Belt Drives and Chain Drives

Material comes for Mott, 2002 and Kurtz, 1999

Power Train

- A power train transmits power from an engine or motor to the load.
- Some of the most common power trains include:
  - Flexible members
    - V-belt
    - Synchronous belt
    - Roller-chain

Power Train

  - Rigid members
    - Gearing
    - Drive shafts
- Other Components
  - Shafts
  - Bearings
  - Springs
  - Seals
  - Universal joints
  - Clutches
  - Brakes
  - Flywheels

Belt Drive

- Belts
  - Flat: use pulley
  - Grooved: use sheaves
- Idler
  - Does not transmit power
  - Used to take up slack or change direction of rotation

Key Belt Drive Design Characteristics

1. Environment
2. Load cycles
3. Service life
4. Belt characteristics
5. Sheave diameter and Center distance
6. Power requirements

Belt Drives

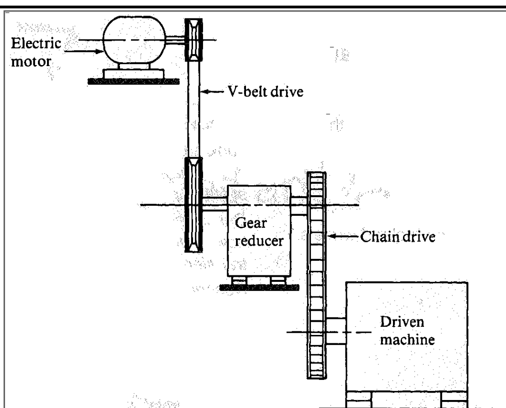
- Rotary power is created by electric motor, combustion engines, wind mills, and etc.
- Motors generally operate too fast and deliver too low a torque to be appropriate for the final drive application.
- The torque is increased in proportion to the amount that rotational speed is reduced.
- The high speed of the motor makes belt drives good for the first stage of reduction.

Belt Drives con't**

- A smaller drive pulley (sheave) is attached to the motor shaft, while a larger diameter pulley is attached to a parallel shaft that operates at a correspondingly lower speed.
- If very large ratios of speed reduction are required in the drive, gear reducers are desirable because they can typically accomplish large reductions in a rather small package.

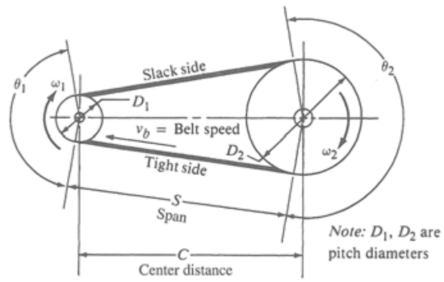
Belt Drives con't

- Gear reducers are only available at discrete reduction ratios, so the output may be adjusted before meeting the requirements of the machine.
- At the low-speed, high-torque condition, chain drives are desirable. The high torque causes high tensile forces to be developed in the chain.
- The links of the chain are engaged in toothed wheels (sprockets) to provide positive mechanical drive, needed at the low-speed, high-torque conditions.



(a) Sketch of combination drive

Basic Belt Drive Geometry



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Types of Belt Drives

- A belt is a flexible power transmission element that fits securely on a set of pulleys or sheaves.
- When the belt is used for speed reduction, the smaller sheave is mounted on the high-speed shaft, like the shaft of an electric motor. The larger sheave is then put on the driven machine.

Types of Belt Drives con't

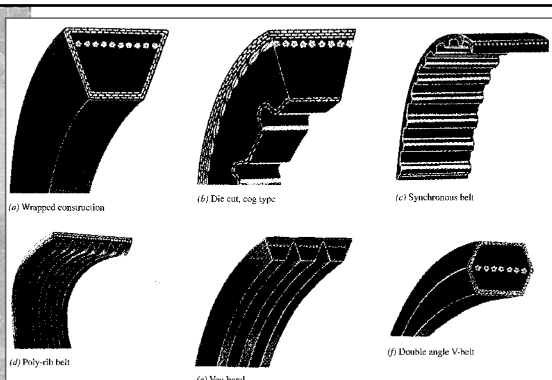
- Many types of belts are available:
  - The flat belt, the simplest type, is often made from leather or rubber-coated fabric.
  - The sheave surface is also flat and smooth, limiting the driving force by the pure fiction between the belt and the sheave.
  - Some designers prefer flat belts for delicate machinery because the belt slips if the torque rises to a high enough level to damage the machine.

Types of Belt Drives con’t

- Synchronous belts, or timing belts, ride on sprockets that have mating grooves that the teeth on the belt seat. This is a positive drive, limited only by the tensile strength of the belt and the shear strength of the teeth.
- Cog belts are applied to standard V-grooved sheaves. The cogs give the belt greater flexibility and higher efficiency compared with standard belts.

Types of Belt Drives con’t

- A common type of belt is the V-belt drive.
- The V-shape causes the belt to wedge tightly into the groove, increasing friction and allowing high torques to be transmitted before slipping occurs.
- Most belts have high-strength cords positioned at the pitch diameter of the belt cross section to increase the tensile strength of the belt.



E 7–3 Examples of belt construction (Dayco Corp., Dayton, OH)

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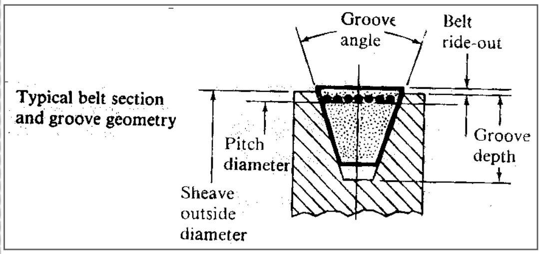
Cross Section of V-belt and Sheave Groove



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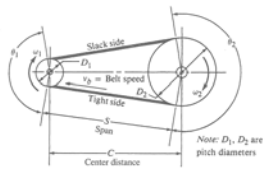
V-belt Drives

1. The pulley, with a circumferential groove carrying the belt, is called a sheave.
2. The size of a sheave is indicated by its pitch diameter, slightly smaller than the outside diameter.
3. The speed ratio between the driving and the driven sheaves is inversely proportional to the ratio of the sheave pitch diameters.

