_{sr3} = \frac{1}{2} \left(\frac{B - W}{B} \right) \cdot Mb \cdot g \cdot \cos \delta + \frac{1}{2} \left(\frac{100 - P}{100} \right) \cdot Mm \cdot g \cdot \frac{\cos \delta}{\cos \lambda} \quad \text{N/m}$$

$$= \frac{1}{2} \times \left(\frac{1.0 - 0.39}{1.0} \right) \times 13 \times 9.81 \times 1.0 + \frac{1}{2} \times \left(\frac{100 - 67}{100} \right) \times 123.39 \times 9.81 \times \frac{1.0}{0.866}$$

$$= 38.89 + 230.62 = 269.51 \text{ N/m}$$

Method-4 (ISO)

As the carrying idler set has 3-equal roller, the value can be also calculated by following formula.

$$F_{sr4} = \frac{1}{2} C_e (M_b + M_m) g \cos\delta = \frac{1}{2} \times 0.4 \times (13 + 123.39) \times 9.81 \times 1.0 = 267.60 \text{ N/m}$$

ISO do not mention formula for carrying idler set which has 3-unequal rollers. In that case the calculation will be by method-1, 2 and 3 only.

Summary of F_{sr}

- Method-1 (Rankin) $F_{sr1} = 281.60 \text{ N/m}$
- Method-2 (Coulomb) $F_{sr2} = 277.15 \text{ N/m}$
- Method-3 (Empirical) $F_{sr3} = 269.51 \text{ N/m}$
- Method-4 (ISO) $F_{sr4} = 267.60 \text{ N/m}$

Chapter 9: BELT CLEANERS

This chapter has **19 pages**. Following 2 pages write-up is typical only.

3) Double blade with pin joint connection to arm

This topic analysis the forces arising on blade assembly during normal dynamic condition and consequently the necessity for proper location of pin joint. The schematic arrangement is shown in figure-120 (exaggerated).

Pin joint ensures equal (finite) contact force between belt and each blade at design drag force, against construction / installation anomaly. The pin joint also makes the obstacle riding easier by good margin, as obstacle causes partial turning of blade assembly around pins, creating deflection for easy ride.

Dynamic condition

Generally the pin joint location is selected such that both the blades exert equal force during normal dynamic condition. Therefore, $F_1 = F_2 = 0.5F_p$. Both the blades are relatively close in comparison to arm length, and therefore, drag forces by both the blades have been considered to act along common line (tangent to pulley at midpoint of blades) as shown in figure-120. Distance b is also common for both the blades. Analysing the stability of blades' assembly with respect to pin joint, clockwise moment is equal to counter clockwise moment. Therefore,

$$0.5 \cdot \mu_3 \cdot F_p \cdot b + 0.5 \mu_3 \cdot F_p \cdot b + 0.5 \cdot F_p \cdot a_1 = 0.5 \cdot F_p \cdot a_2$$

$$\therefore \mu_3 \cdot F_p \cdot b + 0.5 \cdot F_p \cdot a_1 = 0.5 \cdot F_p \cdot a_2 \quad \therefore \mu_3 \cdot b + 0.5 \cdot a_1 = 0.5 \cdot a_2 \quad \therefore \mu_3 \cdot b + 0.5 \cdot (a - a_2) = 0.5 \cdot a_2$$

$$\therefore \mu_3 \cdot b + 0.5 \cdot a - 0.5 \cdot a_2 = 0.5 \cdot a_2$$

$$\therefore \mu_3 \cdot b + 0.5 \cdot a = a_2$$

$$\therefore \frac{a_2}{a} = 0.5 + \frac{\mu_3 \cdot b}{a}$$

For typical case of $a = 200 \text{ mm}$, $b = 50 \text{ mm}$, and $\mu_3 = 0.6$, the above ratio is as below.

$$\therefore \frac{a_2}{a} = 0.5 + \frac{\mu_3 \cdot b}{a} = 0.5 + \frac{0.6 \times 50}{200} = 0.65$$

This means $a_2 = 0.65 \times 200 = 130 \text{ mm}$ and $a_1 = 70 \text{ mm}$.

Above derivation and equation considers that vertical gravity force line from C.G. of twin blade assembly, is passing through pin joint (or very close to pin joint which is the normal case). Therefore its own turning moment is ignored. Also for the above stage -1 design, location of pin itself is not known and accordingly we cannot include turning moment by the blade assembly.

Now location of pin is known. Twin blade assembly mass 'm₁' and its C.G. location is also known as per construction of twin blade assembly. The b₁ is horizontal distance from pin to vertical line from

C.G. of twin blade assembly which is to be decided by drawing. Then stage-2 final equation is as below.

$$\therefore a_2 = \mu_3 \cdot b + 0.5 \cdot a - (m_1 \cdot g \cdot b_1) \div F_p$$

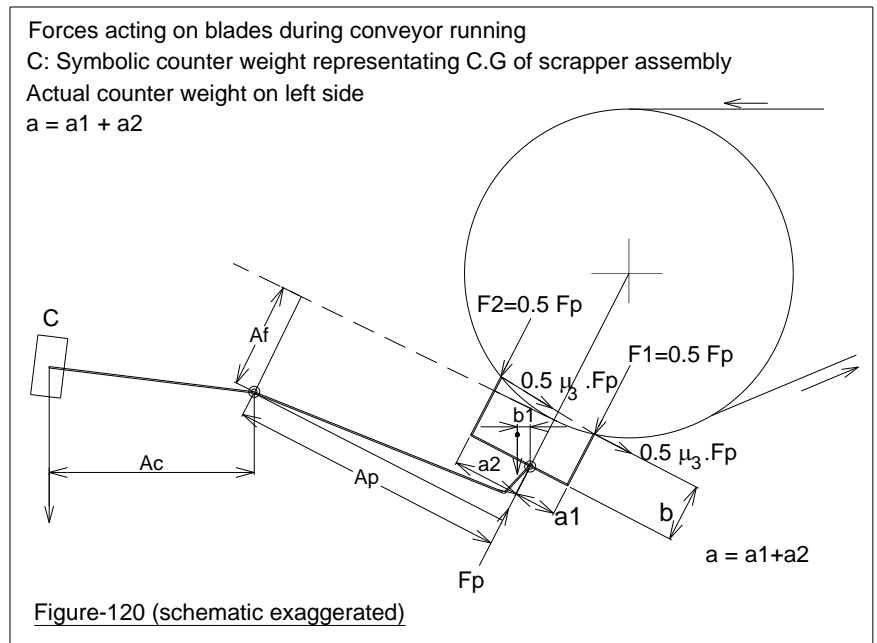
In above equation, b₁ numerical value is positive if b₁ is on left side (away from pulley side) of the pin. Its numerical value is negative if b₁ is on right side (pulley side) i.e. b₁ is between twin blade pin and pulley. Finally use value of a₁ and a₂ as per this stage-2 formula. It is not necessary to make stage-3 calculation. Also refer addendum for some more information and numerical example.

The pin assembly should have stoppers to prevent full rotation of blade assembly in the event of obstacle or abnormal drag force, because in such case above relation does not hold good. Also b should be kept minimum in relation to 'a' for better result and to avoid chattering. The principle of analysis can also be applied for pre-selected proportion of pressing force by each blade i.e. if first blade is thicker and need more pressing force, then the hinge point will shift toward mid point. However this will need maintenance care to fit applicable rubber blade at front or rear!

Static condition

The distribution of pressing force during static condition will be in proportion to distance between pin joint and each blade. The maximum static pressure is (F_p·a₂ ÷ a) for blade which is closer to pin joint. Therefore, ratio of maximum static pressure to dynamic pressure is a₂ ÷ (0.5·a)

For above typical example, this is 130 ÷ (0.5 × 200) = 1.30 i.e. maximum static pressure for one of the blade is 30% more than the dynamic pressure.



Chapter 10: BELT TRACTIVE PULL AND POWER

This chapter has **84 pages**. Following 8 pages write-up is typical only.

Belt flexure resistance

The belt is supported on spaced idlers, and is hanging freely between the idlers. As the belt travels on idlers, it is subjected to two types of flexing actions repetitively viz. flexing of its thickness and flexing of its shape. The flexing of its thickness is denting (or indenting) flexure and flexing of shape is bending flexure (shape can only change by bending / unbending of belt).

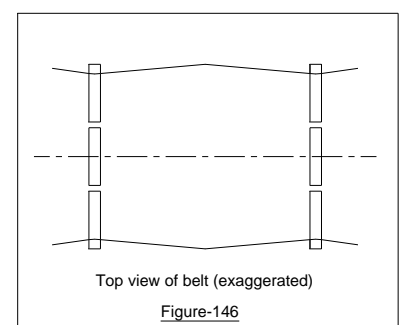
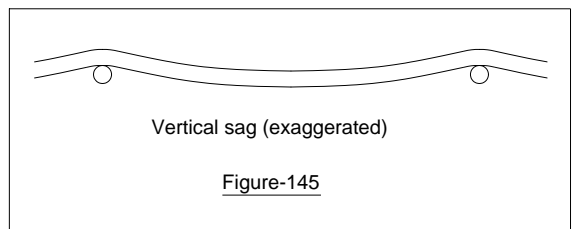
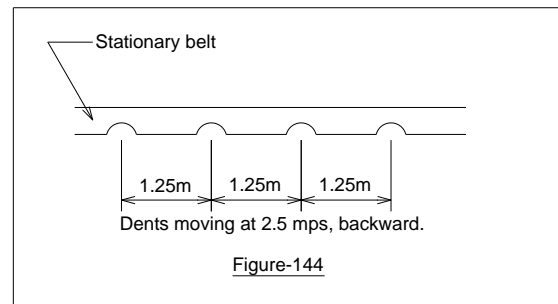
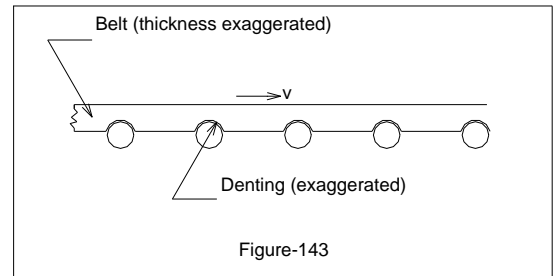
Belt denting is shown in figure-143. The denting in belt thickness occurs because the belt zone on idler is compressed between the weight of belt + material and idler periphery. Any point on belt gets dented when on idler and it gets un-dented while leaving the idler. Suppose a belt is moving at 2.5 mps on idlers spaced at 1.25m pitch, the every point on belt will be subjected to one denting-un-denting cycle every half second; or such two cycles in one second. Alternatively, if we assume the belt as stationary, the idler dent of all the idlers in conveyor would appear to be simultaneously travelling backward at 2.5 mps speed as shown in figure-144. The word dent has been preferred and used instead of indent for this phenomenon, and is to be understood same as indent.

The bending flexure of belt occurs due to sag of belt and slight opening of belt between the idlers, as it travels. The figure-145 shows the vertical sag of belt

due to self-weight and material weight. The belt is subjected to concave and convex bending on longitudinal axis as it travels. In addition the belt is also subjected to bending flexure along its width. Figure-146 shows the slight closing of belt when on idlers, and opening of belt between the idlers, due to material pressure. The belt portion resting on side idlers is changing in shape, which is accompanied by the bending / unbending of its texture (structure). Both the above phenomena cause bending flexure of belt. Such bending flexure will occur twice a second for belt travelling at 2.5 mps speed on idlers located at 1.25m.

All the energy input during denting/bending is not fully recovered during un-denting/unbending (reverse bending), but some energy is lost in these repetitive flexing cycle. This is synonymous to internal friction of belt texture, and opposes the cause, which is belt travel. The basic information about belt denting and bending flexure resistance is as below.

- The rubber is visco-elastic material and thereby it regains the 100% original shape with minute (very small) time delay during

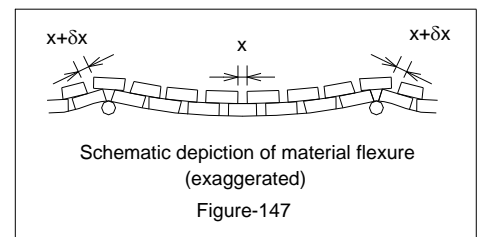


un-denting phase. Also rubber cover under compressive stress bulges; and for moving belt, this bulging is more at belt approach on idler. These result into idler depression (interface) in belt cover slightly non-symmetrical, opposing the belt motion as final consequence for belt denting flexure resistance. The denting flexure coefficient is approximately 0.01 (50% of 0.02) which implies dent asymmetry reaction is at $\tan^{-1}0.01 = 0.573^\circ$ from vertical (only).

- The aforesaid belt denting flexure resistance mainly depends upon the visco-elastic property of belt rubber covers. The superior visco-elastic property implies less denting flexure resistance. However, good performance belt has optimum combination of various required properties viz visco-elastic property, abrasion resistance, tear resistance, oxidization resistance, sensitivity to ambient temperature range, economy etc. Therefore, for extensively used traditional belts of proper quality/make, the visco-elastic property happens to be nearly same, as reflected by commonly accepted basic value $f=0.02$ (obviously with \pm margin as per make.) The improvement in one property often adversely affect some other property/s. However, superior visco-elastic property belts suitable for specific application needs are available, if required by buyer.
- Belt denting flexure resistance is marginally influenced by belt rubber cover hardness and thickness.
- The higher tension in belt keeps it straight with less sag and less variation in shape. Therefore, belt with higher tension has less bending flexure of belt body, and it moves easily on idlers.
- In a given conveyor of specific belt construction and belt tension, the magnitude of belt flexure and thereby its resistance is proportional to material weight + belt weight. More weight of material + belt will cause deeper denting and greater bending, and hence, denting flexure as well as bending flexure resistance will increase numerically with these weights; not coefficients.

Material flexure resistance

The belt supporting the material does not remain straight during its travel from one idler to next idler. As explained earlier, it forms concave shape between the idlers and convex shape directly on the idler. This causes subtle relative oscillating motion of material particles among themselves and also on belt (belt travel imparts slight agitation in to the body of material).



This causes subtle relative oscillating motion of material particles among themselves and also on belt (belt travel imparts slight agitation in to the body of material).

The figure-147 shows schematically the relative rubbing of material particles as the belt travels from sag to hump to sag. This depicts material rubbing in longitudinal direction as seen from conveyor side. This is known as longitudinal flexure of material. Likewise, when seen from top, the belt shape along with material slightly opens between the idlers and slightly closes near the idlers. This causes internal rubbing within material and also on belt in transverse direction. This phenomenon is known as transverse flexure of material.

Thus belt travel causes longitudinal as well as transverse flexure of material resulting in subtle friction between the particles and also friction between belt and particles. The friction opposes the cause, which is belt travel. Therefore, the belt travel encounters the material flexure resistance, which is frictional in nature. The material flexure resistance is influenced by following parameters.

- The higher tension in belt keeps it straight with less sag and less variation in shape. Therefore material flexure resistance is remarkably less when it is conveyed by belt under higher tension. Contrary more effort is needed to move slack belt loaded with material.

- Material particle surface smoothness / roughness and shape effects the resistance. The material of rough surface and or interlocking shapes causes more intense friction and hence more resistance to belt motion. The smooth and rounded lumps / particles have less flexure resistance.
- For a given belt conveyor with specific material and belt tension, the material flexure resistance is proportional to material weight per meter on belt. (The quantity as well as intensity of friction, which is occurring in the body of bulk material is proportional to material weight).

Basis for artificial friction coefficient of conveying 'f'

The foregoing frictional phenomena cause following resistances to belt motion.

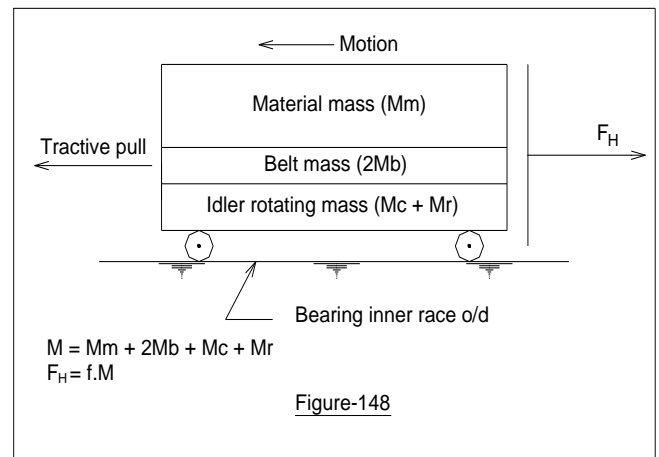
- Idler rotational resistance
- Belt flexure resistance
- Material flexure resistance

The above frictional resistances are directly related to following moving masses.

- Idler rotating mass, per meter length of conveyor.
- Belt mass per meter length of conveyor
- Material mass per meter length of conveyor.

The moving masses relation to earlier said frictional resistances is not one to one, but it is **cumulative** in effect as below.

