ENGINEERING DATA

NOTE: This engineering section provides general engineering information for synchronous belts and sprockets (or pulleys) which are useful in general drive design work. Where we refer to sprockets (for PowerGrip® GT®2 belts), you can substitute pulleys for PowerGrip Timing Belts. If you need additional information, contact Gates Power Transmission Product Application.

Section I

Application Design Considerations

When designing synchronous drives, there are several special circumstances that may require additional consideration:

1. **Gear Motors/ Speed Reducer Drives**
2. **Electric Motor Frame Dimensions**
3. **Minimum Sprocket Diameter Recommendations for Electric Motors**
4. **High-Driven Inertia**
5. **Air Moving Drives**
6. **Linear Motion Drives**
7. **High Performance Applications**
8. **Belt Drive Registration**
9. **Belt Drive Noise**
10. **Use of Flanged Sprockets**
11. **Fixed (Nonadjustable) Center Distance**
12. **Use of Idlers**
13. **Specifying Shaft Locations in Multipoint Drive Layouts**
14. **Minimum Belt Wrap and Tooth Engagement**
15. **Adverse Operating Environments**

Each of these circumstances and special considerations are reviewed below.

1. Gear Motors/ Speed Reducer Drives

When designing a belt drive system to transfer power from the output shaft of a speed reducer to the final driven shaft, the designer must make certain that the belt drive does not exert shaft loads greater than the speed reducing device is rated to carry. Failure to do so can result in premature shaft/ bearing failures whether the belt drive has been designed with the appropriate power capacity or not.

This concept is similar to the National Electric Motor Association (NEMA) establishing minimum acceptable sprocket diameters for each of their standardized motor frames. Abiding by these minimum recommended diameters, when designing a belt drive system, prevents the motor bearings from failing prematurely due to excessive shaft loads exerted by the belt drive.

Overhung load is generally defined as a force exerted by a belt or chain drive, that is perpendicular to a speed reducer shaft, and applied beyond its outermost bearing. Calculated overhung load values are intended to serve as an indication of how heavily loaded the shaft and outermost bearing of a speed reducer actually is.
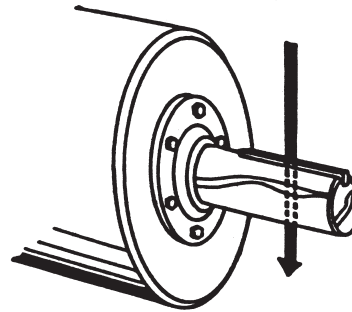

Figure 3 - Overhung Load

Overhung load calculations are generally assumed to apply to the slower output shaft of a speed reducer. It is important to note that these calculations apply to higher speed input shafts as well.

Most speed reducer manufacturers publish allowable overhung load values for every model in their product line. This value represents the maximum load that the shaft and bearings can support without negatively impacting the durability of the speed reducer. When the actual overhung load exceeds the published allowable value, premature shaft or bearing failure may occur. In extreme cases, catastrophic failures can occur.

A general formula used to calculate overhung load (OHL) is as follows:

$$\text{OHL} = \frac{126{,}000 \times \text{HP} \times K_{LCF} \times K_{SF} \times K_{LLF}}{\text{PD} \times \text{RPM}}$$

Where: HP = Actual horsepower being transmitted at the gear motor/reducer output shaft with no service factor applied

K_{LCF} = Overhung load connection factor (1.3 for all synchronous belt drives)

K_{SF} = Service factor for the speed reducer (available from the manufacturer)

K_{LLF} = Load location factor for the speed reducer (available from the manufacturer)

PD = Pitch diameter of the speed reducer output shaft sprocket

RPM = RPM of the speed reducer output shaft

Speed reducer manufacturers each publish their own specific formula and constants to calculate overhung load. They also publish specific overhung load ratings for each speed reducer product that they produce. It is very important to use the correct overhung load calculation procedure in conjunction with the manufacturer's accompanying overhung load rating.

Gates.com/pt

If the calculated overhung load for a particular belt drive system does exceed the speed reducer manufacturer's maximum recommended value, consider altering the belt drive design. In order to reduce the calculated overhung load, consider:

- Increasing sprocket diameters
- Reducing belt width
- Mounting the sprocket closer to the speed reducer outboard bearing
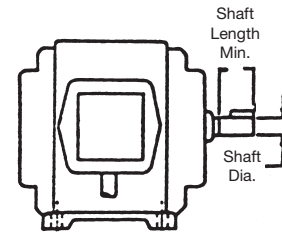
Increasing the sprocket diameter not only reduces calculated overhung load, it also potentially reduces the required belt width. Reducing the belt width and mounting the sprocket as close as possible to the outermost bearing of the speed reducer both move the center of the belt load closer to the speed reducer. This also reduces the calculated overhung load. Alterations to the belt drive design should be made until the calculated overhung load is within the speed reducer manufacturer's recommendations.

2. Electric Motor Frame Dimensions

Motor dimensions can be important considerations depending on the application and its requirements. If motor shaft length, motor shaft diameter, or clearance issues are a concern, refer to the motor dimension table on this page. The table lists common general purpose electric motors by frame size.

Shaft Length Min.
Shaft Dia.

Motor Frame Dimensions

Frame Size	Shaft Dia. (in)	Shaft Length Min. (in)	Key (in)
48	1/2	—	3/64 Flat
56	5/8	—	3/16 x 3/16 x 1-3/8
143T	7/8	2	3/16 x 3/16 x 1-3/8
145T	7/8	2	3/16 x 3/16 x 1-3/8
182	7/8	2	3/16 x 3/16 x 1-3/8
182T	1-1/8	2-1/2	1/4 x 1/4 x 1-3/4
184	7/8	2	3/16 x 3/16 x 1-3/8
184T	1-1/8	2-1/2	1/4 x 1/4 x 1-3/4
213	1-1/8	2-3/4	1/4 x 1/4 x 2
213T	1-3/8	3-1/8	5/16 x 5/16 x 2-3/8
215	1-1/8	2-3/4	1/4 x 1/4 x 2
215T	1-3/8	3-1/8	5/16 x 5/16 x 2-3/8
254U	1-3/8	3-1/2	5/16 x 5/16 x 2-3/4
254T	1-5/8	3-3/4	3/8 x 3/8 x 2-7/8
256U	1-3/8	3-1/2	5/16 x 5/16 x 3-3/4
256T	1-5/8	3-3/4	3/8 x 3/8 x 2-7/8
284U	1-5/8	4-5/8	3/8 x 3/8 x 3-3/4
284T	1-7/8	4-3/8	1/2 x 1/2 x 3-1/4
284TS	1-5/8	3	3/8 x 3/8 x 1-7/8
286U	1-5/8	4-5/8	3/8 x 3/8 x 3-3/4
286T	1-7/8	4-3/8	1/2 x 1/2 x 3-1/4
286TS	1-5/8	3	3/8 x 3/8 x 1-7/8
324U	1-7/8	5-3/8	1/2 x 1/2 x 4-1/4
324T	2-1/8	5	1/2 x 1/2 x 3-7/8
324TS	1-7/8	3-1/2	1/2 x 1/2 x 2
326U	1-7/8	5-3/8	1/2 x 1/2 x 4-1/4
326T	2-1/8	5	1/2 x 1/2 x 3-7/8
326TS	1-7/8	3-1/2	1/2 x 1/2 x 2
364U	2-1/8	6-1/8	1/2 x 1/2 x 5
364US	1-7/8	3-1/2	1/2 x1/2 x 2
364T	2-3/8	5-5/8	5/8 x 5/8 x 4-1/4
364TS	1-7/8	3-1/2	1/2 x 1/2 x 2
365U	2-1/8	6-1/8	1/2 x 1/2 x 5
365US	1-7/8	3-1/2	1/2 x 1/2 x 2
365T	2-3/8	5-5/8	5/8 x 5/8 x 4-1/4
365TS	1-7/8	3-1/2	1/2 x 1/2 x 2
404U	2-3/8	6-7/8	5/8 x 5/8 x 5-1/2
404US	2-1/8	4	1/2 x 4 x 2-3/4
404T	2-7/8	7	3/4 x 3/4 x 5-5/8
404TS	2-1/8	4	1/2 x 1/2 x 2-3/4
405U	2-3/8	6-7/8	5/8 x 5/8 x 5-1/2
405US	2-1/8	4	1/2 x 1/2 x 2-3/4
405T	2-7/8	7	3/4 x 3/4 x 5-5/8
405TS	2-1/8	4	1/2 x 1/2 x 2-3/4
444U	2-7/8	8-3/8	3/4 x 3/4 x 7
444US	2-1/8	4	1/2 x 1/2 x 2-3/4
444T	3-3/8	8-1/4	7/8 x 7/8 x 6-7/8
444TS	2-3/8	4-1/2	5/8 x 5/8 x 3
445U	2-7/8	8-3/8	3/4 x 3/4 x 7
445US	2-1/8	4	1/2 x 1/2 x 2-3/4
445T	3-3/8	8-1/4	7/8 x 7/8 x 6-7/8
445TS	2-3/8	4-1/2	5/8 x 5/8 x 3
447T	3-3/8	8-1/4	7/8 x 7/8 x 6-7/8
447TS	2-3/8	4-1/2	5/8 x 5/8 x 3
449T	3-3/8	8-1/4	7/8 x 7/8 x 6-7/8
449TS	2-3/8	4-1/2	5/8 x 5/8 x 3