Basic Belt Drive Geometry

Basic Belt Drive Geometry

- Power
  - HP=TN/33000
  - HP1 = HP2
- Torque
  - T1/T2 = D1/D2
- Speed
  - N1/N2 = D2/D1



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V-belt Drives

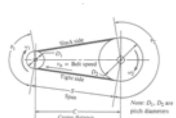
- There is no slipping under normal loads. The linear speed of the pitch line of both sheaves is the same as and equal to the belt speed, v_b.
  - $v_b = R_1\omega_1 = R_2\omega_2$
    - But $R_1 = D_1/2$ and $R_2 = D_2/2$
  - $v_b = D_1\omega_1 / 2 = D_2\omega_2 / 2$
- The angular velocity is: $\omega_1/\omega_2 = D_2/D_1$

V-belt Drives con’t

4. The relationships between pitch length, L, center distance, C, and the sheave diameters are:
   - $L = 2C + 1.57(D_2 + D_1) + [(D_2 - D_1)^2 / 4C]$

$$\mathbf{C} = \frac{\mathbf{B} + \sqrt{\mathbf{B}^2 - 32(\mathbf{D}_2 - \mathbf{D}_1)^2}}{16}$$

Where $B = 4L - 6.28(D_2 + D_1)$



V-belt Drives con't

5. The angle of contact of the belt on each sheave is:

$$\theta_1 = 180^o - 2\sin^{-1}[(D_2 - D_1)/2C]$$

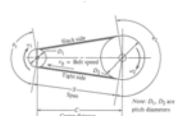
$$\theta_2 = 180^o + 2\sin^{-1}[(D_2 - D_1)/2C]$$

- These angles are important because commercially available belts are rated with an assumed contact angle of 180^o, requiring a lower power rating.

V-belt Drives con't

6. The length of the span between two sheaves, over which the belt is unsupported, is

$$S = \sqrt{C^2 - [(D_2 - D_1)/2]^2}$$



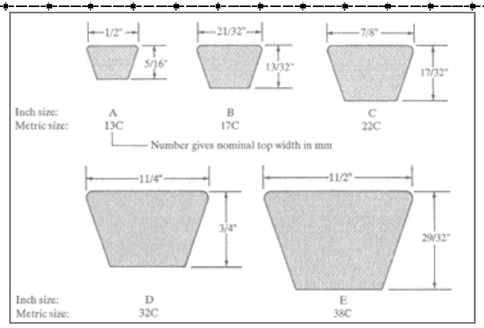
V-belt Drives con't

7. The contributors to the stress in the belt are: (a) the tensile force in the belt, maximum on the tight side of the belt; (b) the bending of the belt around the sheaves, maximum as the tight side of the belt bends around the smaller sheave; (c) centrifugal forces created as the belt moves around the sheaves.
   - The maximum total stress occurs where the belt enters the smaller sheave, and the bending stress is a major part.
8. The design value of the ratio of the tight side tension to the slack side tension is 5.0 for V-belt drives.

Standard Belt Cross Sections

- Commercially available belts are made to one of the standards shown on the next slide.
- The nominal value of the included angle between the sides of the V-groove ranges from 30^o to 42^o. In order to achieve a tight fit in the groove, the angle on the belt may be slightly different. Some belts are designed to slip out of the groove somewhat.

Heavy-duty Industrial V-belts



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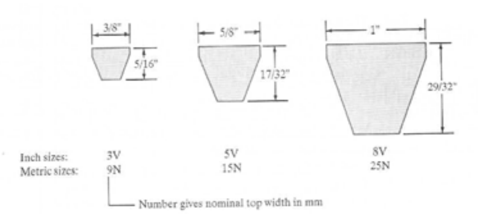
Industrial Narrow-section V-belts



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Power Transmitted

- Unique for belt design and construction

TABLE 8.1 COMMON V-BELT CROSS SECTIONS

Belt Type	SI Designation	US Designation
Classical V-Belts/Sheaves	13C,16C,22C,32C	A,B,C,D
Double-V (Hexagonal) Belts		AA,BB,CC,DD
Narrow Multiple V-Belts	9N,15N,25N	3V,5V,8V
Light-Duty Single V-Belts		2L,3L,4L,5L
V-Ribbed Belts		H,J,K,L,M

Kurtz, 2000, Machine Design for Mobile Applications

Power Transmitted con't

- An equation for a system with 180^o with contact on the smallest sheave.

$$\mathbf{P} = \mathbf{d} * \mathbf{r}[\mathbf{K}_1 - \frac{\mathbf{K}_2}{\mathbf{d}} - \mathbf{K}_3(\mathbf{dr})^2 - \mathbf{K}_4 \log(\mathbf{dr})] + \mathbf{K}_2\mathbf{r}$$

- P = power
- d = pitch diameter of small sheave
- r = RPM / 1000
- K_1, K_2, K_3, K_4 = coefficients

Power Transmitted con't

TABLE 8.2 CLASSICAL V-BELT POWER RATING COEFFICIENTS

Section SI (US)	k_1	k_2	k_3	k_4
13C(A)	0.0382(1.004)	1.232(1.652)	7.04E-09(1.55E-04)	6.244E-03(0.213)
13CX(AX)	0.0585(1.462)	1.482(2.239)	1.00E-08(2.20E-04)	1.192E-02(0.424)
16C(B)	0.0678(1.769)	3.261(4.372)	1.40E-08(3.08E-04)	1.074E-02(0.366)
16CX(BX)	0.0839(2.051)	2.635(3.532)	1.41E-08(3.10E-04)	1.684E-02(0.573)
22C(C)	0.2161(3.325)	9.004(12.07)	2.65E-08(5.83E-04)	2.022E-02(0.689)
22CX(CX)	0.1317(3.272)	4.965(6.655)	2.41E-08(5.30E-04)	2.537E-02(0.864)
32C(D)	0.2703(7.160)	32.23(43.21)	6.30E-08(1.38E-03)	4.270E-02(1.454)

Note: SI units are outside parentheses; traditional units are within parentheses.