The idler bearing supports the weight of rotating parts + belt + material resting on belt. Hence, in a broad sense the idler rotating resistance is proportional to sum total weight of all the three masses.



- The magnitude of belt flexure is effected by belt's own weight + material weight resting on a belt. Therefore, belt flexure resistance is proportional to sum total weight of belt mass + material mass.
- The material flexure resistance depends upon material self-weight only, as it is at the top of all the moving mass.

Thus there could be three different friction coefficients applicable to three different values of mass. However, these are calculated as a single entity to avoid complexities. The practice is to use common equivalent artificial / hypothetical friction coefficient 'f' applied to total moving mass force to calculate overall effect named here as main resistance F_H . This is similar to conventional depiction of block (which has weight of material + belt + idler rotating parts) moving on a surface. The figure-148 shows the equivalent model for the same. The complete mass is assumed to be moving on bearing inner race and is multiplied by artificial friction coefficient 'f' to give same frictional resistance as occurring actually in accordance with the field observation. It is to be noted that 'f' is not the idler friction coefficient but it is a conveying frictional coefficient (engineering parameter / multiplier) which when applied in this manner will give total value of F_H in conformity with field results. The above depiction of mass forces shows direct addition. In fact, these are to be added as vector quantities to account for conveyor inclination, which makes slight difference. The main resistance

$$F_H = f \cdot L \cdot g \cdot [M_c + M_r + (M_m + 2M_b) \cdot \cos \delta]$$

Example-4

Select value of f for 1000 mm belt x 3500 m center to center inclined (up) conveyor for conveying (-) 75 mm limestone having bulk density 1350 kg/m^3 . The conveyor design capacity is 1325 mtph and is operating at 3.5 mps speed. The conveyor is equipped with 152.4 mm o.d. x 3 roll x 35° trough carrying idlers. The idlers' pitch is 1.5 m and 3.0 m for carrying and return run respectively. The ambient temperature is 2° C to 45° C . The conveyor location is external between mine and plant. The conveyor is having steel cord belt of cover thickness 8 mm and 5 mm.

Solution

The designer analyses the requirement and makes following decisions.

- Belt speed $2.75 < v = 3.5 \text{ mps} < 3.75 \text{ mps}$. Therefore, the implication is -10% on f .
- The belt width is more than 800 mm. Therefore, no implication for belt width.
- The material being conveyed is limestone, which is somewhat difficult for flexure. Therefore, the implication is $+10\%$ on f .
- Idler diameter and troughing angle do not have any implication.
- The expected maintenance quality is likely to be neither poor nor excellent because maintenance is spread over long distance, and close supervision of this activity may not be practical. Consider increase of half of $+6.5\%$ i.e. $+3.25\%$.
- Conveyor with steel cord belt, and would include winch take up with tension sensors, which will ensure correct tension all the time. The conveyor is to be designed for maximum sag of 1.0% . This being a long distance conveyor, the belt would be under high tension and actual sag would be even less than 1% . Therefore, no implication due to sag.
- The belt bottom cover thickness is 5 mm. Assuming that the rubber hardness will be around 75 shore-A, the implication can be considered as 50% of full implication. Therefore, increase is $+5.5\%$ in basic value of f .
- No implication due to temperature.

The evaluated value of f is

0.02	basic value
- 0.002	-10% for belt speed
+0.002	+10% for flexure
+0.00065	+3.25% for maintenance
+0.0011	+5.5% for thick rubber cover

The total of above is 0.02175. The design can be done with $f = 0.022$.

Example-5

Select value of f for 1800 mm belt x 1500 m center to center inclined (up) conveyor for conveying (-) 300 mm overburden (earthly material) having bulk density 1300 kg/m^3 . The conveyor design capacity is 5750 mtph and is operating at 4.0 mps speed. The conveyor is equipped with 159 mm o.d. x 3 roll x 40° trough carrying idlers. The idlers' pitch is 1.25 m and 3.125 m for carrying and return run respectively. The ambient temperature is 0° C to 35° C . The conveyor location is part of a shiftable conveying system, and is equipped with steel cord belt of cover thickness 8 mm and 5 mm.

Solution

The designer analyses the requirement and makes following decisions.

- Belt speed $3.75 < v = 4.0 \text{ mps} < 5.0 \text{ mps}$. Therefore, the implication is nil.

- The belt width is more than 800 mm. Therefore, no implication for belt width.
- The material being conveyed is overburden, which is average for flexure. Therefore, no implication.
- Troughing angle is 40° , which is more than 35° . This results into 5% increase in basic value of 'f'.
- Idler diameter is more than 108 mm. Therefore, no implication.
- The expected operational / maintenance quality for shiftable conveying system can be considered as average. This amounts to 15% increase in basic value of 'f'.
- Conveyor with steel cord belt, and would include winch take up with tension sensors, which will ensure correct tension all the time. The conveyor is to be designed for maximum sag of 1.0%. This being a long conveyor, the belt would be under high tension and actual sag would be even less than 1%. Therefore, no implication due to sag.
- The belt bottom cover thickness is 5 mm. Assuming that the rubber hardness will be around 75 shore-A, the implication can be considered as 50% of full implication. Therefore, increase is +5.5% in basic value of 'f'.
- No implication due to temperature.

The evaluated value of 'f' is

0.02	basic value
+0.001	+5% for 40° troughing angle
+0.003	+15% for shiftable conveyor
+0.0011	+5.5% for thick rubber cover

The total of above is 0.0251. The design can be done with 'f' = 0.025.

If the factor for shiftable conveyor is considered 12.5% instead of 15%, then 'f' = 0.0246, say 0.0245.

Refer next page ...

10.18.0 Conveyor resistances summary table - 29

Conveyor resistances summary

<u>No.</u>	<u>Resistance name and formula</u>	<u>Applicable to</u>	
		<u>Carrying</u>	<u>Return</u>
1	Slope resistance : Carrying run, F_{slc} $F_{slc} = Mb.H.g + Mm.Hm.g$ (if H and Hm are different) $F_{slc} = (Mb + Mm).H.g$ (if H = Hm)	√	X
1A	Slope resistance : Return run, F_{slr} $F_{slr} = -Mb.H.g$	X	√
2	Material acceleration resistance, Fa $Fa = l . \gamma . (v - v_0)$	√	X
3	Main resistance : Carrying run, F_{HC} $F_{HC} = f.L.g.[Mc + (Mb + Mm). \cos\delta]$ (exact) $F_{HC} = f.g.[Mc.L + (Mb + Mm). L . \cos\delta]$ (exact) $F_{HC} = f.g.[Mc.L + (Mb + Mm). Lh]$ (exact) Now L is nearly equal to Lh for $\delta < 18^\circ$, in that case formula is F_{HC} = f.g.Lh.[Mc + Mb + Mm] Standard accurate formula for $\delta < 18^\circ$	√	X
3A	Main resistance : Return run, F_{HR} $F_{HR} = f.L.g.[Mr + Mb. \cos\delta]$ (exact) $F_{HR} = f.g.[Mr.L + Mb. L. \cos\delta]$ (exact) $F_{HR} = f.g.[Mr.L + Mb. Lh]$ (exact) Now L is nearly equal to Lh for $\delta < 18^\circ$, in that case formula is F_{HR} = f.g.Lh.[Mr + Mb] Standard accurate formula for $\delta < 18^\circ$	X	√
4	Skirt board resistance-Fsk1 : Acceleration zone $F_{sk1} = \frac{\mu_2 \cdot (l)^2 \cdot \gamma \cdot g \cdot La}{\left(\frac{v_0 + v}{2}\right)^2 \cdot b_1^2} + 80 \cdot La$ $\text{Where, } L_a = \frac{v^2 - v_0^2}{2 \cdot g \cdot f' \cdot \mu_1 \cdot \cos \delta - 2 \cdot g \cdot \sin \delta - \frac{4 \cdot \mu_2 \cdot g \cdot f' (l)}{(v_0 + v) \cdot b_1^2}}$ $\text{Where, } f' = \frac{g \cdot b_1 + 0.3 \cdot v_0 \cdot (v_0 + v)}{g \cdot b_1}$ For vertical feed, the formula for La and f' are different. Refer clause 10.8.1.	√	X

<u>No.</u>	<u>Resistance name and formula</u>	<u>Applicable to</u>	
		<u>Carrying</u>	<u>Return</u>
4A	Skirt board resistance-Fsk2 : Beyond acceleration zone $F_{sk2} = \mu_2 \cdot C_{rank} \cdot \frac{(l)^2 \cdot \gamma \cdot g \cdot (L_{sk} - L_a)}{v^2 \cdot b_1^2} + 80 \cdot (L_{sk} - L_a)$ Value of La is to be taken from above Sr. No. 4	√	X
5	Pulley turning resistance, Ft (all pulleys, excluding drive pulley) $F_t = 0.005 \cdot F_{rx} \cdot \frac{dx}{Dx} \cdot (\text{this pulley quantity}) + 0.005 \cdot F_{ry} \cdot \frac{dy}{Dy} \cdot (\text{this pulley quantity}) + 0.005 \cdot F_{rz} \cdot \frac{dz}{Dz} \cdot (\text{this pulley quantity})$	Only of tripper	All others
6	Pulley wrap resistance, Fw (Following formula not applicable to FPS units. Refer cl. 10.16.0) For fabric belt $F_w = 9 B \left(140 + 0.01 \frac{F_{bd1}}{B} \right) \frac{t}{D1} + 9 B \left(140 + 0.01 \frac{F_{bx}}{B} \right) \frac{t \cdot N_x}{Dx} + 9 B \left(140 + 0.01 \frac{F_{by}}{B} \right) \frac{t \cdot N_y}{Dy} + 9 B \left(140 + 0.01 \frac{F_{bz}}{B} \right) \frac{t \cdot N_z}{Dz}$ For steel cord belt $F_w = 12 B \left(200 + 0.01 \frac{F_{bd1}}{B} \right) \frac{t}{D1} + 12 B \left(200 + 0.01 \frac{F_{bx}}{B} \right) \frac{t \cdot N_x}{Dx} + 12 B \left(200 + 0.01 \frac{F_{by}}{B} \right) \frac{t \cdot N_y}{Dy} + 12 B \left(200 + 0.01 \frac{F_{bz}}{B} \right) \frac{t \cdot N_z}{Dz}$	Only of tripper	All others
7	Idler tilt resistance : For 3 roll trough carrying run, Fitc $Fitc \text{ (Rankine)} = \mu_{oc} \cdot \left[\left(\frac{B-W}{B} \right) M_b \cdot g \cdot \cos \delta + \left(\frac{100-P}{100} \right) M_m \cdot g \cdot \cos \delta \cdot \cos \lambda_c + K_{e1} \cdot \gamma \cdot g \cdot H^2 \cdot \cos \delta \cdot \sin \lambda \right] \cdot L_{itc} \cdot \sin \psi_c$ $Fitc \text{ (Coulomb - Rebhann)} = \mu_{oc} \cdot \left[\left(\frac{B-W}{B} \right) M_b \cdot g \cdot \cos \delta + \gamma \cdot g \cdot p^2 \cdot K_{e2} \cdot \cos \delta \cdot \cos \delta_b \right] \cdot L_{itc} \cdot \sin \psi_c$ In above expression δ_b = material surcharge angle α $Fitc \text{ (Empirical)} = \mu_{oc} \left[\left(\frac{B-W}{B} \right) \cdot M_b \cdot g \cdot \cos \delta + \left(\frac{100-P}{100} \right) \cdot M_m \cdot g \cdot \frac{\cos \delta}{\cos \lambda_c} \right] L_{itc} \cdot \sin \psi_c \quad (\text{For } \lambda < 40^\circ)$ $Fitc \text{ (Empirical)} = \mu_{oc} \left[\left(\frac{B-W}{B} \right) \cdot M_b \cdot g \cdot \cos \delta + \left(\frac{100-P}{100} \right) \cdot (1.1 M_m) \cdot g \cdot \frac{\cos \delta}{\cos \lambda_c} \right] L_{itc} \cdot \sin \psi_c \quad (\text{For } \lambda = 40^\circ \text{ or } 45^\circ)$ $Fitc \text{ (ISO)} = \mu_{oc} \cdot C_e \cdot (M_b + M_m) \cdot g \cdot \cos \delta \cdot L_{itc} \cdot \sin \psi_c$	√	X
7A	Idler tilt resistance : For 2 roll trough return run, Fitr $F_{itr} = \mu_{or} \cdot L_{itr} \cdot M_b \cdot g \cdot \cos \lambda_r \cdot \cos \delta \cdot \sin \psi_r$	X	√

<u>No.</u>	<u>Resistance name and formula</u>	<u>Applicable to</u>	
		<u>Carrying</u>	<u>Return</u>
8	Belt external cleaner resistance, Fce Fce = $\mu_3 \times A_{ce} \times P_{pe} \times n_{ce}$ (Applicable scraper type cleaner) For rotary nylon brush and rotary rubber blade cleaner, refer cl. 10.10.0 and cl. 10.11.0	Only of tripper	All others
8A	Belt internal scraper resistance, Fci Fci = $\mu_3 \times A_{ci} \times P_{pi} \times n_{ci}$	X	√
9	Pulley cleaner resistance, Fcp Fcp = 250 to 750 n/m	Only of tripper	All others
10	Discharge plough resistance Resistance = $F_1 (\mu_r + \mu_s) + F_2 (\mu_m + \mu_s) + F_3 \cdot \mu_s + F_4 \cdot \mu_s$	√	X
11	Resistance due to related machines The resistance due to plough feeder, travelling tripper, reclaimers, fixed tripper, etc, already accounted in above Sr. No. 1 to 9 as applicable.	---	---
12	Return belt inversion (Turn Over) resistance	X	√
13	Belt propelled tripper travel resistance Tripper resistance with respect to conveyor belt = $\frac{V_t}{V} \cdot m \cdot g \cdot (0.08)$	√	X
14	Total of Sr. No. 1 to 13	√	√
15.	Total tractive pull for belt, P (total of Sr. No. 14, Carrying + Return) =		N
16.	Belt pull against non-frictional resistance in conveyor, total of Sr. No. 1, 1A & 2 =		N
17.	Belt pull against frictional resistance in conveyor = (Sr. No. 15) – (Sr. No. 16) =		N
18.	Power kW at belt line, pk1 = $P \cdot v \div 1000$ =		kW
19.	Power kW minimum at motor shaft, pk2 = $pk1 \div \eta$ (i.e. efficiency factor) =		kW
20.	<u>Auxiliary information</u>		
20.1	Drive pulley/s turning resistance = As clause 10.15.0		N
20.2	Equivalent tractive pull P', including drive pulley resistance, Sr. 15 + Sr. 20.1 =		N
20.3	The drive pulley efficiency factor = $100 \times (Sr. 15 \div Sr. 20.2)$ =		%
20.4	Power kW at drive pulley shaft, pk3 = $P' \cdot v \div 1000$ =		kW

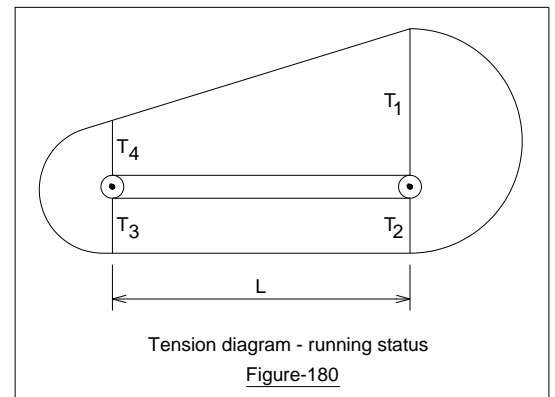
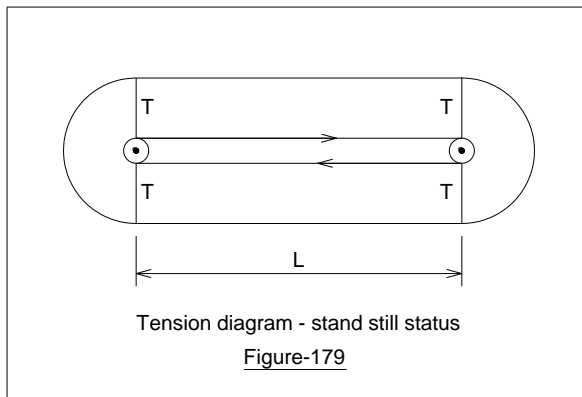
Chapter 11: TAKE-UP DEVICES

This chapter has **21 pages**. Following 5 pages write-up is typical only.

Non-floating take-up pulley

Screw take-up and conventional winch take-up have non-floating pulley i.e. take-up pulley remains at fixed location during conveyor operation. Thus pulley remains at same place during starting phase, steady running phase; braking phase and standstill phase. This is not to be confused with periodic resetting of pulley / tensions. The above conditions again apply after such resetting. Please refer cl-11.4.2 for working of this take-up.

Rule applicable to straight conveyor



First, let us analyse the condition applicable to straight profile conveyor. The figure-179 shows belt tension when conveyor is standstill. The belt will have uniform tension T throughout the length (ignoring marginal variations due to gradient).