3. Minimum Sprocket Diameter Recommendations for Electric Motors

Minimum Recommended Sprocket / Sheave Diameters

NEMA (The National Electric Manufacturers Association) publishes recommendations for the minimum diameter of sprockets and sheaves to be used on General Purpose Electric Motors. The purpose of these recommendations is to prevent the use of excessively small sprockets or sheaves. This can result in motor shaft or bearing damage since belt pull increases as the diameter is reduced.

Table data has been compiled from NEMA Standard MG-1-14-42; 11/78, MG-1-14-43; 1/68, and a composite of electric motor manufacturers data. Values are generally conservative, and specific motors may permit the use of a smaller sprocket or sheave. Consult the motor manufacturer.

Motor Frames and Minimum Diameters for 60 Cycle Electric Motors

Motor Frame Code	Shaft Dia. (in)	Horsepower at Synchronous Speed (rpm)				Synchronous Belts
		3600 (3450)	1800 (1750)	1200 (1160)	900 (870)	Min. Pitch Dia. (in)
143T	0.875	1-1/2	1	3/4	1/2	2.0
145T	0.875	2—3	1-1/2— 2	1	3/4	2.2
182T	1.125	3	3	1-1/2	1	2.2
182T	1.125	5	—	—	—	2.4
184T	1.125	—	—	2	1-1/2	2.2
184T	1.125	5	—	—	—	2.2
184T	1.125	7-1/2	5	—	—	2.7
213T	1.375	7-1/2—10	7-1/2	3	2	2.7
215T	1.375	10	—	5	3	2.7
215T	1.375	15	10	—	—	3.4
254T	1.625	15	—	7-1/2	5	3.4
254T	1.625	20	15	—	—	4.0
256T	1.625	20—25	—	10	7-1/2	4.0
256T	1.625	—	20	—	—	4.0
284T	1.875	—	—	15	10	4.0
284T	1.875	—	25	—	—	4.0
286T	1.875	—	30	20	15	4.7
324T	2.125	—	40	25	20	5.4
326T	2.125	—	50	30	25	6.1
364T	2.375	—	—	40	30	6.1
364T	2.375	—	60	—	—	6.7
365T	2.375	—	—	50	40	7.4
365T	2.375	—	75	—	—	7.7
404T	2.875	—	—	60	—	7.2
404T	2.875	—	—	—	50	7.6
404T	2.875	—	100	—	—	7.7
405T	2.875	—	—	75	60	9.0
405T	2.875	—	100	—	—	7.7
405T	2.875	—	125	—	—	9.5
444T	3.375	—	—	100	—	9.0
444T	3.375	—	—	—	75	8.6
444T	3.375	—	125	—	—	8.6
444T	3.375	—	150	—	—	9.5
445T	3.375	—	—	125	—	10.8
445T	3.375	—	—	—	100	10.8
445T	3.375	—	150	—	—	9.5
445T	3.375	—	200	—	—	11.9

4. High-Driven Inertia

Many drives, such as piston compressors, punch presses and crushers, depend on the driveN pulley acting as a flywheel. This flywheel effect, or WR^2 is used to help moderate or smooth out fluctuations in driven load and speed. Failure to compensate for this during a redesign can result in premature damage to the prime mover or early belt failures. This can be a consideration when replacing older belt drives with new, higher capacity belts.

When replacing large pulleys or sheaves with sprockets, be careful not to remove a designed-in flywheel effect. Ask questions of the user to make sure there is not a concern for a high WR^2. If there is a concern, you may have to use a wider sprocket, a larger diameter, or a special made-to-order sprocket designed with added weight and WR^2.

Drives which have a high driveN inertia and are subjected to high acceleration or emergency stop conditions require additional design expertise. Contact Gates Power Transmission Product Application for further engineering assistance.

5. Air Moving Drives

HVAC Equipment Inspection

Many air handling drives have structures that are not particularly rigid, which can create belt tension and drive alignment problems resulting in unusual and premature belt wear. Synchronous belts are sensitive to fluctuations in center distance that can be caused by inadequate bracketry. Under start up conditions, an AC motor can be required to provide 150% to 200% of its rated capacity. Synchronous belts cannot slip, and must transmit the higher start up torque. Under these conditions, the drive center distance may collapse if the structure is not sufficiently rigid.

With the drive shut off and safely locked out, a simple method to use when inspecting potential drive conversions is to grab the two belt spans and push them together while observing the motor. If any significant relative change in center distance or motor position is noticed, the drive's structural strength is most likely insufficient for a simple conversion. The structure would need to be reinforced to obtain optimum performance from a synchronous belt drive. The best conversion candidates have motors that are mounted solidly on support bracketry that is part of the fan's structural system. When possible, select synchronous drives with diameters similar to existing V-belt sheave diameters. This will maintain similar belt pulls and loads on the shafts and structure.

Air Handling Unit Start-Up Characteristics

Full Load Start Up
Start up loads can be a concern when evaluating potential drives for conversion to synchronous belts. Synchronous belts will transmit all of the start up torque, where V-belts may slip if the load is excessive. Due to the inertia of the fan, start up loads can potentially be 150% to 200% of the normal operating load. It is important that the start up load be considered by selecting appropriate service factors when designing a belt drive system.

Controlled Start Up

An air handling drive with soft start or variable frequency controller (AC Inverter) is ideal for conversion to synchronous belts. The fan will be ramped up to speed slowly, with a corresponding increase in load as the speed increases. Structural flexing is typically not a concern when designing synchronous belt drives on systems using soft starts or variable frequency controllers.

Fan Speed

The volume of air being transmitted and the required horsepower are both sensitive to changes in the driveN fan speed. If designing a synchronous belt drive for energy savings, it is important that the synchronous belt drive be designed to operate at the proper driveN fan speed. All conversions from existing V-belt drives should have the synchronous belt drive speed ratio based on a measured driveN shaft RPM, and not calculated from the theoretical V-belt speed ratio. This measurement can be made by either using a mechanical contact tachometer or a strobe tachometer.