Kurtz, 2000, Machine Design for Mobile Applications

V-belt Forces

- Through mathematical development
- Torque acting on driver:
  - $T_R = (T_1 - T_2)\, D_R / 2$ (N*mm)
  - D_R = driver sheave diameter (mm)
- Driven sheave torque
  - $T_N = (T_1 – T_2)D_N/2$ (N*mm)
  - D_N = driven sheave diameter (mm)
- T_1 = tight side (N)
- T_2 = slack side (N)

Effective Pull

- Pull: $(T_1 – T_2) = 1000P / V$
  - P = transmitted power; kW
  - V = belt velocity; m/s
- Power: $P = (T_1 – T_2)\, V/ 1000$

Effective Pull

Allowable Tension Ratio

- For 180 degree contact $R_A = T_1 / T_2$
- Allowable Tension Ratio
  - 5 to 1 at 180 degree arc of contact for v-belt on sheave
  - 2.5 to 1 for 180 degree arc of contact for v-belt on flat pulley
- At other angles $R_A = \exp[(0.5123)\, \theta\, (\pi/180)]$
  - θ = arc of contact on sheave or pulley

V-belt Drive Design

- The basic data required for drive selection are:
  - The rated power of the driving motor or other prime motor
  - The service factor based on the type of driver and driven load
  - The center distance
  - The power rating for one belt as a function of the size and speed of the smaller sheave
  - The belt length
  - The size of the driving and driven sheaves
  - The correction factor for belt length
  - The correction factor for the angle of wrap on the smaller sheave
  - The number of belts
  - The initial tension on the belt

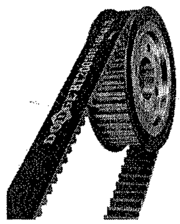
Belt Tension

- The initial tension given to a belt ensures that the belt will not slip under the design load.
- At rest, the two sides of the belt have the same tension. As power is being transmitted, the tension in the tight side increases while the tension in the slack side decreases. Without the initial tension, the slack side would go totally loose.

Synchronous Belt Drives

- Synchronous belts are made with ribs or teeth across the bottom of the belt. The teeth mate with corresponding grooves in the driving and driven pulleys (sprockets), providing a positive drive without slippage.
- There is a fixed relationship between the speed of the driver and the speed of the driven sprocket. For this reason, synchronous belts are also known as timing belts.

FIGURE 7–16
Synchronous belt on driving sprocket (Copyright Rockwell Automation, used by permission)



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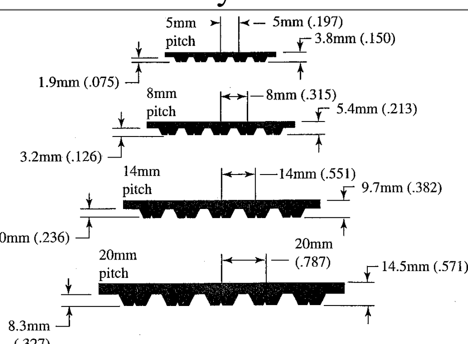
Dimensions of Synchronous Belt



Numbers in parentheses are inches

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Example Problem 7–1 Design a V-belt drive that has the input sheave on the shaft of an electric motor (normal torque) rated at 50.0 hp at 1160-rpm. full-load speed. The drive is to a bucket elevator in a potash plant that is to be used 12 hours (h) daily at approximately 675 rpm.

Solution Objective: Design the V-belt drive.

Given:

Power transmitted = 50 hp to bucket elevator

Speed of motor = 1160 rpm; output speed = 675 rpm

Analysis: Use the design data presented in this section. The solution procedure is developed within the Results section of the problem solution.

Results: ***Step 1.*** Compute the design power. From Table 7–1, for a normal torque electric motor running 12 h daily driving a bucket elevator, the service factor is 1.40. Then the design power is 1.40(50.0 hp) = 70.0 hp.

Driven machine type	<6 h per day	6–15 h per day	>15 h per day	<6 h per day	6–15 h per day	>15 h per day
Agitators, blowers, fans, centrifugal pumps, light conveyors	1.0	1.1	1.2	1.1	1.2	1.3
Generators, machine tools, mixers, gravel conveyors	1.1	1.2	1.3	1.2	1.3	1.4
Bucket elevators, textile machines, hammer mills, heavy conveyors	1.2	1.3	1.4	1.4	1.5	1.6
Crushers, ball mills, hoists, rubber extruders	1.3	1.4	1.5	1.5	1.6	1.8
Any machine that can choke	2.0	2.0	2.0	2.0	2.0	2.0

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Step 2. Select the belt section. From Figure 7–9, a 5V belt is recommended for 70.0 hp at 1160-rpm input speed.

Step 3. Compute the nominal speed ratio:

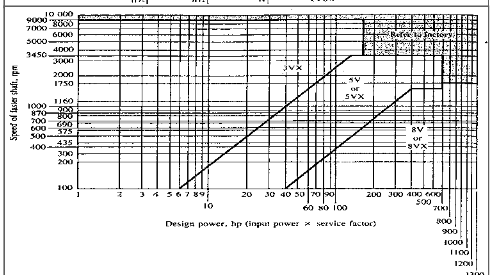
$$\text{Ratio} = 1160/675 = 1.72$$

Step 4. Compute the driving sheave size that would produce a belt speed of 4000 ft/min, as a guide to selecting a standard sheave:

$$\text{Belt speed} = v_b = \frac{\pi D_1 n_1}{12} \text{ft/min}$$

Then the required diameter to give $v_b = 4000$ ft/min is

$$D_1 = \frac{12 v_b}{\pi n_1} = \frac{12(4000)}{\pi n_1} = \frac{15\,279}{n_1} = \frac{15\,279}{1160} = 13.17 \text{ in}$$



Step 5. Select trial sizes for the input sheave, and compute the desired size of the output sheave. Select a standard size for the output sheave, and compute the actual ratio and output speed.