Now the conveyor is started and tensions will occur as shown on figure-180. The tension changes between the two pulleys have straight-line variation (it is true for all practical needs). The belt working being under fully elastic condition, it obeys the following general rule for stress – strain.

E: Modulus of elasticity (constant)

T: Original tension in belt

T': New tension in belt

B: Belt width

y: Original length of belt

δy : Increase in belt length of original length 'y'

Increase in tension = $T' - T$.

$$E = \frac{T' - T}{B} \cdot \frac{y}{\delta y}$$

$$\delta y = \frac{T' - T}{B} \cdot \frac{y}{E} \text{ (general formula)}$$

New average tension in carrying run = $\frac{1}{2} (T_1 + T_4)$. Therefore, increase in tension in carrying run = $\frac{1}{2} (T_1 + T_4) - T$.

This causes increase of δc in belt length 'L' of carrying run. Now using these values in aforesaid general formula, we have tension increase in carrying run as

$$\delta c = \left[\frac{(T_1 + T_4)}{2} - T \right] \cdot \frac{L}{B \cdot E}$$

Now, average tension in return run = $\frac{1}{2} (T_3 + T_2)$. Therefore, increase in tension in return run = $\frac{1}{2} (T_3 + T_2) - T$.

This causes increase of δr in belt length L of return run. Now using these values in aforesaid general formula, we have

$$\delta r = \left[\frac{(T_3 + T_2)}{2} - T \right] \cdot \frac{L}{B \cdot E}$$

But belt total length has not changed because there is no shifting of any pulley.

Therefore, $\delta c + \delta r = 0$

$$\left[\frac{(T_1 + T_4)}{2} - T \right] \cdot \frac{L}{B \cdot E} + \left[\frac{(T_3 + T_2)}{2} - T \right] \cdot \frac{L}{B \cdot E} = 0$$

$$\left[\frac{(T_1 + T_4)}{2} \right] \cdot L - T \cdot L + \left[\frac{(T_3 + T_2)}{2} \right] \cdot L - T \cdot L = 0$$

$$\left[\frac{(T_1 + T_4)}{2} \right] \cdot L + \left[\frac{(T_3 + T_2)}{2} \right] \cdot L = 2 T \cdot L$$

The above signifies that carrying side new tension diagram + return side new tension side diagram = original tension diagram.

Therefore, New tension diagram area of conveyor = Original tension diagram area of conveyor.

Also, eliminating L in equation, we have

$$\frac{1}{2} (T_1 + T_4) + \frac{1}{2} (T_3 + T_2) = 2T.$$

$$T_1 + T_2 + T_3 + T_4 = 4T.$$

We will arrive at same derivation even if we use starting tensions $T_{1s}, T_{2s}, T_{3s}, T_{4s}$ or braking tensions $T_{1b}, T_{2b}, T_{3b}, T_{4b}$ instead of steady state tensions T_1, T_2, T_3, T_4 . Hence,

$$T_1 + T_2 + T_3 + T_4 = T_{1s} + T_{2s} + T_{3s} + T_{4s} = T_{1b} + T_{2b} + T_{3b} + T_{4b} = 4 T.$$

Thus the summation of above **4 tensions** in any operating status is constant and is equal to **4 times** the stand still tension on belt.

SHEAVE FRICTIONAL RESISTANCE EFFECT ON TAKE-UP FORCE

The take-up force is applied to take-up pulley via steel wire rope and sheave system. The steel wire rope and sheave system has frictional resistance during their actual movement or impending movement, which influence the take-up force reaching to take-up pulley. The rope tension prior to sheave and after the sheave is different during such dynamic situation. The general formula about the tension difference at sheave is as below.

T: Rope tension at pulling side T': Rope tension trailing side μ : Friction coefficient at bearing bore

D: Sheave nominal diameter d: Sheave bearing bore diameter

R: Resultant radial force on bearing

Leading side tension = Trailing side tension + Resistance to motion

$$T = T' + (R \cdot \mu \cdot d \div D) + (\text{Rope wrap resistance})$$

$$\text{Tension loss} = T - T' = (R \cdot \mu \cdot d \div D) + (\text{Rope wrap resistance})$$

The resultant force R is vector sum of rope tensions and sheave self weight (for this purpose assuming same tensions at both side is adequate). The calculation of resultant force to this detail is cumbersome,

and we can add about 5% in the resultant force calculated on the basis of tensions alone, to account for self weight of sheave.

$$R \text{ for } 90^\circ \text{ wrap} = (T \cdot \sqrt{2}) \times 1.05 = 1.41 T \times 1.05$$

$$R \text{ for } 180^\circ \text{ wrap} = (2 \times T) \times 1.05 = 2 T \times 1.05$$

In general D/d ratio can be considered 5:1 i.e. ratio d/D = 0.2. Often, the usual belt conveyors horizontal gravity take-up have bush bearing fitted sheaves with provision for grease lubrication along with inbuilt small reservoir of grease (grooves etc.). The sheaves in such take-up make some rotary motion during start-up and oscillating/slow rotary motion during most of the time. Hence lubrication can be classified as 'Boundary, in between thin film and mixed film' with friction coefficient say 0.075 (bronze, gun metal bush) or as chosen by designer.

Regarding rope wrap resistance; designer to use data/formula from wire rope manufacturer only. It can have differing value for rope type and diameter of rope. For the purpose of explaining here, it is considered 0.7 kg per 100 kgf tension in rope (i.e. 7 kgf per 1 tonne force). Now, for 100 kgf tension in rope, the tension loss will as below, at 90° wrap sheave and 180° wrap sheave.

$$\begin{aligned} \text{At } 90^\circ \text{ wrap, tension loss} &= (R) \cdot \mu \cdot (d \div D) + \text{Rope wrap resistance} \\ &= (1.41 \times 100 \times 1.05) \times 0.075 \times (0.2) + (0.7 \times 100 \div 100) \text{ kgf} = 2.22 + 0.7 = 2.92 \text{ kgf per } 100 \text{ kgf force.} \end{aligned}$$

The tension being 100 kgf, it is also 2.92%.

$$\begin{aligned} \text{At } 180^\circ \text{ wrap, tension loss} &= (R) \cdot \mu \cdot (d \div D) + (\text{Rope wrap resistance}) \\ &= (2 \times 100 \times 1.05) \times 0.075 \times (0.2) + (0.7 \times 100 \div 100) \text{ kgf} = 3.15 + 0.70 = 3.85 \text{ kgf per } 100 \text{ kgf force.} \end{aligned}$$

The tension being 100 kgf, it is also 3.85%.

Most of the horizontal gravity take-up can have 90° wrap 3 'active' sheave. If rope 2 parallel lines are creating final take-up force, then there will be 3 set 'active' sheave, but take-up force loss will be same, because each line is dealing with half of tension and 2 of them is making 1 of take-up force.

For first 90° sheave: Pulling side tension 100 kgf. Then trailing side tension $100 - 100 \times 2.92 \div 100 = 97.08 \text{ kgf}$

For second 90° sheave: Pulling side tension 97.08 kgf. Then trailing side tension $97.08 - 97.08 \times 2.92 \div 100 = 94.245 \text{ kgf}$

For third 90° sheave: Pulling side tension 94.245 kgf. Then trailing side tension $94.245 - 94.245 \times 2.92 \div 100 = 91.49 \text{ kgf}$ say 91.5 kgf

Therefore total loss $100 \text{ kgf} - 91.5 = 8.5\%$

As usual, this frictional force does not have fixed direction, and it creates difficulty about the available take-up force. Frictional force direction is opposite to motion, but direction of motion is up or down!

When the belt is stretching; the counter weight moves down by gravity as a pulling side, and take-up pulley is on trailing side. So, force at take-up is counter weight force minus frictional force.

When the belt is contracting, the take-up pulley acts as the pulling side, which lifts the counter weight up against gravity. As the counter weight gravity force is fixed value, the force acting at take-up is equal to counter weight force plus frictional force.

Thus in such horizontal gravity take-up, when counter weight is descending the take-up force will be less, however, when it is ascending the take-up force will be more. The difference could be approximately $\pm 10\%$. Therefore, it is advisable to consider design take-up force with +10% margin, so that even with 10% fall in this value, the required minimum take-up force will be present all the time for

transmission of power, from drive pulley to belt. During stabilized operation, the system will tend to acquire 'neutral' states minimising the frictional residual effect during that particular time frame due to subtle presence of agitation. However its benefit is none during start-up, stoppage and varying tension situation in belt.

It is also necessary to lubricate rope sheave system at regular interval for minimum adverse effect by friction. The sheaves used to equalise tension in 2-lines rope system, to be ignored in calculating the tension loss.

Winch take-up

In case of winch take-up, the effect of this frictional resistance can be easily accounted while setting the winch tensions. For example, if the take-up force requirement is 10,000 kgf and friction total loss is 7.5%, then one can apply the tension corresponding to 10,750 kgf while tightening the winch. If the tension sensor is very close to take-up pulley, then the effect of friction loss in tension setting will be least. When winch take-up is being used after long idle time, it is advisable first to lubricate bush bearings for assured result.

The winch take-up with large mechanical advantage or high class system often uses anti-friction bearing to reduce the frictional problem. Also, the horizontal loop gravity take-up with more number of active sheaves can have antifriction bearing fitted sheaves, for important conveyors. Each sheave can have 2-double row spherical roller bearings or 2 taper roller bearings placed in opposition. Such sheave will have practically negligible turning resistance because their μ value will be 0.0025 etc instead of 0.075 i.e. 30 times less when there are antifriction bearings. However, rope wrap resistance portion will remain same.

Note: The safety guards are not shown in various figures of take-ups for clarity purpose. However, the vertical gravity take-up, horizontal gravity take-up and winch take-ups are provided with adequate safety guards made from wire mesh. Often, the statutory requirements are also applicable in this matter.

Chapter 13: BELT TENSIONS AND CONVEYOR DESIGNING

This chapter has **128 pages**. Following 10 pages write-up is typical only.

BELT TENSIONS AND CONVEYOR DESIGN

The belt conveyor starts operating once the belt is set in motion. However, various elements of belt conveyor, which are in contact with belt, impose resisting-forces to the belt motion; and conveyor would stop unless powered by external means. The calculation of various resisting (opposing) forces to belt motion (conveyor operation) has been already described in chapter-10.

It is necessary to provide continuous tractive pull, equal to sum of all resisting-forces, to ensure continuous operation of belt conveyor. This tractive pull is applied to belt by drive-pulley/s. The tractive pull results into specific pattern of belt tension along conveyor axis. This will be clear from following example on railway train.

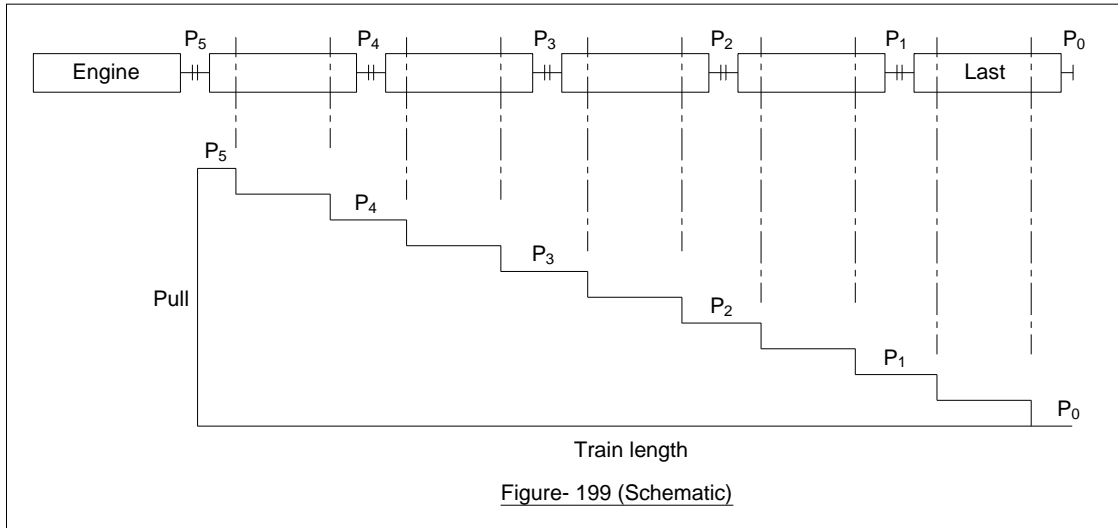


Figure-199 shows the railway engine pulling 5 Nos. of wagons, on horizontal level track, at steady speed. The practical sense indicates that the engine draw-bar pull P_5 will have highest magnitude, because same is pulling all 5 wagons. The next coupling has pull P_4 , which will be somewhat less than the preceding, because it has to overcome resistance of only four wagons. The last-van end coupling will have zero pull, because same is not pulling any thing. Therefore, in this particular example:-

$$P_5 > P_4 > P_3 > P_2 > P_1 > P_0$$

Thus, we have differing values of pull along centre line of train. These values shall be in specific pattern (graph) for specific dynamic condition (i.e. steady speed, accelerating, breaking, level track, up gradient, down gradient, convex track, concave track etc). The aforesaid figure shows the schematic graph of pull (tension) considering track is level, each wagon has 4 wheels, no air resistance, steady speed etc. (The pull occurring in coupling / chassis drops down at each axle, which is source of resistance here). This simple example explains the varied values of pull along train length.

The belt conveyor case is also somewhat similar to train (more particularly endless train). Imagine that the belt is made-up of number of segments connected together along belt width, as series of segments forming continuous belt. Like train, each segment is synonymous to wagon, and thereby, the belt will have presence of varied tension values in different segments of belt. The designer can choose segment length smallest to the largest to find out tension on both the ends of segment.

This simple example explains the varied tension values in belt, when same is subjected to tractive pull. It is essential to know belt tensions at various important / critical points in conveyor to enable designing of related components at those points. Also, the information of tensions at different point enable to establish maximum tension occurring within the belt loop, and thereby selection of belt itself. In fact, conveyor can not be designed without knowing belt tensions. Therefore, calculating and deciding the belt tensions has a center-stage place in conveyor design. Deciding the critical belt-tensions with minimum calculations requires engineering skill and imagination due to many variants present in the calculations, although use of computer has made task less difficult. Broadly speaking the variants present is as under.

A1: Full capacity	B1: Steady speed	C1: Conveying friction coefficient upper value
A2: Partial capacity	B2: Starting	C2: Conveying friction coefficient lower value
A3: Empty	B3: Braking	

During conveyor operation one variant of each group is present. Theoretically, this will require nearly 18 set of calculations for each conveyor, to cover every (most of the) possibilities. These combinations are-

- | | | |
|------------|------------|------------|
| 1) A1B1C1 | 2) A1B2C1 | 3) A1B3C1 |
| 4) A1B1C2 | 5) A1B2C2 | 6) A1B3C2 |
| 7) A2B1C1 | 8) A2B2C1 | 9) A2B3C1 |
| 10) A2B1C2 | 11) A2B2C2 | 12) A2B3C2 |
| 13) A3B1C1 | 14) A3B2C1 | 15) A3B3C1 |
| 16) A3B1C2 | 17) A3B2C2 | 18) A3B3C2 |

In addition, there are more possibility of variants due to material loading on part lengths i.e. only on up incline or down incline etc. This chapter describes the procedures to calculate tensions in belt and thereby conveyors design, and finally to establish the tension verses conveyor length diagram (tension diagram). The tension diagram depicts the variation in tension with respect to different points on conveyor. This provides comprehensive view on the dynamic behaviour of conveyor, which enables to make decisive judgement to finish conveyor design with few sets of calculations instead of 18 or more sets.

The belt tensions are of primary importance for conveyor design. However, as mentioned earlier, one can not design belt conveyor unless he is well conversant with various components of conveyor / machine. Therefore, foregoing chapter-1 to chapter-12 provide necessary information on conveyor components, and basic issues concerning to conveyor design. The **theme** of this chapter is to calculate belt tensions, and thereafter use these tensions to make conveyor design. This aim is implemented in this chapter by covering all the related topics, mainly as below:

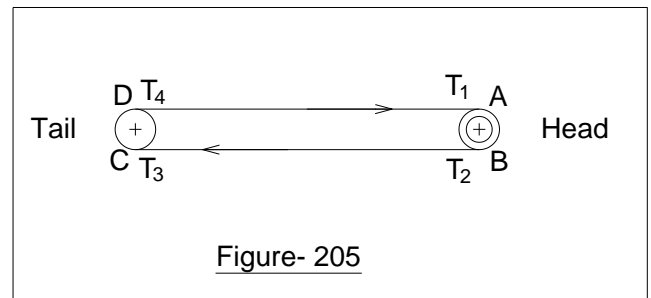
- Introduction to the issue of belt tensions (as described in the beginning of this chapter).
- Minimum tensions to suit power transmission between drive-pulley and belt; and limitation of belt sag between idlers.
- Principles for calculation of belt tensions.
- Take-up implication on belt tensions.
- Principles for calculation of moving mass. The relation between starting factor at belt line and starting multiplier at motor shaft. Also the relation between braking factor at belt line and at brake.
- Derivation of primary engineering quantities such as M_m , M_b , M_c , M_r , I , etc. from the known functional parameters of conveyor; as a preparatory work for conveyor design.
- Step-by-step procedure for conveyor design commencing from primary functional parameters of conveyor to calculation of belt tensions (using information from earlier chapters and also from this chapter).
- The selection of belt to suit tensions, 3 examples for conveyor design, etc...The tension diagram for typical horizontal conveyor, incline-up conveyor and decline (regenerative) conveyor.