The horsepower requirement for fans varies with the cube of the fan speed. A small change in the fan speed makes a much larger difference in the actual horsepower and energy required.

$$HP_1/HP_2 = (RPM_1/RPM_2)^3$$

Where: HP_1 = Initial Horsepower
HP_2 = New Horsepower @ New Fan RPM
RPM_1 = Initial Fan RPM
RPM_2 = New Fan RPM

Air-Cooled Heat Exchanger (ACHE) Applications

Air-cooled heat exchangers are used in Petrochemical, Oil and Gas Production, Power Generation, and Petroleum Refining Industries where process heat must be removed. Electric motors as large as 60 hp commonly drive the cooling fans with either large ratio V-belt or Synchronous belt drives.

According to the American Petroleum Institute (**API 661** - Air-Cooled Heat Exchangers for General Refinery Service), a safety factor of 1.8 must be used in the belt drive design process. Synchronous belt drives typically have higher horsepower capacities than V-belt drives with an equivalent width. This increased capacity results in narrower belt drives and lighter drive hardware. Synchronous belt drive systems are especially beneficial on higher horsepower heat exchanger units, and they are commonly used on new or redesigned units. V-belt drive systems are commonly used on low to medium HP fans because of their relatively low cost and good availability.

Surface rust on sheaves and sprockets is very abrasive, and rapidly wears belts. Sprockets on wet heat exchanger applications (water drawn through heat exchanger coils by fan) such as Cooling Towers, often rust and require the use of electroless nickel plating to prevent excessive corrosion. Cooling Towers are commonly used to cool large buildings (HVAC; Heating-Ventilating-Air Conditioning Systems). Misalignment is a common cause of premature belt failures on ACHE drive systems. Care should be taken to ensure proper sheave / sprocket alignment when installing the belt drive system. See **Gates Belt Drive Preventative**

Maintenance and Safety Manual (Form 14995) for detailed information about proper belt drive alignment.

Proper belt pre-tension is necessary to obtain optimum belt performance. This is particularly true for the high inertia start up loads seen in ACHE applications. If belt installation tension is too low, V-belts will be prone to slippage and synchronous belts will be prone to tooth jump or ratcheting. Motor controllers are sometimes used to bring the fan up to speed slowly (soft start), decreasing the chance of synchronous belt ratcheting.

6. Linear Motion Drives

In linear motion drives, such as a rack and pinion application, the belt is not transmitting a load in the conventional rotational manner. The two cut ends of the belt are connected to clamping fixtures and the belt travels back and forth a specified distance while rotating over a sprocket. Because of these characteristics, the drive design process will typically not follow standard catalog design procedures.

The designer will most likely have available a maximum belt load or pull which will need to be related to the belt's allowable working tension. Reasonably sized sprocket diameters are still required to prevent excessive stress fatigue in the belt. In these applications, the designer may either use endless belts and cut them, or use standard long length belting when available. Product listings are on pages 115-117. Gates Power Transmission Product Application may be consulted for design assistance.

7. High Performance Vehicle Applications

For special high performance applications, such as motorcycles or race car and boat supercharger drives, the design loads will typically exceed published data. Because of the extremely high loads and speeds (as much as 500 HP and belt speeds exceeding 10,000 fpm), it is necessary for the designer to contact Gates Power Transmission Product Application for additional assistance.

Although special considerations may be involved, it is important to remember that reasonable drive recommendations can be provided to the designer in most cases.

8. Belt Drive Registration

The three primary factors contributing to belt drive registration (or positioning) errors are belt elongation, backlash, and tooth deflection. When evaluating the potential registration capabilities of a synchronous belt drive, the system must first be determined to be either static or dynamic in terms of its registration function and requirements.

Static Registration: A static registration system moves from its initial static position to a secondary static position. During the process the designer is concerned only with how accurately and consistently the drive arrives at its secondary position. Potential registration errors that occur during transport are not considered. Therefore, the primary factor contributing to registration error in a static registration system is backlash. The effects of belt elongation and tooth deflection do not have any influence on the registration accuracy of this type of system.

Dynamic Registration: A dynamic registration system is required to perform a registering function while in motion with torque loads varying as the system operates. In this case, the designer is concerned with the rotational position of the drive sprockets with respect to each other at every point in time. Therefore, belt elongation, backlash, and tooth deflection will all contribute to registrational inaccuracies.

Further discussion about each of the factors contributing to registration error is as follows:

Belt Elongation: Belt elongation, or stretch, occurs naturally when a belt is placed under tension. The total tension exerted within a belt results from installation as well as working loads. The amount of belt elongation is a function of the belt tensile modulus, which is influenced by the type of tensile cord and the belt construction. The standard tensile cord used in rubber synchronous belts is fiberglass. Fiberglass has a high tensile modulus, is dimensionally stable, and has excellent flex-fatigue characteristics. If a higher tensile modulus is needed in a rubber synchronous belt, aramid tensile cords can be considered, although they are generally used to provide resistance to harsh shock and impulse loads. Aramid tensile cords used in rubber synchronous belts generally have only a marginally higher tensile modulus in comparison to fiberglass. When needed, belt tensile modulus data is available from Gates Power Transmission Product Application.

Backlash: Backlash in a synchronous belt drive results from clearance between the belt teeth and the sprocket grooves. This clearance is needed to allow the belt teeth to enter and exit the grooves smoothly with a minimum of interference. The amount of clearance necessary depends upon the belt tooth profile. PowerGrip® Timing Belt Drives are known for having relatively little backlash. PowerGrip® HTD® Drives have improved torque carrying capability and resist ratcheting, but have a significant amount of backlash. PowerGrip® GT®3 Drives have considerably improved torque carrying capability, and backlash characteristics in between that of PowerGrip HTD and PowerGrip Timing Drives. In special cases, alterations can be made to drive systems to further decrease backlash. These alterations often result in increased belt wear, increased drive noise and shorter drive life. Contact Gates Power Transmission Product Application for additional information.

Tooth Deflection: Tooth deformation in a synchronous belt drive occurs as a torque load is applied to the system, and individual belt teeth are loaded. The amount of belt tooth deformation depends upon the amount of torque loading, sprocket size, installation tension and belt type. Of the three primary contributors to registration error, tooth deflection is the most difficult to quantify. Experimentation with a prototype drive system is the best means of obtaining realistic estimations of belt tooth deflection.

Additional guidelines that may be useful in designing registration critical drive systems are as follows:

- Design with large sprockets with more teeth in mesh.
- Keep belts tight, and control tension closely.
- Design frame/shafting to be rigid under load.
- Use high quality machined sprockets to minimize radial run out and lateral wobble.

9. Belt Drive Noise

V-belt, synchronous belt, roller chain, and gear drives will all generate noise while transmitting power. Each type of system has its own characteristic sound. V-belt drives tend to be the quietest and synchronous belt drives are much quieter than roller chain drives. When noise is an issue, there are several design and maintenance tips that should be followed to minimize belt drive noise.

Noise: Decibel and Frequency
Noise is an unwanted or unpleasant sound that can be described with two criteria – frequency and decibel (dB) levels. Frequency is measured in Hertz. A perfect human ear is capable of distinguishing frequencies typically from 20 to 20,000 Hertz. The human ear does generally not perceive frequencies higher than 20,000 Hertz.

The sound pressure level or intensity of noise is measured in terms of decibels (dB). The decibel has become the basic unit of measure since it is an objective measurement that approximately corresponds to the subjective measurement made by the human ear. Since sound is composed of several distinct and measurable parts and the human ear doesn't differentiate between these parts, measuring scales that approximate the human ear's reaction have been adopted. Three scales – A, B, and C are used to duplicate the ear's response over the scale's ranges. The A scale is most commonly used in industry because of its adoption as the standard in OSHA regulations. Noise described in decibels (dBA - "A" weighting for the human ear) is generally perceived as the loudness or intensity of the noise.