For this problem, the trials are given in Table 7–3 (diameters are in inches).

The two trials in **boldface** in Table 7–3 give only about 1% variation from the desired output speed of 675 rpm, and the speed of a bucket elevator is not critical. Because no space limitations were given, let's choose the larger size.

TABLE 7–3 Trial sheave sizes for Example Problem 7–1

Standard driving sheave size, D_1	Approximate driven sheave size ($1.72D_1$)	Nearest standard sheave, D_2	Actual output speed (rpm)
13.10	22.5	21.1	720
12.4	**21.3**	**21.1**	**682**
11.7	20.1	21.1	643
10.8	18.6	21.1	594
10.2	17.5	15.9	744
9.65	16.6	15.9	704
9.15	**15.7**	**15.9**	**668**
8.9	15.3	14.9	693

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Step 6. Determine the rated power from Figure 7–10, 7–11, or 7–12.

For the 5V belt that we have selected, Figure 7–11 is appropriate. For a 12.4-in sheave at 1160 rpm, the basic rated power is 26.4 hp. Multiple belts will be required. The ratio is relatively high, indicating that some added power rating can be used. This value can be estimated from Figure 7–11 or taken directly from Figure 7–13 for the 5V belt. Power added is 1.15 hp. Then the actual rated power is 26.4 + 1.15 = 27.55 hp.

Step 7. Specify a trial center distance.

We can use Equation (7–8) to determine a nominal acceptable range for C:

$$D_2 < C < 3(D_2 + D_1)$$

$$21.1 < C < 3(21.1 + 12.4)$$

$$21.1 < C < 100.5 \text{ in}$$

In the interest of conserving space, let's try $C = 24.0$ in.

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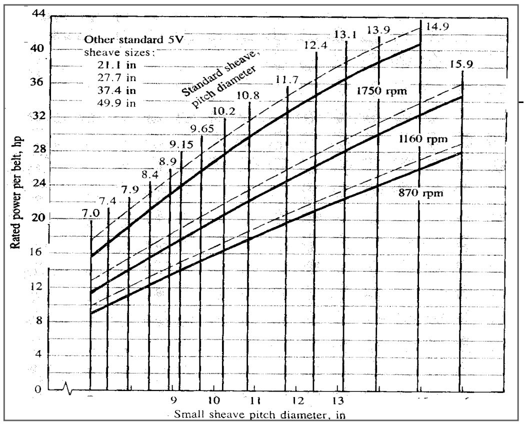

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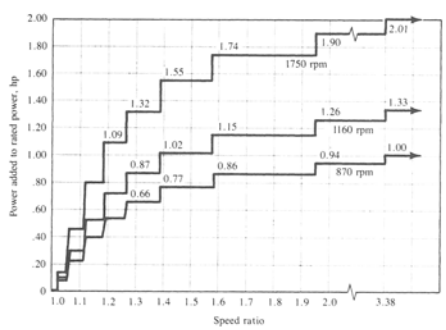

Step 8. Compute the required belt length from Equation (7–3):

$$L = 2C + 1.57(D_2 + D_1) + \frac{(D_2 - D_1)^2}{4C}$$

$$L = 2(24.0) + 1.57(21.1 + 12.4) + \frac{(21.1 - 12.4)^2}{4(24.0)} = 101.4 \text{ in}$$

Step 9. Select a standard belt length from Table 7–2, and compute the resulting actual center distance from Equation (7–4).

In this problem, the nearest standard length is 100.0 in. Then, from Equation (7–4),

$$B = 4L - 6.28(D_2 + D_1) = 4(100) - 6.28\,(21.1 + 12.4) = 189.6$$

$$C = \frac{189.6 + \sqrt{(189.6)^2 - 32(21.1 - 12.4)^2}}{16} = 23.30 \text{ in}$$

Step 10. Compute the angle of wrap of the belt on the small sheave from Equation (7–5):

$$\theta_1 = 180^\circ - 2\sin^{-1}\left[\frac{D_2 - D_1}{2C}\right] = 180^\circ - 2\sin^{-1}\left[\frac{21.1 - 12.4}{2(23.30)}\right] = 158^\circ$$

Step 11. Determine the correction factors from Figures 7–14 and 7–15. For $\theta = 158^\circ$, $C_\theta = 0.94$. For $L = 100$ in, $C_L = 0.96$

Step 12. Compute the corrected rated power per belt and the number of belts required to carry the design power:

$$\text{Corrected power} = C_\theta C_L P = (0.94)(0.96)(27.55 \text{ hp}) = 24.86 \text{ hp}$$

$$\text{Number of belts} = 70.0/24.86 = 2.82 \text{ belts (Use 3 belts.)}$$

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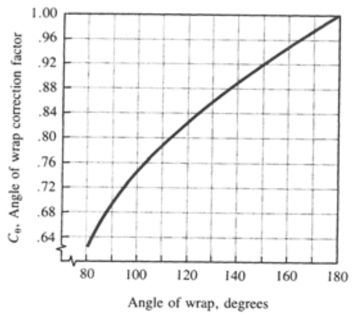

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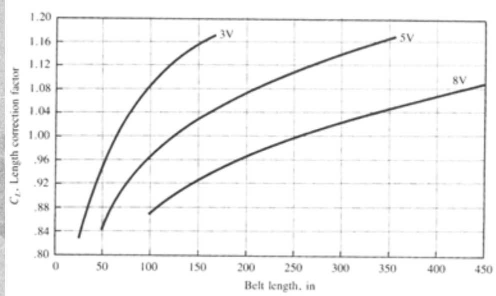

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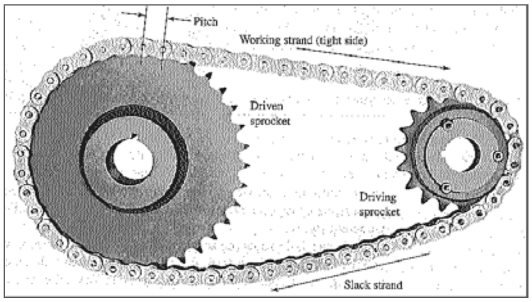
Chain Drives

- A chain is a power transmission element made as a series of pin-connected links. The design provides for flexibility while enabling the chain to transmit large tensile forces.
- When transmitting power between rotating shafts, the chain engages mating toothed wheels, called sprockets.