Principles for calculating tensions in belt, at various locations

During design of belt conveyor, the prime necessity is to calculate tensions T_1 , T_2 , T_3 and T_4 , as shown in figure-205. For this purpose, first of all tractive pull P is calculated, and also the belt wrap angle θ and friction coefficient μ is decided. The values of T_1 and T_2 are derived easily from known values of aforesaid P , θ and μ , as below.

$T_1 = C_1 P$, $T_2 = C_2 \cdot P$, (where C_1 and C_2 are derivatives of $e^{\mu\theta}$)

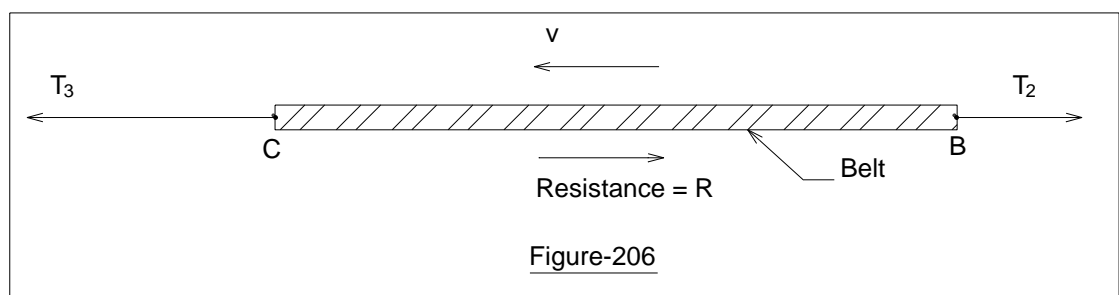
Now, the next step in design is to calculate remaining tensions T_3 and T_4 . This is done on the basis of stability / motion of the belt segment between T_3 and T_2 . The belt segment



is acted upon by force T_3 , conveying resistance force and force T_2 .

It is necessary to consider appropriate direction of forces while generating equation of motion. The forces on belt segment due to tensions have outward direction at both the ends all the time. The conveying resistance direction is opposite to belt motion / velocity. Chapter-10 calculations for conveying resistance also consider its direction opposite to belt motion / velocity, and therefore, it matches to above requirement. Thus, the conveying resistance values as per formulae of chapter-10 are to be directly treated as resistance in equation for motion. Following cl.-13.4.1 and 13.4.2 explain the methods to calculate tensions T_3 in relation to tension T_2 and it also helps to formulate general rule for calculating such tensions. Cl.-13.4.3 explains the method to calculate tension T_4 , and cl.-13.4.4 explains the method for calculation of belt tension at any point, using the rules so formulated.

T_3 during steady state operation:



Consider belt segment BC extending from head end to tail end as shown in figure - 206. The forces acting on this segment are T_2 , resistance R and T_3 as shown in this figure. The resistance is opposing belt motion and hence it is shown in direction opposite to velocity v . Because the belt and thereby the segment is moving at steady speed v , the helping and opposing forces are equal in magnitude.

$$\therefore T_3 = \text{Conveying resistance} + T_2$$

Therefore, rule is '**segment front end tension = conveying resistance + segment rear end tension.**'

The segment front end and rear end are with respect to segment motion (velocity v).

The value of resistance and T_2 are known which enable the calculation of T_3 . In deriving this equation we have considered the direction of resistance as opposite to belt velocity. This is in harmony with

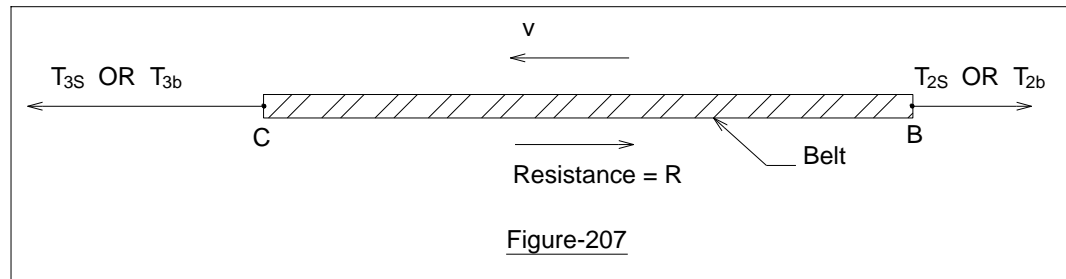
direction of resistance considered in chapter-10. Therefore, the resistance values derived as per formulae in chapter-10 are to be directly used in this equation together with their positive or negative sign.

T₃ during starting / stopping of conveyor:

In above figure, T_{2s} and T_{3s} are tensions during starting phase.

In above figure, T_{2b} and T_{3b} are tensions during braking / stopping phase.

The starting / stopping signifies acceleration / deceleration phase of conveyor. The value of acceleration during starting phase of conveyor is known. The belt and thereby its segment is picking-up speed, and



hence, there is net force Fn in direction of velocity v. Referring to figure-207, the equation for forces is

$$\therefore F_n = T_{3s} - \text{conveying resistance} - T_{2s}$$

$$\therefore T_{3s} = \text{conveying resistance} + F_n + T_{2s}$$

The segment mass M and acceleration a are known. Therefore, $F_n = M \cdot a$, and equation becomes

$$T_{3s} = \text{conveying resistance} + M \cdot a + T_{2s}$$

Therefore, **rule is segment front end tension = conveying resistance + inertial resistance + segment rear end tension.**

The equation for motion is same whether body is accelerating or decelerating. In case of stopping phase, the value of acceleration a' is negative, and hence, a' of negative value is to be used in the above equation. Referring to figure-207, the equation for motion is:

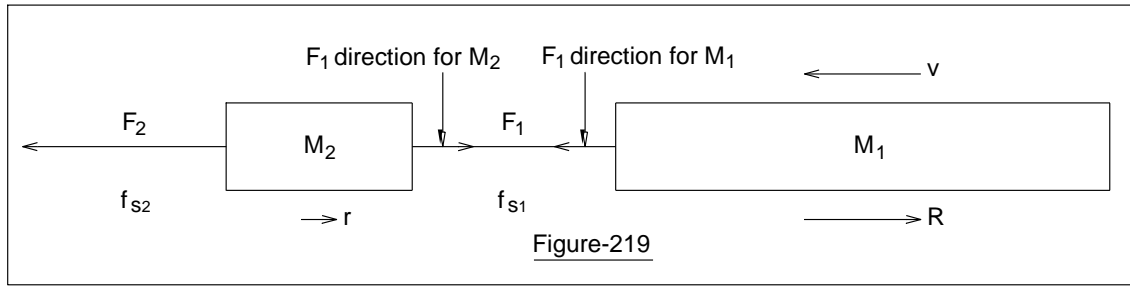
$$T_{3b} = \text{conveying resistance} + M \cdot a' + T_{2b}$$

Therefore, **rule is 'segment front end tension = conveying resistance + inertial resistance + segment rear end tension'.**

During steady state operation, the value of a or a' is zero, and equation of cl.-13.4.2 converts into equation of cl.-13.4.1

Starting factors relationship (for conveyor of + resistance during steady state)

Consider the 'rigidly' coupled masses M1 and M2 as shown in figure-219. Here rigidity is with respect to theory of motion. Belt conveyor is practically constant load machine, and therefore, conveyor motional resistance during starting (changing in speed) can be considered same as during steady speed phase.



These coupled masses form one 'rigid' system. We intend to analyse the force situation during acceleration phase of this system. The system is accelerated by drive unit, which applies force F_2 , and system starts moving (fig-219). The system accelerates against motional (conveying) resistance R and r .
 F_2 : Force being applied by drive (equivalent with respect to belt line)

R : Conveyor specific- case (partial capacity etc) resistance, steady speed, at belt line, opposite to 'v'.

r : Drive unit motion resistance (constant, average during start-up), equivalent at belt line, opposite to 'v'.

η_1 : Drive efficiency (as a fraction of one, and not in %), in relation to value of R and starting status

$$\eta_1 = \frac{1}{1 + \left(\frac{1}{\eta} - 1\right) \cdot \frac{Rd}{R}} = 1 \div \left[1 + \left(\frac{1}{\eta} - 1\right) \cdot \frac{Rd}{R} \right] \quad \text{Refer Addendum for explanation /derivation of the formula.}$$

The Rd is conveyor resistance of largest magnitude (most of the time at full design capacity), excluding drive. The η is drive efficiency at full design load. η is usual efficiency value for drive without fluid coupling. For drive with fluid coupling η is drive usual efficiency value excluding losses in fluid coupling.

$$\text{(As per definition of efficiency)} \quad \eta_1 = \frac{R}{R+r} \quad \therefore \eta_1(R+r) = R \quad \therefore R+r = \frac{R}{\eta_1} \quad \therefore r = \frac{R}{\eta_1} - R \quad \therefore r = \left(\frac{1}{\eta_1} - 1\right) \cdot R$$

The drive unit operates at full power during acceleration, irrespective of value of R . Therefore, drive resistance (transmission loss) ' r ' is same as for full load power transmission, irrespective of value of ' R '. The aforesaid formula for drive efficiency η_1 when applied to R provides ' r ' value as stated above during starting. At full design load, the formula result is usual efficiency $\eta_1 = \eta$.

Also during starting process:

- Gear box losses / resistance related to oil churning / shearing is less, average during starting phase.
- Gear box losses / resistance which is proportionate to torque transmission is more by starting factor.

The aforesaid increase / decrease tends to nullify each other in favour of value of r being considered here. For decisive analysis, the value of r is rarely more than 20% of resistance R . Therefore marginal deviation, if any, in this consideration practically does not affect final result which is based on $R + r$.

Thus, the system is being acted upon by three external forces. There is net force $F_N = F_2 - r - R$

$$\therefore \text{System acceleration } a = (F_2 - r - R) \cdot \frac{1}{M_1 + M_2}$$

Now, the aforesaid three external forces, results into presence of tension force F_1 at connection point between M_1 and M_2 . This being an internal tensile force, it acts on M_1 and M_2 as shown in figure-219.

Now, analyzing the motion of mass M_1 , it is subjected to two forces and has resultant F_n to move mass

$$M_1. \text{ Therefore, } F_n = F_1 - R \quad \text{Therefore, } \therefore \text{Sub-system acceleration} = (F_1 - R) \frac{1}{M_1}$$

But system and subsystem are mechanically coupled and hence moving at same value of acceleration. Therefore equating both the value of acceleration

$$(F_2 - r - R) \cdot \frac{1}{M_1 + M_2} = (F_1 - R) \cdot \frac{1}{M_1} \quad \therefore F_2 - r - R = (F_1 - R) \cdot \frac{M_1 + M_2}{M_1}$$

Now f_{s1} and f_{s2} are defined as starting factor in relation to $R = P$ as below

$$f_{s1} = F_1 \div R = F_1 \div P \quad \text{Hence } F_1 = R \cdot f_{s1}$$

$$f_{s2} = F_2 \div R = F_2 \div P \quad \text{Hence } F_2 = R \cdot f_{s2}$$

Substituting these values in above equation

$$R \cdot f_{s2} - \left(\frac{1}{\eta_1} - 1 \right) R - R = (R \cdot f_{s1} - R) \cdot \frac{M_1 + M_2}{M_1} \quad \therefore f_{s2} - \frac{1}{\eta_1} + 1 - 1 = (f_{s1} - 1) \cdot \frac{M_1 + M_2}{M_1}$$

$$\therefore f_{s2} = \frac{1}{\eta_1} + (f_{s1} - 1) \cdot \frac{M_1 + M_2}{M_1} \quad \text{or alternatively } f_{s1} = 1 + \left(f_{s2} - \frac{1}{\eta_1} \right) \cdot \frac{M_1}{M_1 + M_2}$$

This equation provides relationship between f_{s1} and f_{s2} . As per above definition, f_{s1} is starting factor at belt line with respect to tractive pull P . The f_{s2} is starting factor at motor shaft (output torque / force) with respect to tractive pull P . Henceforth, f_{s2} is named as starting multiplier at motor shaft, to avoid confusion with motor starting factor. Please note that **f_{s1} as well as f_{s2} are with respect to conveyor steady speed tractive pull P only. This is so because at steady speed numerically $P = R$.**

During starting phase, we are measuring power / torque as output quantity at motor shaft, and hence, motional resistance of motor (mechanical efficiency of motor proper) do not play part. However, this will be applicable during braking.

This derivation is for mechanically coupled masses making identical motion. In case of drive unit using fluid coupling, this condition holds good for drive mass up to secondary side of fluid coupling. The primary side of coupling and motor has already acquired nearly 80% of speed prior to load on secondary side begins to move. Thus, when the load will achieve speed from zero to 100%, the motor and primary side of coupling will have speed change from 80% to 100% during same time interval. Therefore, the motor and coupling part attached to it will be accelerating at 1/5 of load acceleration. Now, the inertial resistance being $M \cdot a$; the equal effect is achieved by reducing the 'motor side mass' to 1/5 value while applying acceleration magnitude same as load. Then the system is behaving as if it is rigid and entire body is making identical motion / acceleration. Thus drive unit with fluid coupling has $M_2 = \text{Drive pulley mass} + \text{output coupling mass} + \text{gear box mass} + \text{secondary side of fluid coupling mass} + 0.20 (\text{fluid coupling primary side mass}) + 0.20 (\text{motor mass})$, which is an equivalent mass for application to rigid system, inclusive of motor. In case, the load starts moving at 85% of motor speed, then the above factor will be 0.15 instead 0.20. If need be, the designer can use actual value as applicable for specific type of fluid coupling. This can be also obtained from fluid coupling manufacturers. This applies during load acceleration motion only. The fluid coupling mass should include oil mass, which is present in working chamber when load starts moving.

It is further clarified that f_{s2} is starting factor in relation to P and is different from conventional starting factor of drive which is in relation to normal torque of motor / drive. The f_{s2} is useful to decide required starting torque at motor shaft (**and / or** secondary side of fluid coupling) from the known value of P , depending upon the boundary line for analysis. After knowing this required value of torque at motor shaft, it is easy to derive the conventional starting factor in relation to normal running torque of motor.

This f_{s2} is mentioned here as **starting multiplier**, to distinguish it clearly from conventional starting factor of motor. Also drive efficiency $\eta_1 = \eta$, when conveyor is analysed at full design load and analysis boundary is for the complete drive.

The equation is applicable to conveyor of positive resistance (during steady state) as direction of r and R is same, opposite to belt velocity. The value of f_{s1} can not be less than 1 for accelerating conveyor. Arbitrary/impractical value of any one factor can result in to interesting interpretation.

Example – 7

Make the basic design (primary design) of belt conveyor to suit following application data.

Design input

1000 mm belt, 500 meter c/c horizontal length, -42 meter lift, straight decline regenerative conveyor is conveying iron ore at the design capacity of 2150 mtp. The iron ore design bulk density is 2400 kg / m³, size 100 % (-) 150 mm and surcharge angle 20°.

The belt conveyor is equipped with 1000mm wide belt having St-1000 carcass, covers 6mm and 4 mm and grade-M. The belt design mass is 20.8 kg/m. The belt is running at 2.60 mps speed on 35° trough 3-equal roller carrying idlers, and 10° trough 2-roller return idlers. The carrying idler pitch is 1.0m and return idler pitch is 3.0m. The idlers are without forward tilt. The carrying as well as return idlers have 133mm diameter. The conveyor is operating 16 hrs a day, and 300 days in a year. The ambient temperature is in the range - 10° C to 30° C., B = 1.0 m, H = -42.0m, Hm = -42.0 m Lh = Lm = 500 m Mb = 20.8 kg/m Mc = 20.00 kg/m Mr = 6.5 kg/m Mm=229.7 kg/m C = 1.20 $\eta = 0.95$ (including turning resistance of drive pulley). The drive is without fluid coupling. The conveyor schematic arrangement is as shown in figure-235C.