While the human ear can distinguish frequencies over a broad range, the ear is most sensitive in the range of normal speech – 500 to 2000 Hertz. As a consequence, this is the range most commonly of concern for noise control ("A" weighting gives more weight or emphasis to sounds in the 500 to 2000 hz range). Frequency is most closely related to what the ear hears as pitch. High frequency sounds are perceived as whining or piercing, while low frequency sounds are perceived as rumbling.

The combination of sound pressure level (dB) and frequency describes the overall level of loudness perceived by the human ear. One without the other does not adequately describe the loudness potential of the noise. For example, an 85 dBA noise at 3000 Hertz is going to be perceived as being much louder than an 85 dBA noise at 500 Hertz.

Reducing Noise
Following proper installation and maintenance procedures, as well as some simple design alternatives can reduce belt drive noise.

Belt Drive Tension and Alignment
Properly tensioning and aligning a belt drive will allow the belt drive to perform at its quietest level. Improper tension in synchronous belt drives can affect how the belt fits in the sprocket grooves. Proper tension minimizes tooth to groove interference, and thereby reduces belt noise.

Misaligned synchronous belt drives tend to be much noisier than properly aligned drives due to the amount of interference that is created between the belt teeth and the sprocket grooves. Misaligned synchronous belt drives also may

cause belt tracking that forces the edge of the belt to ride hard against a sprocket flange. Misalignment causing belt contact with a flange will generate noise that is easily detected.

Noise Barriers and Absorbers
Sometimes, even properly aligned and tensioned belt drives may be too noisy for a work environment. When this occurs, steps can be taken to modify the drive guard to reduce the noise level.

Noise barriers are used to block and reflect noise. Noise barriers do not absorb or deaden the noise; they block the noise and generally reflect most of the noise back towards its point of origin. Good noise barriers are dense, and should not vibrate. A sheet metal belt guard is a noise barrier. The more complete the enclosure is, the more effective it is as a noise barrier. Noise barrier belt guards can be as sophisticated as a completely enclosed case, or as simple as sheet metal covering the front of the guard to prevent direct sound transmission.

Noise absorbers are used to reduce noise reflections and to dissipate noise energy. Noise absorbers should be used in combination with a noise barrier. Noise absorbers are commonly referred to as acoustic insulation. Acoustic insulation (the noise absorber) is used inside of belt guards (the noise barrier) where necessary. A large variety of acoustic insulation manufacturers are available to provide different products for the appropriate situation.

A combination of noise barrier (solid belt guard) and noise absorber (acoustic insulation) will provide the largest reduction in belt drive noise. While the noise reduction cannot be predicted, field experience has shown that noise levels have been reduced by 10 to 20 dBA when using complete belt guards with acoustic insulation.

10. Use of Flanged Sprockets

Guide flanges are needed in order to keep the belt on the sprocket. Due to tracking characteristics, even on the best aligned drives, belts will ride off the edge of the sprockets. Flanges will prevent this belt ride-off.

On all drives using stock or made-to-order sprockets, the following conditions should be considered when selecting flanged sprockets:

1. On all two-sprocket drives, the minimum flanging requirements are two flanges on one sprocket or one flange on each sprocket on opposite sides.

2. On drives where the center distance is more than eight times the diameter of the small sprocket, both sprockets should be flanged on both sides. (See Engineering Section II, Belt Installation and Drive Alignment on Pages 182 and 183.)

3. On vertical shaft drives, one sprocket should be flanged on both sides, and all the other sprockets in the system should be flanged on the bottom side only.

4. On drives with more than two sprockets, the minimum flanging requirements are two flanges on every other sprocket or one flange on every sprocket—on alternating sides around the system.

On made-to-order sprockets, flanges must be securely fastened, such as using mechanical fasteners, welding, shrink-fit or other equivalent methods.

11. Fixed (Nonadjustable) Center Distance

Designers sometimes attempt to design synchronous belt drive systems without any means of belt adjustment or take up. This type of system is called a Fixed Center Drive. While this approach is often viewed as being economical, and is simple for assemblers, it often results in troublesome reliability and performance problems in the long run.

The primary pitfall in a fixed center design approach is failure to consider the effects of system tolerance accumulation. Belts and sprockets are manufactured with industry accepted production tolerances. There are limits to the accuracy that the center distance can be maintained on a production basis as well. The potential effects of this tolerance accumulation is as follows:

Low Tension:
Long Belt with Small Sprockets on a Short Center Distance

High Tension:
Short Belt with Large Sprockets on a Long Center Distance

Belt tension in these two cases can vary by a factor of 3 or more with a standard fiberglass tensile cord, and even more with an aramid tensile cord. This potential variation is great enough to overload bearings and shafting, as well as the belts themselves. The probability of these extremes occurring is a matter of statistics, but however remote the chances seem, they will occur in a production setting. In power transmission drives, the appearance of either extreme is very likely to impact drive system performance in a negative manner.

The most detrimental aspect of fixed center drives is generally the potentially high tension condition. This condition can be avoided by adjusting the design center distance. A common approach in these designs is to reduce the center distance from the exact calculated value by some small fraction. This results in a drive system that is inherently loose, but one that has much less probability of yielding excessively high shaft loads. **NOTE:** This approach should not be used for power transmission drives since the potentially loose operating conditions could result in accelerated wear and belt ratcheting, even under nominal loading.

There are times when fixed center drive designs can't be avoided. In these cases, the following recommendations will maximize the probability of success.

1. Do not use a fixed center design for power transmission drives. Consider using a fixed center design only for lightly loaded or motion transfer applications.

2. Do not use a fixed center design for drives requiring high motion quality or registration precision.

3. When considering a fixed center design, the center distance must be held as accurately as possible, typically within 0.002"—0.003" (0.05mm—0.08mm). This accuracy often requires the use of stamped steel framework.

4. Sprockets for fixed center systems should be produced with a machining process for accuracy.

Molding and sintering processes are generally not capable of holding the finished O.D. sufficiently accurate for these systems.

5. The performance capabilities of the drive system should be verified by testing belts produced over their full length tolerance range on drive systems representing the full potential center-distance variation. Contact Gates Power Transmission Product Application for further details.

6. Contact Gates Power Transmission Product Application for design center distance recommendations, and to review the application.

12. Use of Idlers

Use of idlers should be restricted to those cases in which they are functionally necessary. Idlers are often used as a means of applying tension when the center distance is not adjustable.

Idlers should be located on the slack side span of the belt drive. General size recommendations are listed for inside grooved, inside flat, and backside idlers. In some cases, such as high capacity drives utilizing large sprockets, idlers as large as the smallest loaded sprocket in the system may be more appropriate.

Idler Size Recommendations

Belt	Minimum Inside Idler	Minimum Inside Flat Idler	Minimum Backside Idler
XL PowerGrip® Timing	12 grooves	2.50" O.D.	1.00" O.D.
L PowerGrip Timing	10 grooves	4.75" O.D.	1.60" O.D.
H PowerGrip Timing	14 grooves	6.38" O.D.	2.88" O.D.
5M PowerGrip GT®3	14 grooves	2.50" O.D.	1.25" O.D.
8M PowerGrip GT3	22 grooves	4.00" O.D.	2.80" O.D.
14M PowerGrip GT3	28 grooves	7.00" O.D.	6.50" O.D.
20M PowerGrip HTD®	34 grooves	10.00" O.D.	11.00" O.D.

Outside or backside idlers should be flat and uncrowned; flanges may or may not be necessary. Drives with flat inside idlers should be tested, as noise and belt wear may occur.

Idler arc of contact should be held to a minimum. All idlers should be rigidly mounted in place to minimize movement or deflection during drive startup and operation.

13. Specifying Shaft Locations in Multipoint Drive Layouts

When collecting geometrical layout data for multiple sprocket drive layouts, it is important to use a standard approach that is readily understood and usable for drive design calculations. This is of particular importance when the data will be provided to Gates Application Engineering for analysis.