Roller Chain Drive



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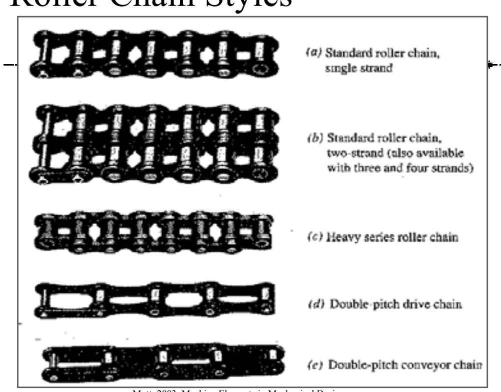
Chain Drives con't

- The most common type of chain is the roller chain, in which the roller on each pin provides exceptionally low friction between the chain and the sprockets.
- Roller chain is classified by its pitch, the distance between corresponding parts of adjacent links. The pitch is usually illustrated as the distance between the centers of adjacent pins.
- Standard roller chain carries a size designation from 40 to 240.

Roller Chain Styles



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Chain Drives con’t

- The digits (other than the final 0) indicate the pitch of the chain in eighths of an inch.
- For example, the no. 100 chain has a pitch of 10/8 or 1 ¼ in. A series of heavy-duty sizes, with the suffix H on the designation (60 H - 240 H), has the same basic dimension as the standard chain of the same number except for thicker side plates. In addition, there are smaller and lighter sizes: 25, 35, and 41.

Chain Drives con’t

- The average tensile strengths of the various chain sizes are listed in the next table. These data can be used for very low speed drives or for applications in which the function of the chain is to apply a tensile force or to support a load.
- It is recommended that only 10% of the average tensile strength be used in such applications.

TABLE 7–4 Roller chain sizes

Chain number	Pitch (in)	Roller diameter	Roller width	**Link plate thickness**	Average tensile strength (lb)
25	1/4	None	–	0.030	925
35	3/8	None		0.050	2100
41	1/2	0.306	0.250	0.050	2000
40	1/2	0.312	0.312	0.060	3700
50	5/8	0.400	0.375	0.080	6100
60	3/4	0.469	0.500	0.094	8500
80	1	0.626	0.625	0.125	14 500
100	$1\frac{1}{4}$	0.750	0.750	0.156	24 000
120	$1\frac{1}{2}$	0.875	1.000	0.187	34 000
140	$1\frac{3}{4}$	1.000	1.000	0.219	46 000
160	2	1.125	1.250	0.250	58 000
180	$2\frac{1}{4}$	1.406	1.406	0.281	80 000
200	$2\frac{1}{2}$	1.562	1.500	0.312	95 000
240	3	1.875	1.875	0.375	130 000

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Design of Chain Drives

- The rating of chain for its power transmission capacity considers 3 modes of failure:

1. Fatigue of the link plates due to the repeated application of the tension in the tight side of the chain.
2. Impact of the rollers as they engage the sprocket teeth.
3. Galling between the pins of each link and the bushings on the pins.

Design of Chain Drives con't

- The ratings are based on empirical data with a smooth driver and a smooth load (service factor =1.0) and with a rated life of approximately 15,000 h.
- Lubrication is critical to the satisfactory operation of a chain drive. Manufacturers recommend the type of lubrication method for given combinations of chain size, sprocket size, and speed.
- The next table lists the rated power for the No. 40 (1/2 in) size of standard chain.