Solution

The belt conveyor design has been presented here using C-factor(stage-1 design). The exercise is to be also done as per detailed formula for tractive pull, power and tensions (i.e. stage-2 design). Both stage design makes the calculations very lengthy, and hence second stage design is not given here due to space limitations,

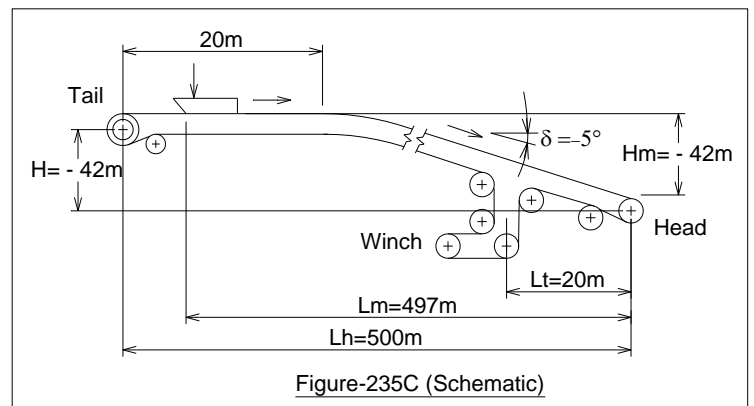


Figure-235C (Schematic)

Nevertheless, principles of calculations remain same and second stage design does not pose any difficulty. The designer will also find that the difference in results between both stage design, is marginal for such long conveyor.

Here, calculations have been made for zero capacity and 100 % capacity. In actual solution, the calculations are also to be done for few partial capacities such as 10%, 25%, 50% etc. to see which capacity results in to worst conditions for design.

This conveyor being practically straight decline, the exercises for material load on part lengths of conveyor is not needed. The belt conveyor with ups and downs will also need checking for worst design condition, considering material load on some of the lengths.

The solution presented here checks for following main conditions only, which are governing for many of the cases.

- 1) Maximum positive power which, is occurring during empty condition (zero capacity). This being straight and sufficiently declined conveyor, such situation will occur for empty condition. (However, if the conveyor is made-up of horizontal, up-incline and decline zones, then it can occur considering material load on non-decline lengths. The analysis is to be done considering usual value of f.
- 2) Maximum negative power on the basis of $f = 0.012$ applicable to main resistance, and balance frictional resistances as 75 % of usual value. The analysis is done for 100 % capacity.

There is certain ambiguity regarding this. When resistances calculations are made as per detailed formula, $f = 0.012$ is applied to main resistance only in accordance with various standards but other frictional resistances will remain as per usual values. Now when calculations are made with C-factor, it will not be consistent to apply $f = 0.012$ for all frictional resistance included in C. Therefore, we are splitting C-factor for main resistance and for other frictional resistances. Hence, it is proposed here to use $f=0.012$ for main resistance (friction) of C-factor, and balance frictional resistances, reflected in (C-1) are considered as 75 % of usual value, for design safety and consistency. This is also applicable to design being done according to detailed formula wherein main resistance will be based on $f = 0.012$, and balance frictional resistances will be considered 75% of usual value, to have common platform for both methods of calculation.

The analysis is made for 100 % capacity.

Component (C-1) includes material acceleration resistance, which will also get multiplied by 0.75, because it can not be separated. However this is to be accepted, as an insignificant compromise in design calculations.

- 3) Power and tensions considering usual value of f and usual value of other frictional resistances. The analysis is for 100 % capacity.
- 4) Actual expected power and tensions considering all frictional resistances as 85 % of usual. The analysis is for 100 % capacity.

In all the above analysis viz. 1), 2), 3) and 4), the non-friction resistance (slope resistance) is not subjected to any modification, because it is absolute (definite) in nature.

The value of 'f'

The adjustment in basic value of 'f' is attributed to speed, material flexure and temperature condition only. All other conditions are being considered ideal / standard. The maximum value of sag is limited to 1 %

0.02	Basic value of 'f.'
- 0.003	15 % reduction due to speed.
+ 0.003	15 % increase for difficult flexure of material.
+ <u>0.002</u>	10 % increase for temperature
0.022	

As would be seen the drive is without fluid coupling. So $\eta = 0.95$ corresponding to complete drive and largest numerical value of tractive pull.

Maximum permissible acceleration and deceleration

Friction coefficient $\mu_1 = 0.6$ or $\tan (1.2 \times 20)$ whichever is less. Therefore $\mu_1 = 0.45$

The maximum value of permissible acceleration $= g \cdot (\mu_1 \cdot \cos\delta - \sin\delta) = 9.81 \times [0.45 \times \cos(0) - \sin(0)]$
 $= 9.81 \times [0.45 - 0] = 4.41 \text{ m/sec}^2$

Max. value of permissible deceleration = $g (\mu_1 \cdot \cos\delta + \sin\delta) = 9.81 [0.45 \cdot \cos(-5) + \sin(-5)] = 3.54 \text{ m/sec}^2$

Minimum tension for sag (1 %)

$$T_C = 12.5 M \cdot g \cdot p_C \div (\text{sag \%}) = 12.5 \times (229.7 + 20.8) \times 9.81 \times 1 \div 1 = 30718 \text{ N}$$

$$T_R = 12.5 M \cdot g \cdot p_r \div (\text{sag \%}) = 12.5 \times 20.8 \times 9.81 \times 3.0 \div 1 = 7652 \text{ N}$$

Pulley drive factors

The drive factors on the basis of herringbone rubber lagging, friction coefficient 0.35, 200° wrap angle and single pulley snub drive are : $C_1 = 1.418 \cong 1.40$, and $C_2 = 0.40$. The drive pulley is located at tail end.

Take-up

Fixed belt length type winch take-up is possibly more suited to \pm power drive. This can be located at suitable place. If provided near head end, it will have minimum tension during rated load steady state operation. If gravity take-up, then its location near head end is likely to be more suited for design.

1) Power empty belt (steady state).

This is based on $M_m = 0$ and $f = 0.022$

$$P = c \cdot f \cdot L_h (M_m + 2 M_b + M_c + M_r) \cdot g + H \cdot M_m \cdot g = 1.2 \times 0.022 \times 500 \times (0.0 + 2 \times 20.8 + 20.0 + 6.5) \times 9.81 + (-42.0) \times (0.0) \times 9.81 = 8818.4 \text{ N}$$

Drive is sized for $P = 72292.2 \text{ N}$ of Sr. No. 2 below. Design input drive efficiency 0.95 is with respect to full load power transmission. Hence here at part load η_1 is as per following formula.

$$\eta_1 = 1 \div \left[1 + \left(\frac{1}{\eta} - 1 \right) \cdot \frac{R_d}{R'} \right] = 1 \div \left[1 + \left(\frac{1}{0.95} - 1 \right) \cdot \frac{72292.2}{18818.4} \right] = 0.6985$$

$$N_a = 8818.4 \times 2.6 \div 1000 = 22.92 \text{ kW} \quad N_m = 22.92 \div 0.6985 = 32.80 \text{ kW}$$

2) Maximum negative power (steady state).

This considers $f = 0.012$ for main resistance, balance friction force 75 % of usual value, and operation at 100 % capacity, steady speed.

$$\begin{aligned} P &= \text{Main resistance} + \text{balance frictional resistances} + \text{slope resistance} = (1.0) \cdot f \cdot L_h \cdot (M_m + 2M_b + M_c + M_r) \cdot g + (C-1) \cdot f \cdot L_h \cdot (M_m + 2M_b + M_c + M_r) \cdot g \cdot 0.75 + H \cdot M_m \cdot g \\ &= 1.0 \times 0.012 \times 500 \times (229.7 + 2 \times 20.8 + 20.0 + 6.5) \times 9.81 + (1.2-1.0) \times 0.022 \times 500 \times (229.7 + 2 \times 20.8 + 20.0 + 6.5) \times 9.81 \times 0.75 + (-42.0) \times 229.7 \times 9.81 \\ &= 17528.5 + 4820.3 - 94641 = 22348.8 - 94641 = -72292.2 \text{ N} \end{aligned}$$

The negative value of P signifies regenerative condition

$$N_a = -(72292.2) \times 2.60 \div 1000 = -187.96 \text{ kW} \quad N_m = -187.96 \times 0.95 = -178.56 \text{ kW} = 178.6 \text{ kW}$$

3) Power considering usual value of 'f' (steady state).

This considers $f = 0.022$, 100 % capacity, and steady state

$$\begin{aligned} P &= C \cdot f \cdot L_h \cdot (M_m + 2M_b + M_c + M_r) \cdot g + H \cdot M_m \cdot g \\ &= 1.2 \times 0.022 \times 500 \times (229.7 + 2 \times 20.8 + 20.0 + 6.5) \times 9.81 + (-42.0) \times (229.7 \times 9.81) \\ &= 38562.7 - 94641 = -56078.3 \text{ N} \quad N_a = -56078.3 \times 2.60 \div 1000 = -145.8 \text{ kW} \end{aligned}$$

$$\eta_1 = 1 \div \left[1 + \left(\frac{1}{\eta} - 1 \right) \cdot \frac{R_d}{R} \right] = 1 \div \left[1 + \left(\frac{1}{0.95} - 1 \right) \cdot \frac{72292.2}{56078.3} \right] = 0.9364 \therefore N_m = -145.8 \times 0.9364 = -136.5 \text{ kW}$$

4) Actual expected power (steady state)

This considers all frictional resistance as 85 % of usual value, steady state and 100 % capacity.

$$P = 38562.7 \times 0.85 - 94641 = 32778.3 - 94641 = -61862.7 \text{ N} \quad (\text{based on sr. no.3})$$

$$N_a = -61682.7 \times 2.6 \div 1000 = -160.8 \text{ kW}$$

$$\eta_1 = 1 \div \left[1 + \left(\frac{1}{\eta} - 1 \right) \cdot \frac{Rd}{R} \right] = 1 \div \left[1 + \left(\frac{1}{0.95} - 1 \right) \cdot \frac{72292.2}{61862.7} \right] = 0.942 \sim 0.95$$

$$N_m = -160.8 \times 0.95 = -152.76 \text{ kW}$$

Such power condition can be expected during major portion of time for conveyor operation. In this problem, the difference in result between sr.no.-2 and 4 is not very large, because conveyor is adequately regenerative. However, if the conveyor regenerative by less margin according to sr.no.-2, then although conveyor is classified as regenerative, but it may be working in positive mode for most of the time!

The calculation portion pertaining to belt tensions **has not been shown here**. The calculated belt tension summary is as per following table.

Belt tensions (Newton) **summary-1** : These tensions suit power transmission only.

1A) Empty belt, $f = 0.022$ (usual value of all frictional forces)

Steady : $P = + 8818.4$	$T_1 = 240.9$	$T_2 = 240.9$	$T_3 = 12346.0$	$T_4 = 3527.6$	$\Sigma T = 16355.4$
Starting : $P_s = + 64021.6$	$T_{1s} = 52468.4$	$T_{2s} = 52468.4$	$T_{3s} = 89630.2$	$T_{4s} = 25608.6$	$\Sigma T_s = 220175.6$
Stoppage: $P_b = - 60829.3$	$T_{1b} = 43839.6$	$T_{2b} = 43839.6$	$T_{3b} = 24331.7$	$T_{4b} = 85161$	$\Sigma T_b = 197171.9$

2A) 100 % capacity, $f = 0.012$ for main resistance, and balance frictional forces 75 % of usual

Steady : $P = - 72292.2$	$T_1 = 18298.12$	$T_2 = 18298.12$	$T_3 = 28916.9$	$T_4 = 101209.1$	$\Sigma T = 166722.24$
Starting: $P_s = +75987.77$	$T_{1s} = 79863.7$	$T_{2s} = 79863.7$	$T_{3s} = 106382.8$	$T_{4s} = 30395.1$	$\Sigma T_s = 296505.3$
Stoppage : $P_b = - 98367.3$	$T_{1b} = 31524.2$	$T_{2b} = 31524.2$	$T_{3b} = 39346.9$	$T_{4b} = 137714.2$	$\Sigma T_b = 240109.5$

3A) 100 % capacity, $f = 0.022$ (usual value) and balance frictional forces also as per usual values

Steady : $P = - 56078.3$	$T_1 = 10326.19$	$T_2 = 10326.19$	$T_3 = 22431.32$	$T_4 = 78509.62$	$\Sigma T = 121593.32$
Starting : $P_s = + 77970.7$	$T_{1s} = 82679.5$	$T_{2s} = 82679.5$	$T_{3s} = 109159$	$T_{4s} = 31188.3$	$\Sigma T_s = 305706.3$
Stoppage: $P_b = - 96384$	$T_{1b} = 30770.5$	$T_{2b} = 30770.5$	$T_{3b} = 38553.6$	$T_{4b} = 134937.6$	$\Sigma T_b = 235032.2$

4A) 100 % capacity, $f = 0.022$ (usual values). Then all usual frictional forces are multiplied by 0.85.

Steady : $P = - 61862.7$	$T_1 = 13170.22$	$T_2 = 13170.22$	$T_3 = 24745.08$	$T_4 = 86607.78$	$\Sigma T = 137693.3$
Starting: $P_s = +77742.8$	$T_{1s} = 82294.9$	$T_{2s} = 82294.9$	$T_{3s} = 108839.9$	$T_{4s} = 31097.1$	$\Sigma T_s = 304526.8$
Stoppage: $P_b = -96607.26$	$T_{1b} = 30794.0$	$T_{2b} = 30794.0$	$T_{3b} = 38642.9$	$T_{4b} = 135250.16$	$\Sigma T_b = 235481.06$

Although starting torque / force at motor shaft is same in all cases, but value of P_s reaching to belt is different due to acceleration effect. Similarly, braking torque / force at brake is same in all cases, but value of P_b reaching to belt is different due to deceleration effect.

Refer next page for tensions satisfying power transmission as well as winch take-up and sag.

Belt tensions (Newton) **summary-2** : Tensions suit power transmission and Winch take-up, and are final tensions.

These figures are rounded up to 0.5 N. The figures in last column are rounded up to 1.0 N.

1A) Empty belt, $f = 0.022$ (usual value of all frictional forces)

- Steady state tension adjustment by $(305706.3 - 16355.4) \div 4 = 72337.725$
- Starting state tension adjustment by $(305706.3 - 220175.6) \div 4 = 21382.675$
- Braking state tension adjustment by $(305706.3 - 197171.9) \div 4 = 27133.6$

Steady : $P = + 8818.4$	$T_1 = 72578.6$	$T_2 = 72578.6$	$T_3 = 84683.7$	$T_4 = 75865.4$	$\Sigma T = 305706.3$
Starting : $P_S = + 62575.3$	$T_{1S} = 73851.0$	$T_{2S} = 73851.0$	$T_{3S} = 111012.9$	$T_{4S} = 46991.4$	$\Sigma T_S = 305706.3$
Stoppage: $P_b = - 62293.2$	$T_{1b} = 70973.2$	$T_{2b} = 70973.2$	$T_{3b} = 51465.3$	$T_{4b} = 112294.6$	$\Sigma T_b = 305706.3$

2A) 100 % capacity, $f = 0.012$ for main resistance. Balance frictional forces 75 % of usual

- Steady state tension adjustment by $(305706.3 - 166722.2) \div 4 = 34746$
- Starting state tension adjustment by $(305706.3 - 296505.3) \div 4 = 2300.25$
- Braking state tension adjustment by $(305706.3 - 240109.5) \div 4 = 16399.2$

Steady : $P = - 72292.2$	$T_1 = 53044.1$	$T_2 = 53044.1$	$T_3 = 63662.9$	$T_4 = 135955.2$	$\Sigma T = 305706.3$
Starting: $P_S = + 75987.77$	$T_{1S} = 82164$	$T_{2S} = 82164$	$T_{3S} = 108683$	$T_{4S} = 32695.3$	$\Sigma T_S = 305706.3$
Stoppage : $P_b = - 98367.3$	$T_{1b} = 47923.4$	$T_{2b} = 47923.4$	$T_{3b} = 55746.1$	$T_{4b} = 154113.4$	$\Sigma T_b = 305706.3$

3A) 100 % capacity, $f = 0.022$, and all frictional forces of usual values.