Multipoint Drive
When working with a drive system having more than three shafts, the geometrical layout data must be collected in terms of X-Y coordinates for analysis.
For those unfamiliar with X-Y coordinates, the X-Y cartesian coordinate system is commonly used in mathematical and engineering calculations and utilizes a horizontal and vertical axis as illustrated in Fig. 4.
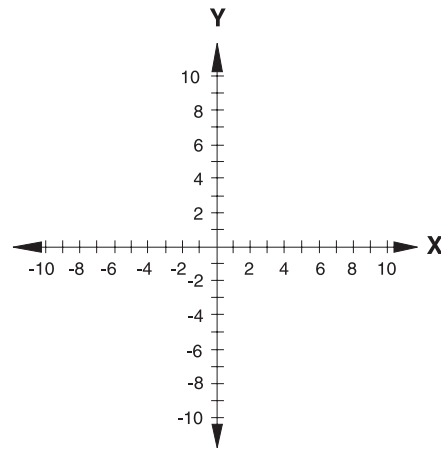

Figure 4

The axes cross at the zero point, or origin. Along the horizontal, or "X" axis, all values to the right of the zero point are positive, and all values to the left of the zero point are negative. Along the vertical, or "Y" axis, all values above the zero point are positive, and all values below the zero point are negative. This is also illustrated in Figure 4. When identifying a shaft center location, each X-Y coordinate is specified with a measurement in the "X" as well as the "Y" direction. This requires a horizontal and vertical measurement for each shaft center in order to establish a complete coordinate. Either English or Metric units of measurement may be used.

A complete coordinate is specified as follows:

(X,Y) where X = measurement along X-axis (horizontal)
Y = measurement along Y-axis (vertical)

In specifying X-Y coordinates for each shaft center, the origin (zero point) must first be chosen as a reference. The driveR shaft most often serves this purpose, but any shaft center can be used. Measurements for all remaining shaft centers must be taken from this origin or reference point. The origin is specified as (0,0).

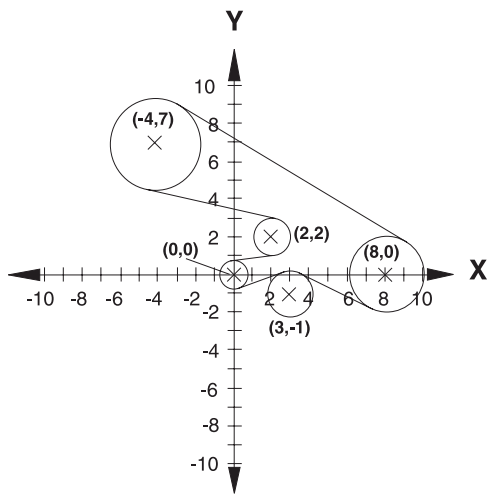An example layout of a 5-point drive system is illustrated

Figure 5

in Figure 5. Here each of the five shaft centers are located and identified on the X-Y coordinate grid.

When specifying parameters for the moveable or adjustable shaft (for belt installation and tensioning), the following approaches are generally used:

Fixed Location: Specify the nominal shaft location coordinate with a movement direction.

Slotted Location: Specify a location coordinate for the beginning of the slot, and a location coordinate for the end of the slot along its path of linear movement.

Pivoted Location: Specify the initial shaft location coordinate along with a pivot point location coordinate and the pivot radius.

Performing belt length and idler movement/positioning calculations by hand can be quite difficult and time consuming. With a complete geometrical drive description, we can make the drive design and layout process quite simple for you. Contact Gates Power Transmission Product Application for computer-aided assistance.

14. Minimum Belt Wrap and Tooth Engagement

Horsepower ratings listed in this catalog are based on a minimum of six teeth in mesh between the belt and the sprocket. The ratings must be corrected for excessive tooth loading if there are less than six teeth in mesh. For nonstock drives not listed in the Drive Selection Tables, the teeth in mesh may be calculated by using this formula:
In cases where fewer than six teeth are in full contact, 20%

Formula 1

$$\text{Teeth in Mesh} = \left[0.5 - \left(\frac{D - d}{6C} \right) \right] N_g$$

Where: D = pitch diameter, large sprocket, inches
d = pitch diameter, small sprocket, inches
C = center distance between shafts, inches
N_g = number of grooves in small sprocket

of the horsepower rating must be subtracted for each tooth less than six not in full contact. After computing the teeth in mesh, the belt rating should be multiplied by the appropriate K_{TM} factor shown in the following table.
In addition to the number of teeth in mesh, some drives

Teeth In Mesh Correction Factor

Teeth in Mesh	Factor K_{TM}
6 or more	1.00
5	0.80
4	0.60
3	0.40
2	0.20

with more than two shafts may have a greater potential for the belts to ratchet where loaded sprockets have six teeth in mesh, but a small arc of contact. In order to minimize this condition, each loaded sprocket in the drive system should have an arc of contact or belt wrap angle of at least 60 degrees. Non-loaded idler sprockets do not have tooth meshing or wrap angle requirements.

15. Adverse Operating Environments

Debris
Be very careful when using synchronous drives in high debris environments. Debris can be more damaging to a positive belt drive than a V-belt drive, which has a tendency to remove debris from the sheave grooves through drive operation. Entrapment of debris in synchronous drives is a major concern. Debris can be packed into sprocket grooves causing improper belt tooth engagement, reducing belt life and accelerating belt and sprocket wear. Care must be taken to provide adequate shielding to drives in environments where debris is likely. Completely enclosing a synchronous belt drive may be acceptable. Since synchronous belts generate minimal heat during drive operation, air circulation is not critical except where extremely high temperatures already are present. Depending on the type and abrasive characteristics of the debris, excessive wear can be generated on both belt and sprockets.

Temperature
Belt performance is generally unaffected in ambient temperature environments between -30° and 185°F (-34° and 85°C). Temperature extremes beyond these limits should be reviewed by Gates Power Transmission Product Application.

Chemical Resistance
Based on lab and field testing, PowerGrip® belts provide excellent resistance to most chemicals. Actual performance characteristics will be determined by the degree of concentration of the chemical, the time of exposure and the type of exposure (drip, splash, immersion, etc.). In addition to possible belt degradation, these chemicals can act as a lubricant in the drive system. As with any positive belt drive, PowerGrip drives which run where excessive lubrication is present have an increased tendency to ratchet (See Engineering Section II-14, Self Generated Tension on Page 184). Special attention should be given to assure that recommended tension is maintained (See Engineering Section II-8, Belt Installation Tension on Page 180).

High Humidity/Corrosive Environments

Many industrial applications face problems associated with rusting parts. Numerous applications in the food and beverage industry are located in areas that require periodic wash down. Unless a drive is completely shielded and protected from wash down, rust and corrosion will be rapidly apparent in these types of environments. This is equally true of sprockets when used in very wet or humid environments, such as seen with air moving drives on cooling towers or wood kilns. The constant effects of the wet air surrounding the belt drive can cause excessive rust.

Corrosion attacks sprocket grooves, building up rust deposits. The corrosion will increase over time, building up in the sprocket grooves and non-driving surfaces (flanges, sprocket faces, bushing face). Sprockets with corrosion in the grooves will rapidly wear the belt's teeth and wear through the abrasion resistant tooth fabric, resulting in tooth shear and premature belt failure.

When an application is in a corrosive environment, the designer may elect to use special sprockets and bushings to prevent premature failures. Using special stainless steel sprockets and bushings or electroless nickel-plated sprockets can help eliminate corrosion as a cause of failure on belt drives located in these damaging environments.

Section II

Engineering Design Considerations

All synchronous belt drives require proper installation procedures for optimum performance. In addition, topics such as tooth profile advantages, sprocket rim speed limitations, efficiency, and tolerances are common to all Gates synchronous belt drives.