TABLE 7–5 Horsepower ratings—single strand roller chain no. 40

0.500 inch pitch — Rotational speed of small sprocket, rev/min

No. of teeth	10	25	50	100	180	200	300	500	700	900	1000	1200	1400	1600	1800	2100	2500	3000	3500	4000	5000	6000	7000	8000	9000
11	0.06	0.14	0.27	0.52	0.91	1.00	1.48	2.42	3.34	4.25	4.70	5.60	6.49	5.57	4.66	3.70	2.85	2.17	1.72	1.41	1.01	0.77	0.61	0.50	0.00
12	0.06	0.15	0.29	0.56	0.99	1.09	1.61	2.64	3.64	4.64	5.13	6.11	7.09	6.34	5.31	4.22	3.25	2.47	1.96	1.60	1.15	0.87	0.69	0.57	0.00
13	0.07	0.16	0.31	0.61	1.07	1.19	1.75	2.86	3.95	5.02	5.56	6.62	7.68	7.15	5.99	4.76	3.66	2.79	2.21	1.81	1.29	0.98	0.78	0.00	
14	0.07	0.17	0.34	0.66	1.15	1.28	1.88	3.08	4.25	5.41	5.98	7.13	8.27	7.99	6.70	5.31	4.09	3.11	2.47	2.02	1.45	1.10	0.87	0.00	
15	0.08	0.19	0.36	0.70	1.24	1.37	2.02	3.30	4.55	5.80	6.41	7.64	8.86	8.86	7.43	5.89	4.54	3.45	2.74	2.24	1.60	1.22	0.97	0.00	
16	0.08	0.20	0.39	0.75	1.32	1.46	2.15	3.52	4.86	6.18	6.84	8.15	9.45	9.76	8.18	6.49	5.00	3.80	3.02	2.47	1.77	1.34	0.00		
17	0.09	0.21	0.41	0.80	1.40	1.55	2.29	3.74	5.16	6.57	7.27	8.66	10.04	10.69	8.96	7.11	5.48	4.17	3.31	2.71	1.94	1.47	0.00		
18	0.09	0.22	0.43	0.84	1.48	1.64	2.42	3.96	5.46	6.95	7.69	9.17	10.63	11.65	9.76	7.75	5.97	4.54	3.60	2.95	2.11	1.60	0.00		
19	0.10	0.24	0.46	0.89	1.57	1.73	2.56	4.18	5.77	7.34	8.12	9.66	11.22	12.64	10.59	8.40	6.47	4.92	3.91	3.20	2.29	0.09	0.00		
20	0.10	0.25	0.48	0.94	1.65	1.82	2.69	4.39	6.07	7.73	8.55	10.18	11.81	13.42	11.44	9.07	6.99	5.31	4.22	3.45	2.47	0.00			
21	0.11	0.26	0.51	0.98	1.73	1.91	2.83	4.61	6.37	8.11	8.98	10.69	12.40	14.10	12.30	9.76	7.52	5.72	4.54	3.71	2.65	0.00			
22	0.11	0.27	0.53	1.03	1.81	2.01	2.96	4.83	6.68	8.50	9.40	11.20	12.99	14.77	13.19	10.47	8.06	6.13	4.87	3.98	2.85	0.00			
23	0.12	0.28	0.56	1.08	1.90	2.10	3.10	5.05	6.98	8.89	9.83	11.71	13.58	15.44	14.10	11.19	8.62	6.55	5.20	4.26	3.05	0.00			
24	0.12	0.30	0.58	1.12	1.98	2.19	3.23	5.27	7.28	9.27	10.26	12.22	14.17	16.11	15.03	11.93	9.18	6.99	5.54	4.54	0.87	0.00			
25	0.13	0.31	0.60	1.17	2.06	2.28	3.36	5.49	7.59	9.66	10.69	12.73	14.76	16.78	15.98	12.68	9.76	7.43	5.89	4.82	0.00				
26	0.13	0.32	0.63	1.22	2.14	2.37	3.50	5.71	7.89	10.04	11.11	13.24	15.35	17.45	16.95	13.45	10.36	7.88	6.25	5.12	0.00				
28	0.14	0.35	0.67	1.31	2.31	2.55	3.77	6.15	8.50	10.82	11.97	14.26	16.53	18.79	18.94	15.03	11.57	8.80	6.99	5.72	0.00				
30	0.15	0.37	0.72	1.41	2.47	2.74	4.04	6.59	9.11	11.59	12.82	15.28	17.71	20.14	21.01	16.67	12.84	9.76	7.75	6.34	0.00				
32	0.16	0.40	0.77	1.50	2.64	2.92	4.31	7.03	9.71	12.38	13.68	16.30	18.89	21.48	23.14	18.37	14.14	10.76	8.54	1.41					
35	0.18	0.43	0.84	1.64	2.88	3.19	4.71	7.69	10.62	13.52	14.96	17.82	20.67	23.49	26.30	21.01	16.17	12.30	9.76	0.00					
40	0.21	0.50	0.96	1.87	3.30	3.65	5.38	8.79	12.14	15.45	17.10	20.37	23.62	26.85	30.06	25.67	19.76	15.03	0.00						
45	0.23	0.56	1.08	2.11	3.71	4.10	6.08	9.89	13.66	17.39	19.24	22.92	26.57	30.20	33.82	30.63	23.58	5.53	0.00						

Type A — Type B — Type C

Type A: Manual or drip lubrication
Type B: Bath or disc lubrication
Type C: Oil stream lubrication

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Design of Chain Drives con't

- Notice these features of the data:
  - The ratings are based on the speed of the smaller sprocket and an expected life of approximately 15,000 hours.
  - For a given speed, the power capacity increases with the number of teeth on the sprocket.
  - For a given sprocket size (given # of teeth), the power capacity increases with increasing speed up to a point; then it decreases. Fatigue due to the tension in the chain governs at the low to moderate speeds; impact on the sprockets governs at the higher speeds.

con't…

Design of Chain Drives con't

- The ratings are for a single strand of chain. Although multiple stands do increase the power capacity, they do not provide a direct multiple of the single-strand capacity. Multiply the capacity in the tables by the following factors:
  - Two stands: Factor = 1.7
  - Three strands: Factor = 2.5
  - Four strands: Factor = 3.3
- The ratings are for a service factor of 1.0. Specify a service factor for a given application according to the next table.

TABLE 7–8 Service factors for chain drives

Load type	Type of driver: Hydraulic drive	Electric motor or turbine	Internal combustion engine with mechanical drive
Smooth (agitators; fans; light, uniformly loaded conveyors)	1.0	1.0	1.2
Moderate shock (machine tools, cranes, heavy conveyors, food mixers and grinders)	1.2	1.3	1.4
Heavy shock (punch presses, hammer mills, reciprocating conveyors, rolling mill drive)	1.4	1.5	1.7

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Design Guidelines

1. The minimum number of teeth in a sprocket should be 17 unless the drive is operating at a very low speed, under 100 rpm.
2. The maximum speed ratio should be 7.0, although higher ratios are feasible. Two or more stages of reduction can be used to achieve higher ratios.
3. The center distance between the sprocket axes should be approximately 30 to 50 pitches (30 to 50 times the pitch of the chain).

Design Guidelines con't

4. The larger sprocket should normally have no more than 120 teeth.
5. The preferred arrangement for a chain drive is with the centerline of the sprockets horizontal and with the tight side on top.
6. The chain length must be an integral multiple of the pitch, and an even number of pitches is recommended. The center distance should be made adjustable to accommodate the chain length and to take up for tolerances and wear. Excessive sag on the slack side should be avoided, especially on drives that are not horizontal.