- Steady state tension adjustment by $(305706.3 - 121593.3) \div 4 = 46028.25$
- Starting state tension adjustment by $(305706.3 - 305706.3) \div 4 = 0.0$
- Braking state tension adjustment by $(305706.3 - 235032.2) \div 4 = 17668.525$

Steady : $P = - 56078.3$	$T_1 = 56354.4$	$T_2 = 56354.4$	$T_3 = 68459.6$	$T_4 = 124537.9$	$\Sigma T = 305706.3$
Starting : $P_S = + 77970.7$	$T_{1S} = 82679.5$	$T_{2S} = 82679.5$	$T_{3S} = 109159$	$T_{4S} = 31188.3$	$\Sigma T_S = 305706.3$
Stoppage: $P_b = - 96384$	$T_{1b} = 48439.1$	$T_{2b} = 48439.1$	$T_{3b} = 56222.1$	$T_{4b} = 152606$	$\Sigma T_b = 305706.3$

4A) 100 % capacity, $f = 0.022$; then all frictional forces multiplied by 0.85

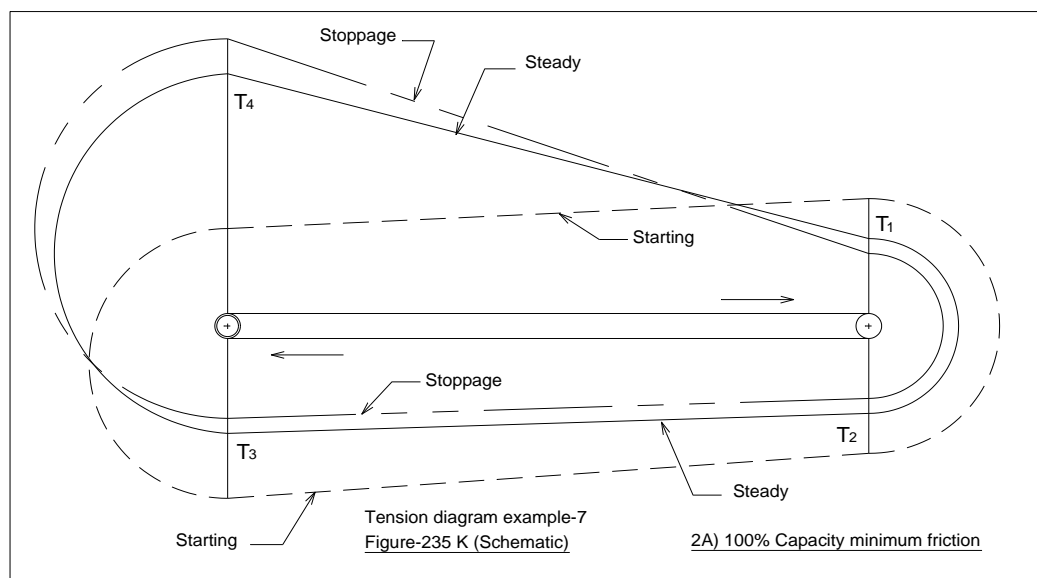
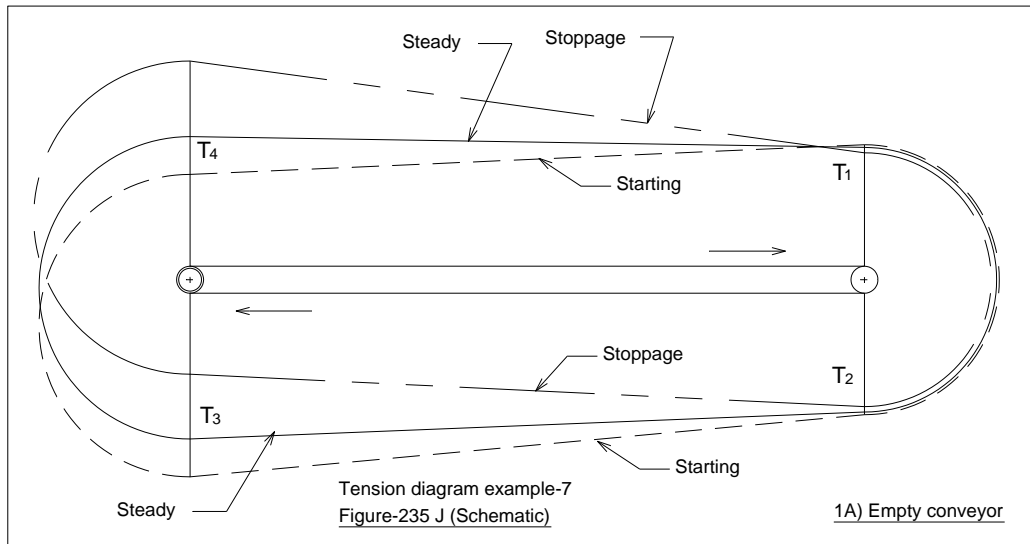
- Steady state tension adjustment by $(305706.3 - 137693.3) \div 4 = 42003.25$
- Starting state tension adjustment by $(305706.3 - 304527) \div 4 = 294.82$
- Braking state tension adjustment by $(305706.3 - 235481.1) \div 4 = 17556.3$

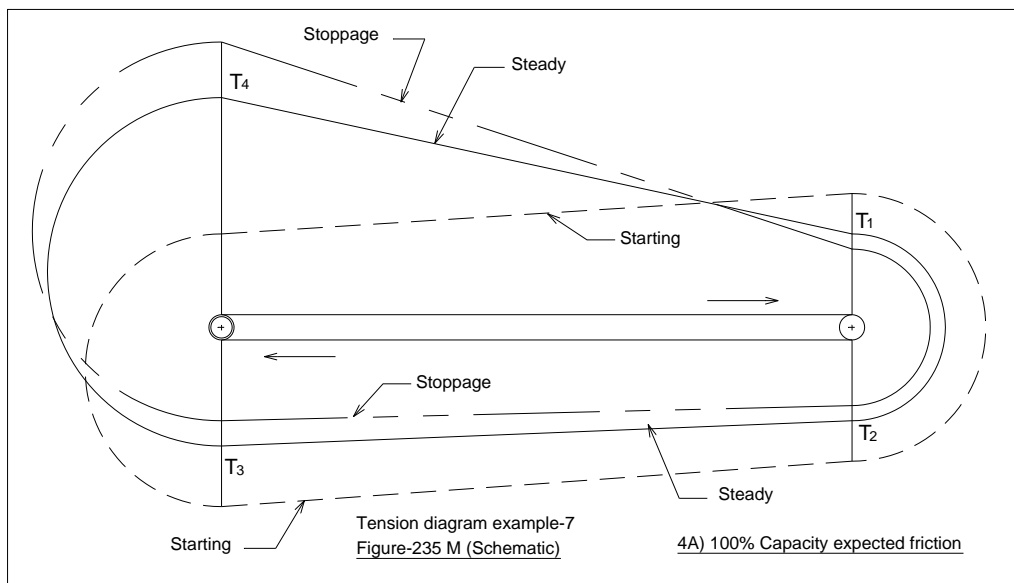
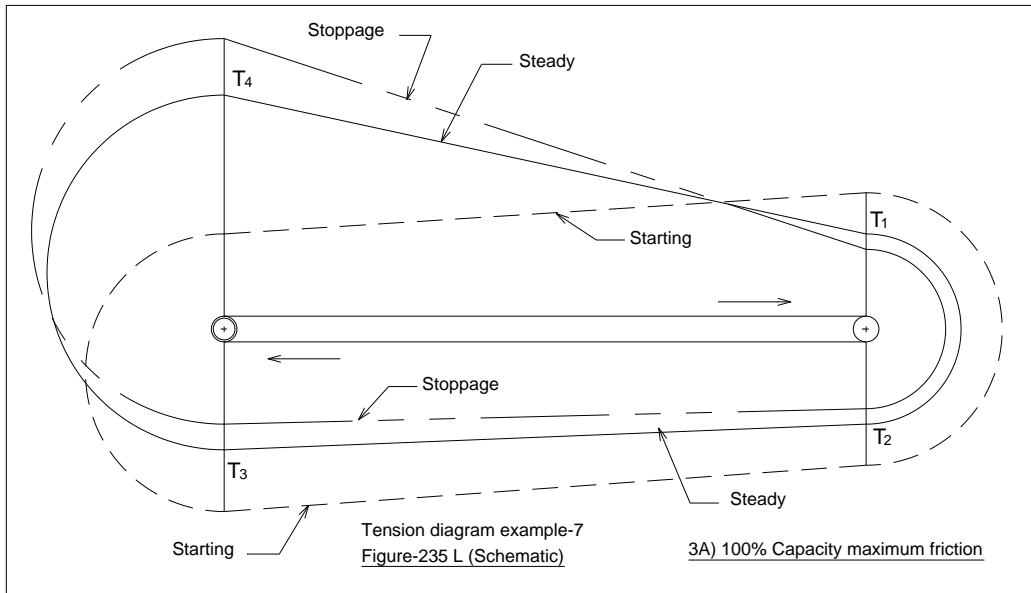
Steady : $P = - 61862.7$	$T_1 = 55173.5$	$T_2 = 55173.5$	$T_3 = 66748.3$	$T_4 = 128611$	$\Sigma T = 305706.3$
Starting: $P_S = + 77742.8$	$T_{1S} = 82589.8$	$T_{2S} = 82589.8$	$T_{3S} = 109134.8$	$T_{4S} = 31391.9$	$\Sigma T_S = 305706.3$
Stoppage $P_b = - 96607.26$	$T_{1b} = 48350.3$	$T_{2b} = 48350.3$	$T_{3b} = 56199.2$	$T_{4b} = 152806.5$	$\Sigma T_b = 305706.3$

The minimum required tensions for sag are $T_C = 30718$ and $T_R = 7652$. Now in all the above tensions $T_1/T_{1S}/T_{1b} > T_C$ and $T_4/T_{4S}/T_{4b} > T_C$. Also $T_2/T_{2S}/T_{2b} > T_R$ and $T_3/T_{3S}/T_{3b} > T_R$. Hence these tensions are also suitable for sag. Thus, aforesaid tensions are suitable for all 3 conditions and are final tensions for conveyor design. The conveyor has convex kink close to tail end. The tail end tensions are quite high which suggests adequacy of tensions near kink, and accordingly above tensions appears o.k. However, tensions at kink should be also checked, and if found less, all tension values are to be equally increased.

Tension diagram for example-7

Decline regenerative conveyor, equipped with tail end drive and fixed take-up, will have tension diagram similar to this depiction. The regenerative conveyor has plus minus fluctuation in tractive pull and, and hence tension diagram have been shown in figure-235J, 235K, 235L and 235M in accordance with four cases of analysis in example-7.





Chapter 14: Drive-unit functional requirements

CALCULATION FOR STARTING TIME

Torque developed by drive system during starting process has varying value. Therefore, truly speaking conveyor picks up speed under varied value of acceleration. The belt-starting factor is generally understood with reference to maximum value of torque occurring during starting process. The details of such torque variation can only be analysed with the help of electrical designer / fluid coupling manufacturer, in accordance with specific type, design, constructional features and test data. However, reasonably accurate time can be calculated by considering average constant value of torque and thereby constant value of acceleration during starting process, by application of empirical factor on maximum value of torque. Such empirical factor will have value in relation to each type of drive / coupling.

The starting time of belt conveyor can be calculated by different methods. Here, two methods have been described. The belt conveyor designer's preference is likely to be for method-1.

Method-1 (Applicable to drive type-1 and drive type-2)

From the known functional parameters of the drive system, find out the value of f_{S1} i.e. starting factor at belt line. Generally stated starting factors are for maximum value during starting process. Therefore:

The maximum value of starting pull on belt = $f_{S1} \cdot P$

The maximum value of accelerating pull on belt = $f_{S1} \cdot P - P = (f_{S1} - 1) \cdot P$

The average value of accelerating pull on belt = $k \cdot (f_{S1} - 1) \cdot P$

where k is applicable reduction factor according to shape of torque graph, for graph portion above load line.

Conveyor equivalent moving mass for the items being moved by the belt (including belt) is M_1

Therefore conveyor acceleration average during starting, $a = k \cdot (f_{S1} - 1) \cdot P \div M_1$

Acceleration time = $v \div a$

Total starting time from the moment the motor is switched on = time till drive develops torque equal to conveyor resistance + acceleration time, as above.

Method-2 (Applicable to drive type-1 and drive type-2)

Method-1 uses linear motion as a basis for calculation, with reference to belt line. This method uses rotary motion as a basis for calculation, with reference to motor shaft for drive type-1 and with reference to secondary side of fluid coupling for drive type-2.

If drive type-1 (without fluid coupling)

I : moment of inertia of system as referred to motor shaft, including motor.

T_{QM} : Motor shaft torque steady running condition (this is not rated torque, but it is demand torque).

T_{QMS} : Motor shaft maximum torque during starting process.

S_m : Motor maximum torque starting ratio $T_{QMS} \div T_{QM}$ (this is different from motor starting factor which is motor starting torque \div rated torque)

The maximum value of accelerating torque = $T_{QMS} - T_{QM} = S_m \cdot T_{QM} - T_{QM} = (S_m - 1) \cdot T_{QM}$

The average value of accelerating torque = $k \cdot (S_m - 1) \cdot T_{QM}$

where k is applicable reduction factor according to shape of torque graph, for graph portion above load line.

ω : Speed radians / sec (normal running speed)

α : Angular acceleration radian / sec² (average during starting phase)

Now $\alpha = (\text{Average accelerating torque}) \div (\text{Moment of inertia}) = k \cdot (S_m - 1) \cdot T_{QM} \div I$

Acceleration time $t = \omega \div \alpha$

Total starting time from the moment the motor is switched on = time till drive develops torque equal to conveyor resistance + acceleration time, as above.

If drive type-2 (with fluid coupling)

I : moment of inertia of system as referred to fluid coupling output shaft, which is being driven by it, including secondary side of fluid coupling.

T_{QC} : Fluid coupling torque on secondary side, steady running condition.

S_f : Fluid coupling starting factor, measured at secondary side of fluid coupling.

T_{QCS} : Fluid coupling torque on secondary side, maximum during starting process = $S_f \cdot T_{QC}$

The maximum value of accelerating torque = $T_{QCS} - T_{QC} = S_f \cdot T_{QC} - T_{QC} = (S_f - 1) \cdot T_{QC}$

The average value of accelerating torque = $k \cdot (S_f - 1) \cdot T_{QC}$

where k is applicable reduction factor according to shape of torque graph, for graph portion above load line.

ω : Speed radians / sec (normal running speed)

α : Angular acceleration radian / sec² (average during starting phase)

Angular acceleration average during starting, $\alpha = k \cdot (S_f - 1) \cdot T_{QC} \div I$

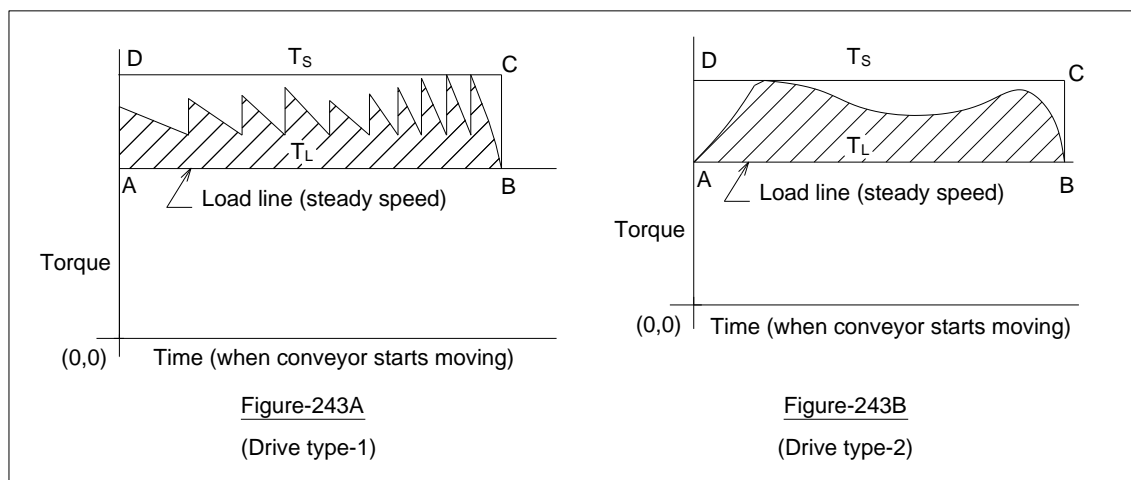
Acceleration time $t = \omega \div \alpha$

Total starting time from the moment the motor is switched on = time till drive develops torque equal to conveyor resistance + acceleration time, as above.

The value of 'k' for average constant acceleration.

The value 'k' in above calculation for acceleration time is as below (refer figure-243A and 243B).

$k = (\text{Hatched area}) \div (\text{Area of rectangle ABCD})$



Chapter 15: PULLEYS

Example Input data

Decide the shaft diameter at hub for secondary drive pulley and non drive tail pulley, considering keyed connection and following application data (reference example-5, chapter – 13)

The force in N and dimensions in cm have been used for conveniency.

Shaft material: Steel M.S (ductile), ultimate tensile strength $S_{ut} = 42000 \text{ N / cm}^2$. Yield tensile strength $S_y = 22000 \text{ N / cm}^2$, and modulus of elasticity $E = 205000 \times 100 \text{ N / cm}^2$

Belt width = 1400 mm = 140 cm. Pulley shaft bending arm $a = 30 \text{ cm}$

Pulley shaft bearings centre distance $L = 195 \text{ cm}$ (external bearing units). $L - 2a = 195 - 60 = 135 \text{ cm}$

Drive nature: Includes fluid coupling

For secondary drive pulley:-

Pulley diameter $D_d = 630 \text{ mm} + 24 \text{ mm} = 654 \text{ mm} = 65.4 \text{ cm}$

Steady state $T_1 = 62465 \text{ N}$, $T_2 = 25301 \text{ N}$. Starting state $T_{1s} = 78431 \text{ N}$, $T_{2s} = 24543 \text{ N}$

For tail pulley (consider full effect of gravity take-up for the sake of common size LT pulleys):-

Diameter $D_x = 500 \text{ mm} + 24 \text{ mm} = 524 \text{ mm} = 52.4 \text{ cm}$ Steady state $T_3 = 25229$, starting $T_{3s} = 28969 \text{ N}$

Solution (drive pulley)

Firstly it is convenient to decide necessary shaft diameter to suit deflection. The deflection is to be calculated during continuously occurring flexing (keyed connection), hence steady state tensions are applicable for deflection calculation.