1. **Belt Storage and Handling**
2. **Center Distance and Belt Length**
3. **Tooth Profiles**
4. **Static Conductivity**
5. **Sprocket Diameter - Speed**
6. **Efficiency**
7. **Belt Tolerances**
8. **Belt Installation Tension**
9. **Center Distance Allowances for Installation and Tensioning**
10. **Drive Alignment**
11. **Belt Installation**
12. **Belt Pull Calculations**
13. **Bearing/Shaft Load Calculations**
14. **Self-Generated Tension**

Each of these circumstances and special considerations are reviewed below.

1. Belt Storage and Handling

Storage Recommendations
In order to retain their serviceability and dimensions, proper storage procedures must be followed for synchronous belts. Quite often premature belt failures can be traced to improper belt storage procedures that damaged the belt before it was installed on the drive. By following a few guidelines, these types of belt failures can be avoided.

Recommended
Belts should be stored in a cool and dry environment with no direct sunlight. Ideally, belts should be stored at less than 85° F and with lower than 70% relative humidity.

Belts should be stored in original packaging.

Not Recommended
Belts should not be stored near windows, which may expose the belts to direct sunlight or moisture.

Belts should not be stored near heaters, radiators, or in the direct airflow of heating devices.

Belts should not be stored near any devices that generate ozone such as transformers and electric motors.

Belts should not be stored where they are exposed to solvents or chemicals in the atmosphere.

Do not store belts on the floor unless they are in a protective container. Floor locations are exposed to traffic that may damage the belts.

Do not crimp belts during handling or while being stored. To avoid this, belts must not be bent to diameters smaller than what is recommended (minimum recommended sprocket diameter for inside bends and 1.3 times the minimum recommended sprocket diameter for back side bends). Do not use ties or tape to pull belt spans tightly together near the end of the belt. Do not hang on a small diameter pin that suspends all of the belt weight and bends the belt to a diameter smaller than the minimum recommended sprocket diameter. Improper storage will damage the tensile cord and the belt will fail prematurely. Handle belts carefully when removing from storage and moving to the application.

Storage Effects
Belts may be stored up to six years if properly stored at temperatures less than 85°F and relative humidity less than 70%.

For every 15°F increase in storage temperature above 85°F, the time the belt can be stored without reduced performance decreases by one-half. Belts should never be stored at temperatures above 115°F.

At relative humidity levels above 70%, fungus or mildew may form on stored belts. This has minimal affect on belt performance, but should be avoided if possible.

When equipment is stored for prolonged periods of time (over six months), the belt tension should be relaxed so that the belt does not take a set, and the storage environment should meet the 85°F and 70% or less relative humidity condition. If this is not possible, belts should be removed and stored separately in a proper environment.

2. Center Distance and Belt Length

The approximate relationship between a center distance and belt pitch length is given by the following formula:

Formula 2

$$L_p = 2C + 1.57(D + d) + \frac{(D - d)^2}{4C}$$

Where: L_p = belt pitch length, inches
D = diameter of large sprocket, inches
d = diameter of small sprocket, inches
C = center distance, inches

A more precise formula is given below:

Formula 3

$$L_p = 2C \cos \varphi + \frac{\pi (D + d)}{2} + \frac{\pi \varphi (D - d)}{180}$$

Where: L_p = belt pitch length, inches
C = center distance, inches
D = pitch diameter of large sprocket, inches
d = pitch diameter of small sprocket, inches

$$\varphi = \sin^{-1}\left(\frac{D - d}{2C}\right) \text{ degrees}$$

The approximate center distance can be found by this formula:

Formula 4

$$C = \frac{K + \sqrt{K^2 - 32 (D - d)^2}}{16}$$

Where: $K = 4L_p - 6.28 (D + d)$
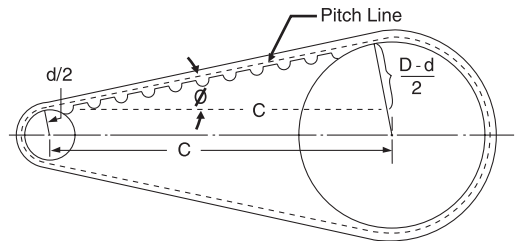
The exact center distance can be calculated using an iterative process between the center distance (Formula 4) and belt length (Formula 3) equations. The exact center distance has been found when the two equations converge. The pitch length increment of a synchronous belt is equal to a multiple of the belt pitch.

3. Tooth Profiles

Conventional trapezoidal belts (MXL, XL, etc.) were the earliest developments of positive drive belts. In more recent years, new curvilinear profiles have entered the market. The most predominant of these profiles is the HTD® system (5mm, 8mm, etc.). While these curvilinear profiles provide many advantages, they also can provide significant disadvantages.

With the development of the Gates GT® tooth profile, the combined advantages of the various curvilinear profiles have now been optimized. Characteristics such as ratcheting resistance, improved load/life and noise reduction were prime factors in the design of the Gates GT profile. Additionally, it allowed optimization in incorporating premium materials into its superior construction.

The GT tooth profile is based on the tractix mathematical function. Engineering handbooks describe this function as a "frictionless" system. This early development by Schiele is described as an involute form of a catenary. With this system, the belt and sprocket teeth move substantially tangentially during entry and exit, thus improving significantly

the belt's performance characteristics. This is illustrated in Fig. 6. For information on belt/sprocket interchangeability between various Gates products as well as interchange with other manufacturers, consult Gates Belt/Sprocket Interchange Guide (12998-B) or contact Gates Power Transmission Product Application.
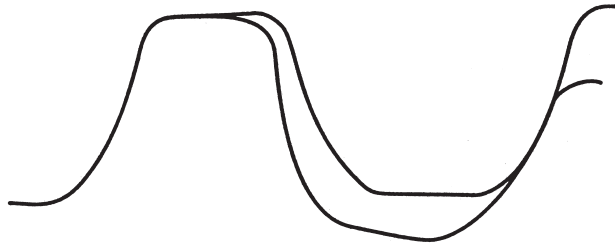
Figure 6

4. Static Conductivity

Static discharge can pose a hazard with belt drives that operate in potentially explosive environments. Static discharge can also interfere with radios, electronic instruments, or controls used in a facility. While uncommon, static discharge can also cause bearing pitting if the discharge travels through bearings. Effectively dissipating static charges from belt drives can prevent these issues. One way of achieving this is with conductive belt constructions.

Industry recognized standards for belt conductivity specify test procedures as well as allowable measured values for belt resistance. The Association for Rubber Products Manufacturers (ARPM, formerly RMA) publication IP-3-3 applies to both V-belts and synchronous belts. The international Standards Organization (ISO) standard 9563 applies to synchronous belts only, and is recognized more broadly on an international basis.

PowerGrip® Timing and PowerGrip HTD® belts in endless, Twin Power® and long length belting forms do not meet the static conductivity requirements specified in ARPM Bulletin IP 3-3 nor ISO 9563 so are not considered to be static conductive. These belts can be manufactured in a static conductive construction on a made-to-order basis.

8M and 14M PowerGrip GT®3 belts are conductive in accordance with ISO 9563, but not 12 mm wide 8M PowerGrip GT3 belts. 5M PowerGrip GT3 and 5M, 8M and 14M Twin Power and long length belting forms do not meet static conductivity requirements, but can be manufactured in a static conductive construction on a made-to-order basis.

When belts are used in hazardous environments, additional considerations should be given to assure that accidental static spark discharges do not occur. Note that industry ARPM IP-3-3 and ISO 9563 standards for belt conductivity apply only to new belts. Conductivity properties is known to decline over time after belts are placed into service, but are not generally monitored. Also note that dissipation of static charges to ground occurs initially between belt teeth and the sprockets. Unusual or excessive debris or contaminants on belt contact surfaces or sprocket grooves may hinder the conductivity of static charges, so should be cleaned and removed. Finally

note that a conductive path all the way from the sprockets through shafts, bearings, structure and other system components to ground is critical in safely dissipating static charges.