Design Guidelines con't

6. Con't… A convenient relation between center distance (C), chain length (L), number of teeth in the small sprocket (N_1), and the number of teeth in the large sprocket (N_2), expressed in pitched, is:

$$\mathbf{L} = 2\mathbf{C} + \frac{\mathbf{N}_2 + \mathbf{N}_1}{2} + \frac{(\mathbf{N}_2 - \mathbf{N}_1)^2}{4\pi^2 \mathbf{C}}$$

Design Guidelines con't

6. The center distance for a given chain length, again in pitches, is:

$$C = \frac{1}{4}[L - \frac{N_2 + N_1}{2} + \sqrt{[L - \frac{N_2 + N_1}{2}]^2 - \frac{8(N_2 - N_1)^2}{4\pi^2}}]$$

- The computed center distance assumes no sag in either the tight or the slack side of the chain, and thus it is a maximum. Negative tolerances or adjustments must be provided. Adjustment for wear must also be provided.

Design Guidelines con't

7. The pitch diameter of a sprocket with N teeth for a chain with a pitch of p is:

$$D = \frac{p}{\sin(180^o / N)}$$

8. The minimum sprocket diameter and therefore the minimum number of teeth in a sprocket are often limited by the size of the shaft on which it is mounted. Check the sprocket catalog.

Design Guidelines con't

9. The arc of contact, θ_1, of the chain on the smaller sprocket should be greater than 120^o.

$$\theta_1 = 180^o - 2\sin^{-1}[(D_2 - D_1)/2C]$$

10. For reference, the arc of contact, θ_2, on the larger sprocket is:

$$\theta_2 = 180^o + 2\sin^{-1}[(D_2 - D_1)/2C]$$

Lubrication

- It is essential that adequate lubrication be provided for chain drives. There are numerous moving parts within the chain, along with the interaction between the chain and the sprocket teeth. The designer must define the lubricant properties and the method of lubrication.

Example Problem 7–2 Design a chain drive for a heavily loaded coal conveyor to be driven by a gasoline engine through a mechanical drive. The input speed will be 900 rpm, and the desired output speed is 230 to 240 rpm. The conveyor requires 15.0 hp.

Solution Objective Design the chain drive.

Given Power transmitted = 15 hp to a coal conveyor

Speed of motor = 900 rpm; output speed range = 230 to 240 rpm

Analysis Use the design data presented in this section. The solution procedure is developed within the Results section of the problem solution.

Results ***Step 1.*** Specify a service factor and compute the design power. From Table 7–8, for moderate shock and a gasoline engine drive through a mechanical drive, $SF = 1.4$.

$$\text{Design power} = 1.4(15.0) = 21.0\ \text{hp}$$

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TABLE 7–8 Service factors for chain drives

	Type of driver		
Load type	Hydraulic drive	Electric motor or turbine	Internal combustion engine with mechanical drive
Smooth (agitators; fans; light, uniformly loaded conveyors)	1.0	1.0	1.2
Moderate shock (machine tools, cranes, heavy conveyors, food mixers and grinders)	1.2	1.3	1.4
Heavy shock (punch presses, hammer mills, reciprocating conveyors, rolling mill drive)	1.4	1.5	1.7

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Step 2. Compute the desired ratio. Using the middle of the required range of output speeds, we have

$$\text{Ratio} = (900\ \text{rpm})/(235\ \text{rpm}) = 3.83$$

Step 3. Refer to the tables for power capacity (Tables 7–5, 7–6, and 7–7), and select the chain pitch. For a single strand, the no. 60 chain with $p = 3/4$ in seems best. A 17-tooth sprocket is rated at 21.96 hp at 900 rpm by interpolation. At this speed, type B lubrication (oil bath) is required.

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TABLE 7–6 Horsepower ratings—single strand roller chain no. 60

No. of teeth	0.750 inch pitch											Rotational speed of small sprocket, rev/min											
	10	25	50	100	120	200	300	400	500	600	800	1000	1200	1400	1600	1800	2000	2500	3000	3500	4000	4500	5000
11	0.19	0.46	0.89	1.72	2.05	3.35	4.95	6.52	8.08	9.63	12.69	15.58	11.85	9.41	7.70	6.45	5.51	3.94	3.00	2.38	1.95	1.63	1.39
12	0.21	0.50	0.97	1.88	2.24	3.66	5.40	7.12	8.82	10.51	13.85	17.15	13.51	10.72	8.77	7.35	6.28	4.49	3.42	2.71	2.22	1.86	1.59
13	0.22	0.54	1.05	2.04	2.43	3.96	5.85	7.71	9.55	11.38	15.00	18.58	15.23	12.08	9.89	8.29	7.08	5.06	3.85	3.06	2.50	2.10	1.79
14	0.24	0.58	1.13	2.19	2.61	4.27	6.30	8.30	10.29	12.26	16.15	20.01	17.02	13.51	11.05	9.26	7.91	5.66	4.31	3.42	2.80	2.34	0.41
15	0.26	0.62	1.21	2.35	2.80	4.57	6.75	8.90	11.02	13.13	17.31	21.44	18.87	14.98	12.26	10.27	8.77	6.28	4.77	3.79	3.10	2.60	0.00
16	0.27	0.66	1.29	2.51	2.99	4.88	7.20	9.49	11.76	14.01	18.46	22.87	20.79	16.50	13.51	11.32	9.66	6.91	5.26	4.17	3.42	1.78	0.00
17	0.29	0.70	1.37	2.66	3.17	5.18	7.65	10.08	12.49	14.88	19.62	24.30	22.77	18.07	14.79	12.40	10.58	7.57	5.76	4.57	3.74	0.00	
18	0.31	0.75	1.45	2.82	3.36	5.49	8.10	10.68	13.23	15.76	20.77	25.73	24.81	19.69	16.11	13.51	11.53	8.25	6.28	4.98	4.08	0.00	
19	0.33	0.79	1.53	2.98	3.55	5.79	8.55	11.27	13.96	16.63	21.92	27.16	26.91	21.35	17.48	14.65	12.50	8.95	6.81	5.40	0.20	0.00	
20	0.34	0.83	1.61	3.13	3.73	6.10	9.00	11.86	14.70	17.51	23.08	28.59	29.06	23.06	18.87	15.82	13.51	9.66	7.35	5.83	0.00		
21	0.36	0.87	1.69	3.29	3.92	6.40	9.45	12.46	15.43	18.38	24.23	30.02	31.26	24.81	20.31	17.02	14.53	10.40	7.91	6.28	0.00		
22	0.38	0.91	1.77	3.45	4.11	6.71	9.90	13.05	16.17	19.26	25.39	31.45	33.52	26.60	21.77	18.25	15.58	11.15	8.48	0.00			
23	0.40	0.95	1.85	3.61	4.29	7.01	10.35	13.64	16.90	20.13	26.54	32.88	35.84	28.44	23.28	19.51	16.66	11.92	9.07	0.00			
24	0.41	0.99	1.93	3.76	4.48	7.32	10.80	14.24	17.64	21.01	27.69	34.31	38.20	30.31	24.81	20.79	17.75	12.70	9.66	0.00			
25	0.43	1.04	2.01	3.92	4.67	7.62	11.25	14.83	18.37	21.89	28.85	35.74	40.61	32.23	26.38	22.11	18.87	13.51	10.27	0.00			
26	0.45	1.08	2.09	4.08	4.85	7.93	11.70	15.42	19.11	22.76	30.00	37.17	43.07	34.18	27.98	23.44	20.02	14.32	10.90	0.00			
28	0.48	1.16	2.26	4.39	5.23	8.54	12.60	16.61	20.58	24.51	32.31	40.03	47.68	38.20	31.26	26.20	22.37	16.01	0.00				
30	0.52	1.24	2.42	4.70	5.60	9.15	13.50	17.79	22.05	26.26	34.62	42.89	51.09	42.36	34.67	29.06	24.81	17.75	0.00				
32	0.55	1.33	2.58	5.02	5.98	9.76	14.40	18.98	23.52	28.01	36.92	45.75	54.50	46.67	38.20	32.01	27.33	19.56	0.00				
35	0.60	1.45	2.82	5.49	6.54	10.67	15.75	20.76	25.72	30.64	40.39	50.03	59.60	53.38	43.69	36.62	31.26	1.55	0.00				
40	0.69	1.66	3.22	6.27	7.47	12.20	18.00	23.73	29.39	35.02	46.16	57.18	68.12	65.22	53.38	44.74	38.20	0.00					
45	0.77	1.86	3.63	7.05	8.40	13.72	20.25	26.69	33.07	38.39	51.92	64.33	76.63	77.83	63.70	53.38	12.45	0.00					