Total belt force on pulley = $T_1 + T_2 = 62465 + 25301 = 87766 \text{ N}$

Resultant force on shaft including self weight = $1.04 \times 87766 = 91276 \text{ N}$

Therefore, force at each bearing / diaphragm including misalign effect $F = 1.05 \times 91276 \div 2 = 47919 \equiv 47920 \text{ N}$

Say designer opts for 7 minutes deflection at hubs considering hypothetical freely deflecting shaft

$$I = 3440 \cdot \frac{F \cdot a \cdot (L - 2a)}{2 \cdot E \cdot \theta} = \frac{3400 \times 47920 \times 30 \times 135}{2 \times 205000 \times 100 \times 7} = 2326.2 \text{ cm}^4$$

But $I = \pi \cdot d^4 \div 64$ Therefore $d = 14.75 \text{ cm}$ i.e. 147.5 mm

Thus for the shaft portion between hubs; the shaft required minimum diameter is 147.5 mm . This value will remain same whether shaft material is MS (st-42), En-8 or C-45; because their 'E' value is same.

Shaft size at hubs to suit stress (method -1) :-

Starting torque on shaft $T = (T_{1s} - T_{2s}) \times \text{pulley radius} = (78431 - 24543) \times 65.4 \div 2 = 1762137 \text{ N cm}$

Steady state bending moment $M = F \cdot a = 47920 \times 30 = 1437600 \text{ N cm}$ (F from deflection calculation)

For keyed shaft, selected working (occurring) stress S_s are as below.

$S_s = 0.3 S_y \times 0.9 \times 0.75 = 0.3 \times 22000 \times 0.9 \times 0.75 = 4455 \text{ N / cm}^2$

Or $S_s = 0.18 S_{ut} \times 0.9 \times 0.75 = 0.18 \times 42000 \times 0.9 \times 0.75 = 5103 \text{ N / cm}^2$

Selected $S_s = 4455 \text{ N / cm}^2$. The drive has fluid coupling, hence $C_t = 1.0$ and $C_m = 1.5$

$$d = \left[\frac{16}{\pi \cdot S_s} \cdot \sqrt{(C_t \cdot T)^2 + [C_m \cdot M]^2} \right]^{\frac{1}{3}} = \left[\frac{16}{\pi \cdot 4455} \cdot \sqrt{(1.0 \times 1762137)^2 + [1.5 \times 1437600]^2} \right]^{\frac{1}{3}} = 14.70 \text{ cm}$$

Thus for the shaft portion at hubs; the shaft required minimum diameter is 147.0 mm . If the shaft is made of material like En-8 (new name 080M40) C45 etc., then for shaft diameter upto 150 mm ; the stress value are $S_{ut} = 55000 \text{ N / cm}^2$ and $S_y = 28000 \text{ N / cm}^2$, the diameter will be 136 mm .

Shaft size at hubs to suit stress (method-2) :-

Selected $n = 2.40$. $T = 1762137$ N cm (for starting); $M = 1437600$ N cm (for steady state), as before
 $S_y = 22000$ N / cm², $S_e = 0.45 S_{ut} = 0.45 \times 42000 = 18900$ N / cm² (without keyway)
 $k = 0.75$. This is stress reduction factor for keyway effect.

$$d = \left\{ \frac{32 \cdot n}{\pi} \cdot \left[\left(\frac{T}{k \cdot S_y} \right)^2 + \left(\frac{M}{k \cdot S_e} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} = \left\{ \frac{32 \times 2.40}{\pi} \cdot \left[\left(\frac{1762137}{0.75 \times 22000} \right)^2 + \left(\frac{1437600}{0.75 \times 18900} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} = 15.32 \text{ cm}$$

Thus for the shaft portion at hubs; the shaft required minimum diameter is 153.2 mm as per this method.

The value of 'd' for material like En-8 (080M40) C45 normalised free cutting steel; $d = 140.8$ mm.

The shaft diameter as per method-2 is more. The designer can choose the diameter as per preferred method. The shaft diameter at bearing can be calculated for aforesaid torque value and app. 40% bending moment. The shaft size at coupling will be for aforesaid torque and zero bending moment. The shaft step difference coupling to bearing and bearing to hub should not be more than 10 / 12.5 % at each place. The shaft step difference hub to between hubs can be more as there is enough space for gradual tapering.

Solution nondrive pulley : This has been narrated briefly

$$F' = 1.075 \times (25229 \times 2) = 54242$$

$$\square F = 1.05 \times (54242 \square 2) = 28477 \text{ N}$$

Based on deflection: shaft diameter between hubs

$$d = \left[\frac{64}{\pi} \cdot 3440 \cdot \frac{F \cdot a \cdot (L - 2a)}{2 \cdot E \cdot \theta} \right]^{\frac{1}{4}} = \left[\frac{64 \times 3440 \times 28477 \times 30 \times 135}{\pi \times 2 \times 205000 \times 100 \times 7} \right]^{\frac{1}{4}} = 12.95 \text{ cm} = 129.5 \text{ mm}$$

For stress (method-1) : Shaft diameter for portion in hubs

$$d = \left[\frac{16}{\pi \cdot S_s} \cdot \sqrt{(C_t \cdot T)^2 + [C_m \cdot M]^2} \right]^{\frac{1}{3}} = d = \left[\frac{16}{\pi \times 4455} \cdot \sqrt{0 + [2.0 \times 28477 \times 30]^2} \right]^{\frac{1}{3}} = 12.49 \text{ cm} = 124.9 \text{ mm}$$

For stress (method-2) : Shaft diameter for portion in hubs

$$d = \left\{ \frac{32 \cdot n}{\pi} \cdot \left[\left(\frac{T}{k \cdot S_y} \right)^2 + \left(\frac{M}{k \cdot S_e} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} = \left\{ \frac{32 \times 2.90}{\pi} \cdot \left[0 + \left(\frac{28477 \times 30}{0.75 \times 18900} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} = 12.12 \text{ cm} = 121.2 \text{ mm}$$

The designer is to decide the shaft diameter steps to suit the above results and whether the shaft is removable or non-removable type.

Non-drive pulleys shaft diameter at bearings can be calculated for app. 40% of bending moment.

Note: The designer will observe that the pulley shaft size is often governed by deflection except for very narrow belt widths or very large diameter shaft. Further more, steel modulus of elasticity is same whether it is MS (common steel) or alloy steel. Hence, many times there are only limited advantages by using expensive steel, for general application. For conveyor equipped with fixed location take-up pulley (screw/winch take-up); the belt tensions during empty running or standstill condition often govern shaft diameter of non-drive pulleys.

Nature of load, stress and safety factor

As per well known mechanical /structural theory, when the load is applied suddenly on elastic body; the momentary stress are double than the subsequent (usual) stress. Mechanical or structural items are not

for one time use, but have life in months or years. So occurring stress should never exceed the elastic limit, even for a moment, or else mechanical item will go on deforming and will be useless very quickly.

Suddenly applied load (force) : Suppose there is 4 legged stool, each leg cross section area 1 cm^2 . Now water filled bucket weighing 40 kgf is gently placed on it (i.e. zero drop height), then instantaneous stress induced in the legs by effect of this load will be $2 \times 40 \text{ kgf} \div (1 \times 4 \text{ cm}^2) = 20 \text{ kgf/cm}^2$. This is a typical case for sudden application of load.

Gradually applied load force : Now, the above empty bucket (of negligible weight) is placed on stool and then it is being filled with 40 kg of water. This is the case of gradually applied load and stress induced in the legs by effect of this load will be $40 \text{ kgf} \div (1 \times 4 \text{ cm}^2) = 10 \text{ kgf/cm}^2$.

In general, all engineering formulae are for gradually applied load (or as if existing load)! That is force F acts on body and resulting stress due to this ' F ' is calculated. It is convenient to have only one formulae for design calculation and amplify (to apply multiplication factor to readily known) force to convert it into F for using in formula.

Hybrid (mixed) situation : For most of the cases, stress situation is often hybrid i.e. of mixed nature. In above example if stool own weight is 20 kgf then situation become as below.

Stress in leg prior to placement of water filled bucket $20 \text{ kgf} \div (1 \text{ cm}^2 \times 4) = 5 \text{ kgf/cm}^2$.

The stress by placement of 40 kgf of bucket weight = $40 \div (1 \times 4 \text{ cm}^2) = 10 \text{ kgf/cm}^2$.

But this being suddenly applied load, the instantaneous stress due to this is $2 \times 10 = 20 \text{ kgf/cm}^2$.

The total stress in leg = $20 + 5 = 25 \text{ kgf/cm}^2$.

Above is exact method. But frequently used other method is to apply over all multiplying factor, for particular category of items/components. For the aforesaid specific example multiplying factor to total load is 1.6667 (inferred from exact method). Then calculation becomes simple as below.

Total force acting at bottom end of stool leg = stool weight + water filled bucket weight = $20 + 40 = 60 \text{ kg}$.

The load multiplier 1.6667 to account for effect of suddenly applied load to this system of stool and water filled bucket.

Accordingly stress = $60 \times 1.6667 \div (1 \text{ cm}^2 \times 4) = 25 \text{ kgf/cm}^2$ (as above).

In case of belt conveyors pulley shaft design; engineers have found the aforesaid method of overall multiplier convenient; and so it is the general practice. In case of Soderberg formula; the stated value of ' n ' accounts for influence of suddenly applied load as well as safety factor. In case of method-1, the multiplier C_t and C_m to load, and reduced value of permissible stress; together accounts for suddenly applied load effect plus safety factor.

The belt conveyor equipped with screw or winch take-up (i.e. fixed take-up); the designer will find that when conveyor is started/running; the belt tension increases at drive pulley, where as tensions at other non-drive pulleys mostly decreases! So, at non-drive pulley of such conveyor, there is no addition in load due to sudden application etc. during starting. It is rather reduction! Accordingly, there is multiplier of less magnitude for pulleys of such conveyors. In case of belt conveyor, the load situation is not as clear as water bucket and stool, but it is somewhat blurred. **(Refer addendum for other information.)**

Chapter 16: CONVEYOR FRAME

Shiftable conveyors application

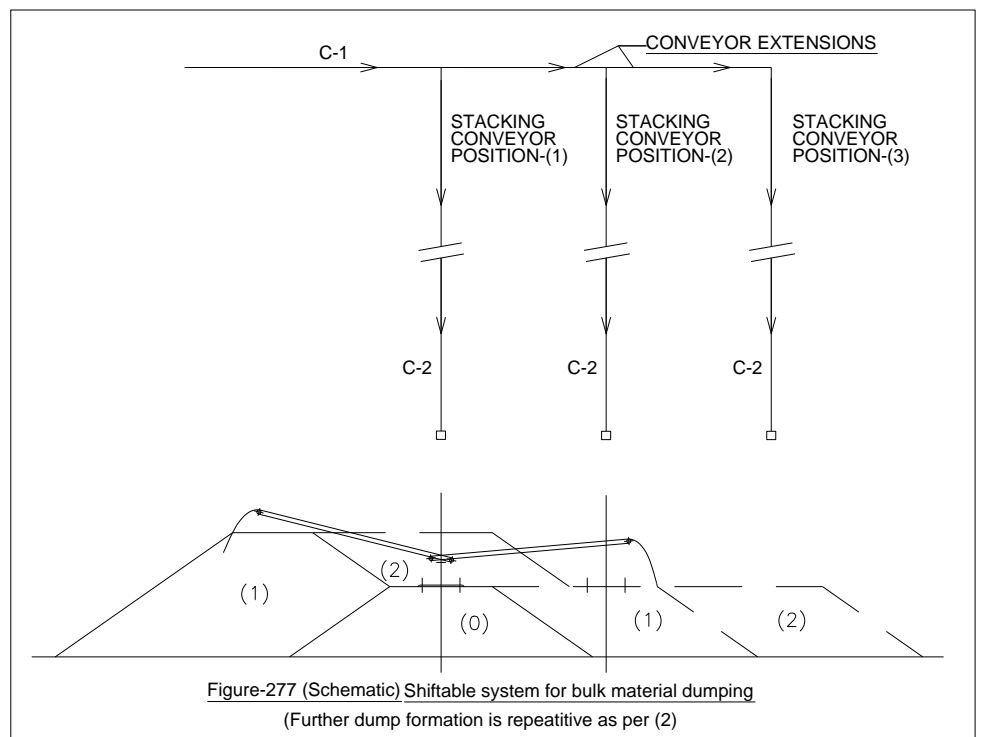
The development of shiftable conveying system alongwith the associated equipment took place in Germany, who are world leader in this field. Their experience in design and operation of such system is the original source for development all over the world.

Two typical areas of application have been narrated here which explain the need and advantage of shiftable conveying system in certain situation. These are typical normal cases, showing the underlying basic principles in simplistic form.

Use of shiftable conveyor system for creating dump of waste bulk material

Fig-277 schematically shows the use of shiftable conveyors to form waste material dump in a area of 1 km². Just two conveyors of 1.2 km and 1 km length can form dump of 1 km x 1 km x 25 m to accommodate 25 million cubic meter of material!

This is possible because the shiftable conveyor enables it to pass/shift over each and every point in 1 km² area, to dump material. The reader will observe that the creation of dump of such magnitude by other means will be clumsy and expensive.



Conveyor C-1 receives feed of waste bulk material from left direction. Initially a ramp as marked by (0) is formed by this system itself or by other mechanical means or is simply chosen as a natural ground feature. Use of natural hill-line is quite common, if available. Conveyor C-2 is installed on this ramp. This conveyor is equipped with rail-mounted spreader or rail-mounted tripper together with crawlers mounted spreader with slewing boom. The dump formations marked '1' is one time formation. The formation marked '2' is repetitive till the full area is covered.

As can be seen:

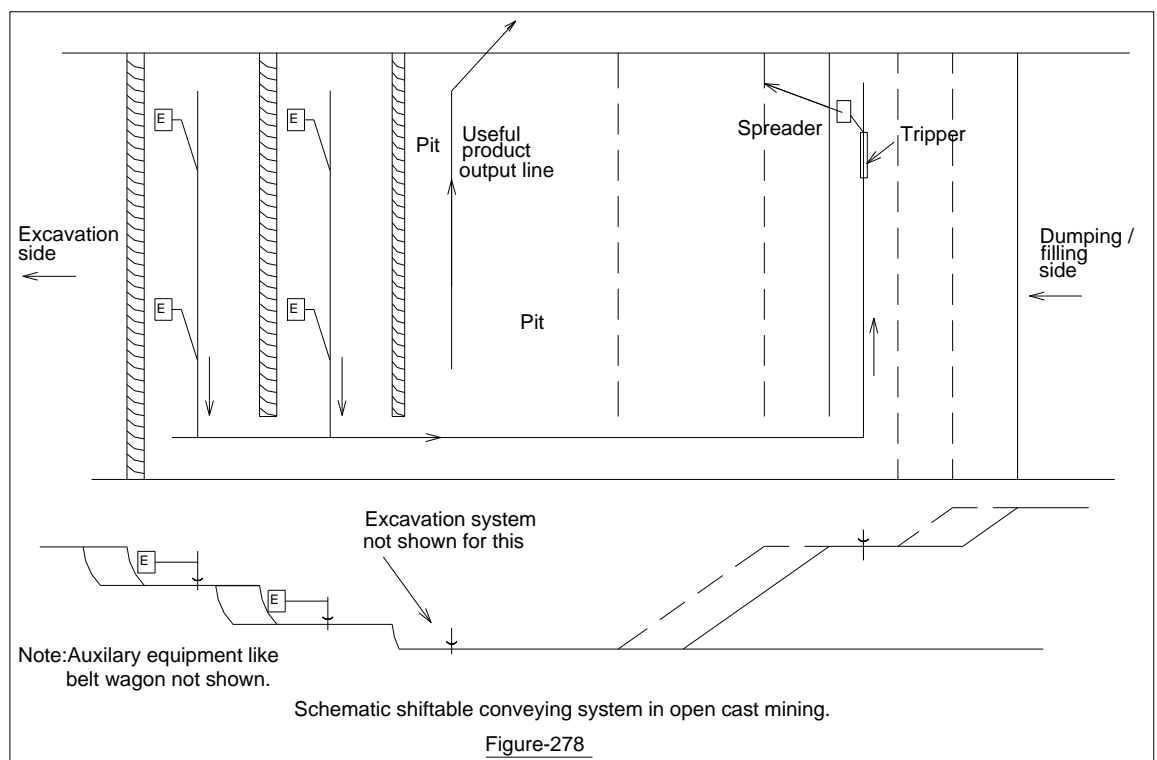
- The system is operated such that the dump surface becomes working surface for next step of operation (for particular installation level).
- The spreader can be utilised to create dump for down-level as well as up-level, for lesser frequency of shifting.