As an additional measure of protection in hazardous environments, a static-conductive brush or similar device should be employed to bleed off any residual static buildup that might remain around the belt. The user must ensure that belt drives operating in potentially hazardous or explosive environments are designed and installed in accordance with existing building codes, OSHA requirements, and/or recognized safety-related organizations.

5. Sprocket Diameter—Speed
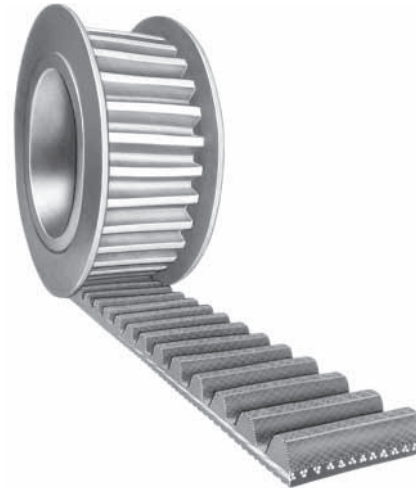
Drives shaded in the Belt Width Selection Tables use sprocket diameters that may reduce belt life. The amount of reduction will depend on speed—the higher the speed, the greater the reduction. The drives are included for use where speed ratio or space requirements must be met. Blanks in the lower right-hand portions of the Belt Width Selection Tables occur because sprocket rim speed exceeds 6,500 feet per minute. Centrifugal forces developed beyond this speed may prohibit the use of stock gray cast iron sprockets. For rim speeds above 6,500 feet per minute, contact Gates Power Transmission Product Application for other alternatives.

Sprockets Recommended
For maximum performance, we recommend using Gates PowerGrip® belts only with Gates PowerGrip® Sprockets

6. Efficiency

When properly designed and applied, PowerGrip belt drive efficiency will be as high as 98%. This high efficiency is primarily due to the positive, no slip characteristic of synchronous belts. Since the belt has a thin profile, it flexes easily, thus resulting in low hysteresis losses as evidenced by low heat buildup in the belt.

Gates synchronous belts are uniquely constructed because they use high performance materials. Optimization of these high-technology features provide maximum performance and efficiency.

Synchronous belt drive efficiency can be simply defined as shown in the following equation:

When examining the loss of energy, it is necessary to con-

$$\text{Efficiency, percent} = \frac{\text{DN RPM} \times \text{DN Torque}}{\text{DR RPM} \times \text{DR Torque}} \times 100$$

sider belt losses in terms of shaft torque and shaft speed. Torque losses result from bending stress and friction. Chain drives running unlubricated may generate significant heat build up due to increased friction in the roller joints. Even properly lubricated chains running at higher speeds tend to throw off the oil due to centrifugal forces, making it difficult to maintain proper lubrication at the load bearing surfaces. Consequently, chain drives are typically only 92-98% efficient.

Speed losses result from belt slip and creep. Unlike V-belts, slip is not a factor with synchronous belts. Well maintained V-belt drives are typically in the range of 95-98% efficient. However, on a poorly designed or maintained drive, the efficiency may drop as much as 5% or more. If proper maintenance cannot be scheduled for a V-belt drive or it is located in an inaccessible area, a positive belt drive system should be considered.
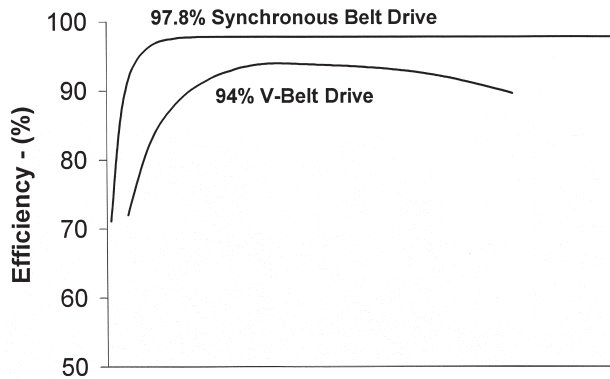
Increasing DriveN Torque

The belt drive is only part of the total system. Motors should be properly sized for the application. They must have sufficient capacity to meet the power needs, yet over-designed motors will lead to electrical inefficiencies. DriveN machines also may have inherent inefficiencies which may contribute to overall system efficiency.

7. Belt Tolerances

These tolerances are for reference only. For fixed center drive applications and special tolerances, contact Gates Power Transmission Product Application.

Stock Belt Center Distance Tolerances				
Belt Length		(mm) (in)	Center Distance Tolerance	(mm) (in)
over 127 5	to	254 10	±	0.20 .008
over 254 10	to	381 15	±	0.23 .009
over 381 15	to	508 20	±	0.25 .010
over 508 20	to	762 30	±	0.30 .012
over 762 30	to	1016 40	±	0.33 .013
over 1016 40	to	1270 50	±	0.38 .015
over 1270 50	to	1524 60	±	0.41 .016
over 1524 60	to	1778 70	±	0.43 .017
over 1778 70	to	2032 80	±	0.46 .018
over 2032 80	to	2286 90	±	0.49 .019
over 2286 90	to	2540 100	±	0.52 .020
over 2540 100	to	2794 110	±	0.54 .021
over 2794 110	to	3048 120	±	0.56 .022
over 3048 120	to	3302 130	±	0.58 .023
over 3302 130	to	3556 140	±	0.60 .024
over 3556 140	to	3810 150	±	0.63 .025
over 3810 150	to	4064 160	±	0.66 .026
over 4064 160	to	4318 170	±	0.69 .027
over 4318 170	to	4572 180	±	0.72 .028
over 4572 180			add ± .03 .001 for	
			every 254 10 increment	

Stock Belt Width Tolerances							
	Belt Width Tolerances						
Belt Width (mm) (in)	**Belt Lengths** (mm) (in) 0 to 0 838 33		**Belt Lengths** over (mm) (in) 838 to 33 1676 66		**Belt Lengths** over (mm) (in) 1676 66		
over 11.1 to 38.1 0.438 1.500	+ .8 0.032	− .8 0.032	+ .8 0.032	− 1.2 0.047	+ .8 0.032	− 1.2 0.047	
over 38.1 to 50.8 1.500 2.000	+ .8 0.032	− 1.2 0.047	+ 1.2 0.047	− 1.2 0.047	+ 1.2 0.047	− 1.6 0.063	
over 50.8 to 63.5 2.000 2.500	+ 1.2 0.047	− 1.2 0.047	+ 1.2 0.047	− 1.6 0.063	+ 1.6 0.063	− 1.6 0.063	
over 63.5 to 76.5 2.500 3.000	+ 1.2 0.047	− 1.6 0.063	+ 1.6 0.063	− 1.6 0.063	+ 1.6 0.063	− 2.9 0.078	
over 76.2 to 101.6 3.000 4.000	+ 1.6 0.063	− 1.6 0.063	+ 1.6 0.063	− 2.0 0.078	+ 2.0 0.078	− 2.0 0.078	
over 101.6 to 177.8 4.000 7.000	+ 2.4 0.094	− 2.4 0.094	+ 2.4 0.094	− 2.8 0.109	+ 2.4 0.094	− 3.2 0.125	
over 177.8 to 7.000					+ 4.8 0.188	− 6.4 0.250	

8. Belt Installation Tension

Standard Belt Tensioning Procedure

When installing a Gates PowerGrip® belt:

A. Be sure it is tensioned adequately to prevent tooth jumping (ratcheting) under the most severe load conditions which the drive will encounter during operation.

B. Avoid extremely high tension which can reduce belt life and possibly damage bearings, shafts and other drive components.