Type A Type B Type C

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Step 4. Compute the required number of teeth on the large sprocket:

$$N_2 = N_1 \times \text{ratio} = 17(3.83) = 65.11$$

Let's use the integer: 65 teeth.

Step 5. Compute the actual expected output speed:

$$n_2 = n_1(N_1/N_2) = 900\ \text{rpm}(17/65) = 235.3\ \text{rpm (Okay!)}$$

Step 6. Compute the pitch diameters of the sprockets using Equation (7–11):

$$D_1 = \frac{p}{\sin(180°/N_1)} = \frac{0.75\ \text{in}}{\sin(180°/17)} = 4.082\ \text{in}$$

$$D_2 = \frac{p}{\sin(180°/N_2)} = \frac{0.75\ \text{in}}{\sin(180°/65)} = 15.524\ \text{in}$$

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Step 7. Specify the nominal center distance. Let's use the middle of the recommended range, 40 pitches.

Step 8. Compute the required chain length in pitches from Equation (7–9):

$$L = 2C + \frac{N_2 + N_1}{2} + \frac{(N_2 - N_1)^2}{4\pi^2 C}$$

$$L = 2(40) + \frac{65 + 17}{2} + \frac{(65 - 17)^2}{4\pi^2 (40)} = 122.5 \text{ pitches} \qquad \textbf{(7–9)}$$

Step 9. Specify an integral number of pitches for the chain length, and compute the actual theoretical center distance. Let's use 122 pitches, an even number. Then, from Equation (7–10):

$$C = \frac{1}{4}\left[L - \frac{N_2 + N_1}{2} + \sqrt{\left[L - \frac{N_2 + N_1}{2}\right]^2 - \frac{8(N_2 - N_1)^2}{4\pi^2}}\right]$$

$$C = \frac{1}{4}\left[122 - \frac{65 + 17}{2} + \sqrt{\left[122 - \frac{65 + 17}{2}\right]^2 - \frac{8(65 - 17)^2}{4\pi^2}}\right] \qquad \text{(7–10)}$$

$C = 39.766$ pitches $= 39.766(0.75 \text{ in}) = 29.825$ in

Step 10. Compute the angle of wrap of the chain for each sprocket using Equations (7–12) and (7–13). Note that the minimum angle of wrap should be 120 degrees.

For the small sprocket,

$$\theta_1 = 180° - 2\sin^{-1}[(D_2 - D_1)/2C]$$
$$\theta_1 = 180° - 2\sin^{-1}[(15.524 - 4.082)/(2(29.825))] = 158°$$

Because this is greater than 120°, it is acceptable.

For the larger sprocket,

$$\theta_2 = 180° + 2\sin^{-1}(D_2 - D_1)/2C]$$
$$\theta_2 = 180° + 2\sin^{-1}[(15.524 - 4.082)/(2(29.825))] = 202°$$

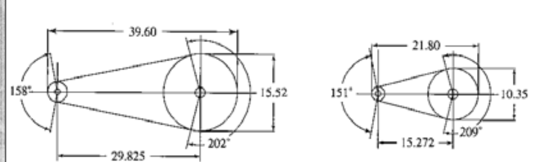
Comments

Summary of Design

Figure 7–25(a) shows a sketch of the design to scale.

Pitch: No. 60 chain, 3/4-in pitch
Length: 122 pitches = 122(0.75) = 91.50 in
Center distance: C = 29.825 in (maximum)
Sprockets: Single-strand, no. 60, 3/4-in pitch
Small: 17 teeth, D = 4.082 in
Large: 65 teeth, D = 15.524 in

Type B lubrication is required. The large sprocket can dip into an oil bath.



(*a*) Chain drive system for Example Problem 7–2 (*b*) Chain drive system for Example Problem 7–3