- The figure does not show intricate details such as auxiliary equipment like belt wagon, etc, which are necessary to dump material at the ends of conveyor C-2.
- Conveyor C-1 gets extended, as the dump progresses. C-2 gets shifted, as the dump progresses.

Use of shiftable conveyor system in open cast mine

Figure-278 schematically shows the use of shiftable conveying system at open cast mine. Initially, an appropriate pit is formed by the system itself and/or by other mechanical means. The shiftable system is placed in working position, in aforesaid pit as shown in figure. The useful product (coal, lignite, mineral) is few tens of meter beneath the overburden (earthen material overlying the useful product is called overburden).

The overburden is excavated / removed from one side of pit to expose fresh deposit of product. The product is removed by separate shiftable system (this is not shown in the figure, because it has only excavation line, without re-filling). The overburden being removed from one side of pit is refilled on opposite side of pit. Thus, once the pit is formed, new dumping area is not needed for continuous working of mine. In effect, the pit expands on excavation side and contracts on fill-side, appearing to be slowly moving in the direction of excavation (E stands for an excavator machine). As can be seen:



- The system is operated such that the newly excavated surface automatically becomes working surface for the next step of operation for excavation.
- On the dump side, the system is operated such that the newly formed dump surface automatically becomes working surface for the next step of filling operation.
- The material swells after excavation, and therefore, the fill total height will be more than excavation depth. This will be appearing as a hill-line on area already mined.

Chapter 17: SUPER STRUCTURE

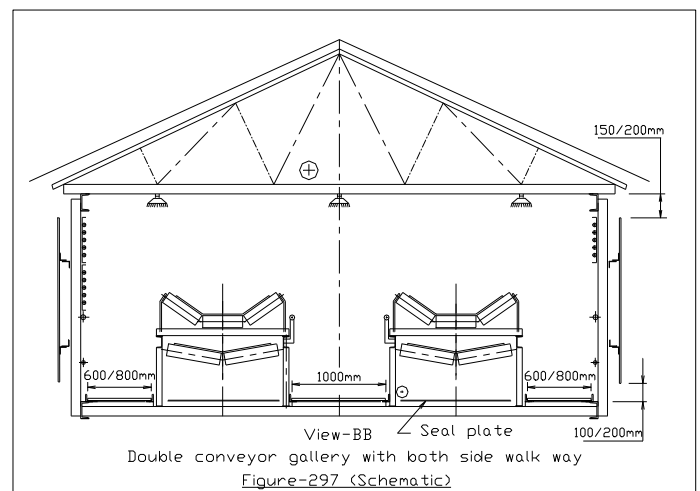
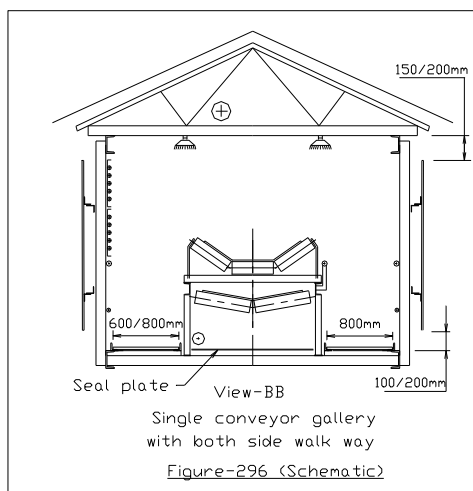
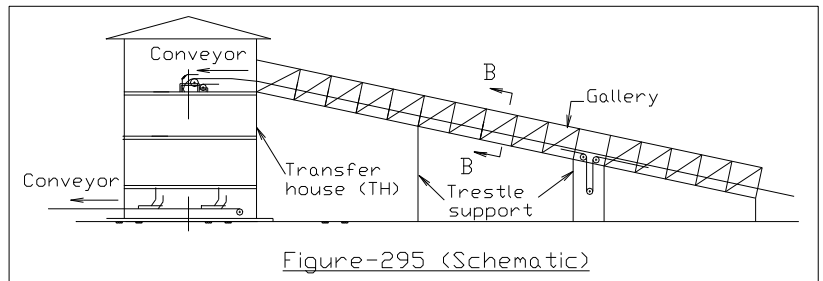
Gallery (Closed type gantry)

Gallery is an alternative method to gantry for supporting elevated length of conveyor. Gallery is also a braced type structure, made mostly from angles and some channels. The gallery clear height is minimum 2200 mm, and hence it has comparatively longer span (span is proportional to structure height / depth). Generally, the span can range from 18 m to 27 m, however, gallery of any span is feasible. The gallery span is selected such that total tonnage of gallery plus trestles is minimum. Figure-295 shows conveyor within gallery. Different cross sectional arrangements for gallery are shown in figure-296 to figure-299.

The gallery is clad on roof and sides by corrugated sheets. The corrugated sheets can be galvanized iron type or special coated iron type or even aluminum type.

Asbestos cement sheets had

been widely used; however, presently these are under statutory restrictions in many countries. Translucent fiberglass sheets in place of normal sheets, at suitable interval are provided for natural light. Continuous opening can be provided at top and bottom of side cladding for natural thermal ventilation. Such opening height depends upon climate condition whether the installation is in hot country or cold country.



Walkways are fitted with grating or chequered plates. The grating is self-cleaning in nature, however, it does not prevent fall of small size material below conveyor, if same is objectionable. The walkway with anti-skid feature is desirable for conveyor of inclination more than 5°. In case of gratings, the anti-skid feature can be part of grating construction. For chequered plate walkway, the anti-skid feature can be in the form of 10 mm dia twisted reinforcement steel bar welded on chequered plate, along walkway width, at every 500 mm interval. Stepped walkway is preferred for inclination more than 10°. Necessary handrails of galvanized steel tubes are provided along walkway.

2.5 mm / 3.0 mm thick steel sheet bottom plate (seal plate) can be provided below the conveyor-frame. The seal plate, together with walkway, will enclose the gallery bottom for full width, to give protection to underneath area from falling material/objects.

Chapter 19: INTRODUCTORY INFORMATION ON EQUIPMENT ASSOCIATED WITH BELT CONVEYOR

Stockyard machines

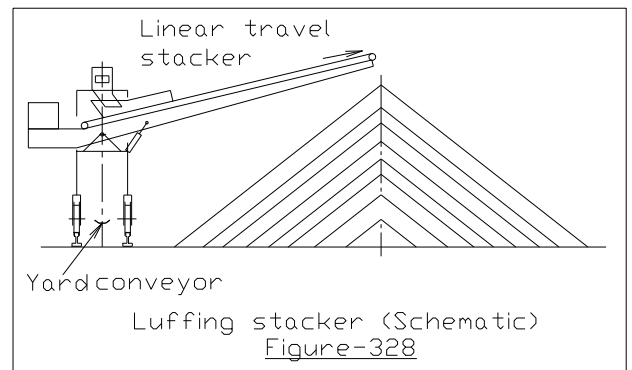
The material arriving at plant can not be consumed immediately. The material arrival time will be much shorter compared to consumption time. Therefore, stackers are needed to create stock-piles of bulk material.

The material so stored needs to be retrieved i.e. reclaimed to suit gradual consumption in plant. Therefore, store-yard needs reclaimer machines to reclaim material

The stackers as well as reclaimers are big machines, weighing few hundred tonnes, depending upon conveying capacity and width of stockpile. Mostly, such machines are mounted on rails. The weight of machine is shared by large numbers of travelling wheels, in accordance with permissible wheel load. The wheel support arrangement is designed for determinate load on each wheel, even with certain unevenness in rail track. 3-point in space determines the specific plane, and therefore, the machine body is supported on wheel assemblies, as '3-point support' to avoid twisting of structure. 3-point support can have 4-corner wheel assemblies. The design ensures machine stability in all applicable conditions. The design of these machines is a highly specialised engineering subject. The reader is advised to refer to [TransTech publication "Stacking Blending Reclaiming of Bulk Materials"](#) for specific information on the subject matter.

Stackers

The stacker includes an enlarged tripper and main body. The main body includes stacking boom conveyor. The stationary yard conveyor passes through tripper and discharges material on to boom conveyor. The boom conveyor discharges on storage yard to form stock-pile, while machine is travelling to-and-fro or in steps. This forms linear stock-pile on storage yard. The boom conveyor can be only luffing type or luffing



as well as slewing type. The stockpile is formed parallel to rails, on one side or both sides. The typical arrangement is shown in figure-328.

Reclaimers

There are wide varieties of reclaimers to suit specific need i.e. function and price. Primarily, two types of reclaimers have wide spread use viz. scraper reclaimers and bucket wheel reclaimers. The each type has varied designs to suit application. Only main / popular versions have been mentioned here.

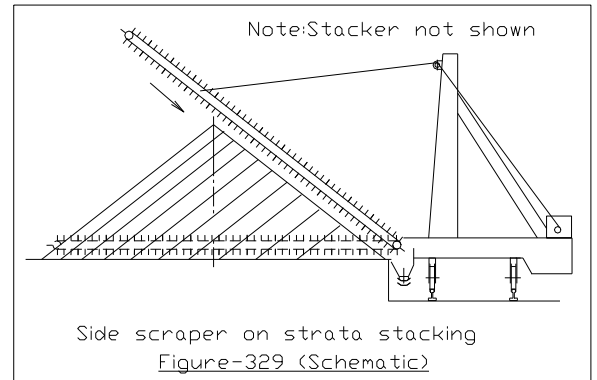
The scraper machine has double strand chain, running on chain guides. The scraper blades (toothed rectangular steel plates) are attached with double strand chain, at regular interval. The travel of chain along with scraper blades, scrapes / drags the material for discharge on to yard conveyor.

The bucket wheel reclaimer machine has a reclaiming wheel, to which scooping buckets are attached. The rotating wheel is made to scoop the material from stock-pile cross-sectional face. The material so scooped by the buckets flows / falls onto boom conveyor placed near center of bucket wheel. The boom (or receiving) conveyor discharges the material onto yard conveyor.

The scraper reclaimer machine is lighter in weight, less in price and has better blending efficiency. More numbers of components (chain links, pins, guides and scraper blades) are subjected to material abrasion. Where as, in case of bucket wheel reclaimer, only bucket edges are subjected to abrasion during reclaiming operation, which can be easily constructed for strength, wear resistance and long term replacement. Therefore, in general, bucket wheel reclaimers are meant for heavier duty. These machines are heavier and costlier.

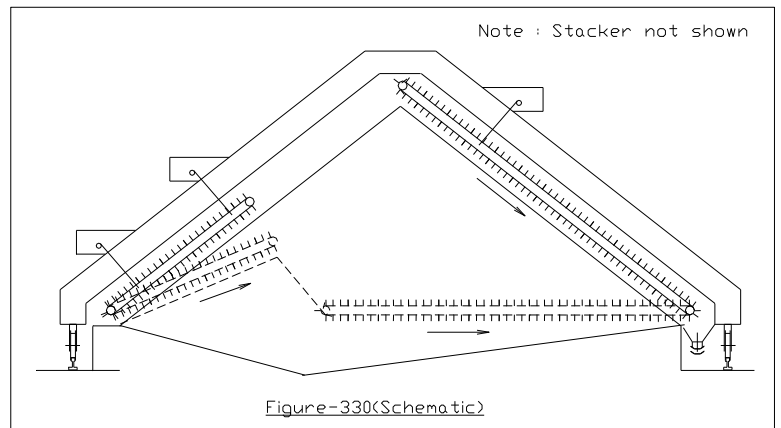
Side scraper reclaimer

The machine operates on rail track, located on one side of stockpile, as shown in figure-329. The machine reclaims material from longitudinal face of stock-pile, as it travels to-and-fro. The boom is precisely lowered at the end of each travel pass to get depth of cut, and for continuity of reclaiming operation. The machine rides on / over stock-pile and hence, it can have access to any of the stock-piles in series.



Portal scraper reclaimer

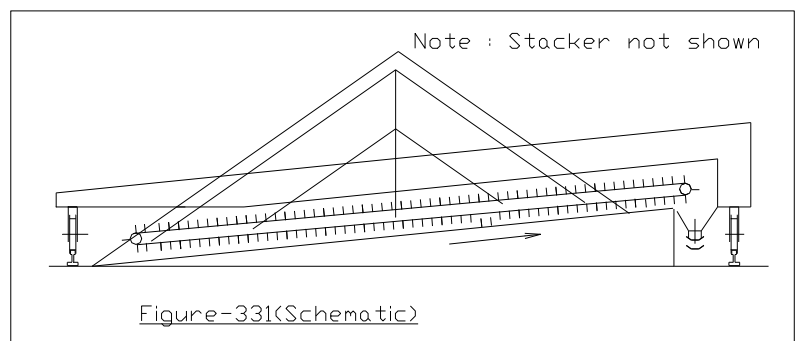
The portal scraper reclaimer also reclaims the material from longitudinal side face/s, and in this context, it is a side scraper. However, scraper boom being mounted from portal, it can suit very wide base of stockpile. The machine arrangement is shown in figure-330.



The machine reclaims material from longitudinal face of stock-pile, as it travels to-and-fro. The boom is precisely lowered at the end of each travel pass to get depth of cut, and for continuity of reclaiming operation. The machine rides on / over stockpile and hence, it can have access to any of the stock-piles in series. The machine can have two booms in series or two parallel booms for bigger capacity.

Bridge type scraper-reclaimer

The machine arrangement is shown in figure-331. The scraper conveyor is at the base of stockpile. The machine reclaims the stockpile across the full cross-section,



and hence, it has best possible blending effect. The material flow, from stockpile cross-section face to the scraper conveyor at the base, is consistently maintained by oscillating rake/rope system (to assist gravity flow in more positive and steady manner). The machine moves at a slow steady speed into stockpile face during reclaiming operation.

Bucket wheel on slewing boom reclaimer

The machine has slewing boom. The boom is supported at machine center by large diameter anti-friction slew bearing. The bucket wheel is at far end of the boom. This is an excellent reclaiming machine, but blending effect is marginal. The machine travel motion is in steps, to create depth of cut into stock-pile face. The reclaiming operation comprises of travel step, followed by slewing pass, followed by travel step in a repetitive manner, for continuous and automatic reclaiming of selected bench. The reclaiming of material primarily occurs during slewing motion of the boom. The bucket wheel rotates at steady speed, whereas the slewing speed varies automatically during slewing pass. The slewing speed is least when boom is nearly parallel to yard conveyor. The slew speed is highest when boom angle with yard conveyor is nearly perpendicular. The boom operation angle to yard conveyor is generally limited up to 75°. The typical arrangement of machine is shown in figure-332.

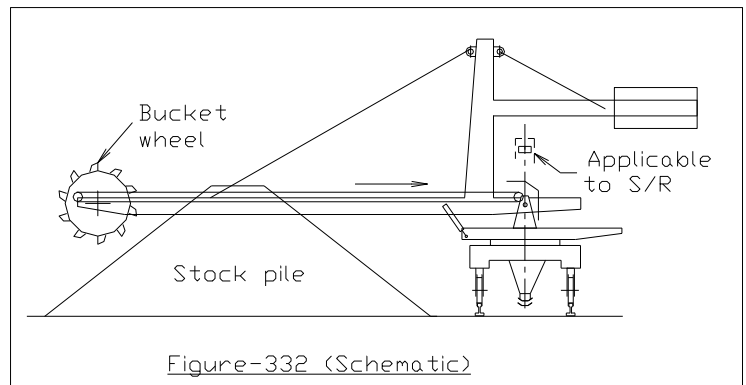


Figure-332 (Schematic)

The boom operation angle to yard conveyor is generally limited up to 75°. The typical arrangement of machine is shown in figure-332.

Bucket wheel on bridge

The machine has horizontal bridge supported on rail track spanning the stockpile width. The rotating bucket-wheel/s make to-and-fro linear travel along bridge. The machine moves at steady slow speed into stockpile face. The to-and-

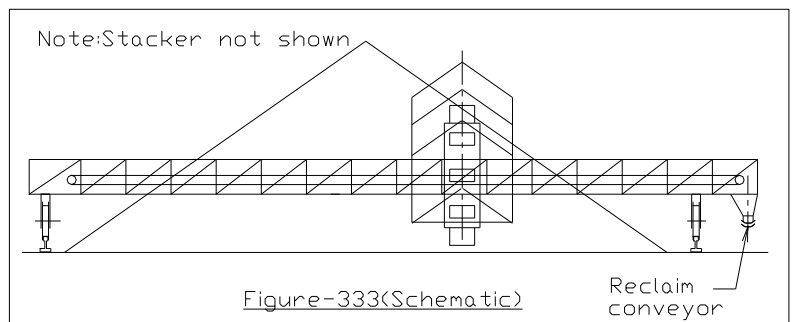


Figure-333(Schematic)

fro motions of rotating bucket wheel and steady advancement of machine results in to reclaiming operation. This is reclaiming-cum-blending machine for heavy-duty application. The typical arrangement of machine is shown in figure-333.