The proper way to check belt tension is to use a tension tester. Gates has a variety of tension testers, ranging from the simple spring scale type tester to the sophisticated Sonic Tension Meter. The spring scale type tester is used by measuring how much force is required to deflect the belt at the center of its span by a specified distance (force deflection method), as shown in the sketch below.
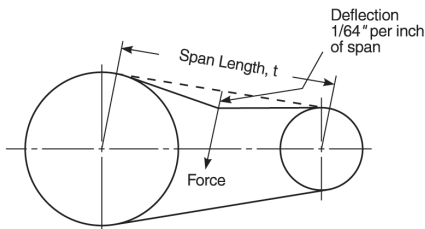
The Sonic Tension Meter measures the vibration of the belt span and instantly converts the vibration frequency into belt static tension (span vibration method).

When you wish to use a numerical method for calculating recommended belt installation tension values, the following procedure may be used.

STEP 1: Calculate the required base static installation tension.

Use Formula 5 to calculate the required base static installation tension.

Formula 5

$$T_{st} = \frac{20HP}{S} + MS^2$$

Where: T_{st} = base static installation tension, Pounds
HP = Horsepower
$S = \frac{PD \times RPM}{3820}$
M = Value from Table 3
PD = Sprocket Pitch Diameter, inches
RPM = Sprocket revolutions per minute

Table 3

Pitch	Belt Width	M	Y	Min T_{st} (lb) per span
5M PowerGrip GT3	9mm 15mm 25mm	0.17 0.29 0.48	14.88 24.80 41.33	9.0 15.0 25.0
8M PowerGrip GT3	12mm 20mm 30mm 50mm 85mm	0.32 0.54 0.81 1.35 2.29	25.38 42.29 63.44 105.73 179.74	13.5 22.5 33.8 56.2 95.6
14M PowerGrip GT3	40mm 55mm 85mm 115mm 170mm	1.80 2.48 3.83 5.18 7.66	93.04 127.93 197.72 267.50 395.43	106.0 145.8 225.2 304.8 450.5
5M PowerGrip HTD	15mm 25mm	0.27 0.45	24.80 41.33	13.5 22.5
20M PowerGrip HTD	115mm 170mm 230mm 290mm 340mm	6.84 10.11 13.68 17.25 20.23	402.35 594.78 804.70 1014.63 1189.56	425.5 629.0 851.0 1073.0 1258.0
XL PowerGrip Timing	1/4 in. 3/8 in.	0.07 0.11	3.30 4.94	3.2 4.9
L PowerGrip Timing	1/2 in. 3/4 in. 1 in.	0.19 0.28 0.38	7.20 10.80 14.40	12.5 18.8 25.0
H PowerGrip Timing	3/4 in. 1 in. 1-1/2 in. 2 in. 3 in.	0.35 0.46 0.69 0.92 1.38	32.23 42.97 64.45 85.94 128.91	56.2 75.0 112.5 150.0 225.0
XH PowerGrip Timing	2 in. 3 in. 4 in.	2.67 4.00 5.34	129.88 194.82 259.77	210.0 315.0 420.0
XXH PowerGrip Timing	2 in. 3 in. 4 in. 5 in.	3.52 5.28 7.04 8.80	144.04 216.06 288.09 360.11	260.0 390.0 520.0 650.0

Because of the high performance capabilities of PowerGrip belts, it is possible to design drives that have significantly greater load than are necessary to carry the actual design load. Consequently, Formula 5 can provide T_{st} values less than are necessary for the belt to operate properly, resulting in poor belt performance and reduced service life.

If a more appropriately sized drive cannot be designed, minimum recommended T_{st} values are provided in Table 3 to assure that the PowerGrip® belts function properly when lightly loaded.

Always use the greater T_{st} value; i.e., from T_{st} Formula 5 or Table 3.

NOTE: When applying static belt tension values directly, multiply the required base static installation tension(T_{st}) calculated in Formula 5 by the following factors:

For New Belts:
   Minimum Static Tension = 1.0 x T_{st}
   Maximum Static Tension = 1.1 x T_{st}

For Used Belts:
   Minimum Static Tension = 0.7 x T_{st}
   Maximum Static Tension = 0.8 x T_{st}

STEP 2: Calculate the minimum and maximum recommended deflection forces.

A. Measure the span length of your drive (see sketch).

$$t = \sqrt{C^2 - \left(\frac{D-d}{2}\right)^2}$$
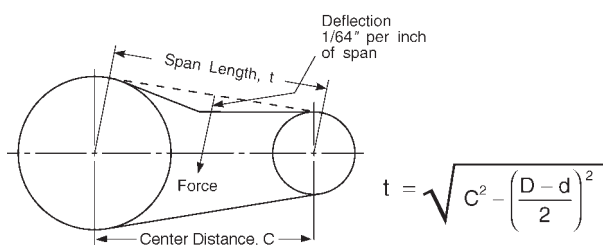
B. New belt minimum recommended force:

Formula 6

$$\text{deflection force, Min.} = \frac{1.0\,T_{st} + \left(\frac{t}{L}\right)Y}{16}, \text{lb}_f$$

C. New belt maximum recommended force:

Formula 7

$$\text{deflection force, Max.} = \frac{1.1\,T_{st} + \left(\frac{t}{L}\right)Y}{16}, \text{lb}_f$$

Where: T_{st} = Base Static tension, lb_f
     t = span length, inches
     L = belt pitch length, inches
     Y = constant from Table 3

USED BELT NOTE: For re-installation of a used belt, a recommended tension of 0.7 T_{st} to 0.8 T_{st} value should be used in calculating the deflection forces, instead of the 1.0 T_{st} to 1.1 T_{st} shown for new belts.

STEP 3: Applying the tension.

Force deflection tension method
A. At the center of the span (t) apply a force perpendicular to the span large enough to deflect the belt on the drive 1/64 inch per inch of span length from its normal position. One sprocket should be free to rotate. Be sure the force is applied evenly across the entire belt width. If the belt is a wide synchronous belt place a piece of steel or angle iron across the belt width and deflect the entire width of the belt evenly.
B. Compare this deflection force with the range of forces calculated in Step 2.
  1. If it is less than the minimum recommended deflection force, the belt should be tightened.
  2. If it is greater than the maximum recommended deflection force, the belt should be loosened.

Span vibration tension method
The Sonic Tension Meter detects the vibration frequency in the belt span, and converts that measurement into the actual static tension in the belt. To use the Sonic Tension Meter, begin by entering the belt unit weight, belt width, and the span length. To measure the span vibration, press the "Measure" button on the meter, tap the belt span, and hold the microphone approximately 1/4" away from the back of the belt. The Sonic Tension Meter will display the static tension, and can also display the span vibration frequency.

The belt unit weights for use with the Gates Sonic Tension Meter are shown in the following table.

Belt Product Family	Belt Cross Section	Adjusted Belt Weight (grams/meter)
PowerGrip® Timing	XL	2.4
	L	3.2
	H	3.9
	XH	11.3
	XXH	14.9
PowerGrip Timing Twin Power®	XL	1.9
	L	3.2
	H	4.6
PowerGrip GT®3 (5M, 8M,14M) and HTD® (20M)	5M	5.8
	8M	5.5
	14M	9.7
	20M	12.8
PowerGrip GT2 Twin Power	8M	6.93
	14M	11.44

9. Center Distance Allowances for Installation and Tensioning

Since fixed center drives are not recommended, center distance allowances for a Gates PowerGrip® belt drive are necessary to assure that the belt can be installed without damage and then tensioned correctly. The standard installation allowance is the minimum decrease in center distance required to install a belt when flanged sprockets are removed from their shafts for belt installation. This is shown in the first column of Table 4. This table also lists the minimum increase in center distance required to assure that a belt can be properly tensioned over its normal lifetime. If a belt is to be installed over flanged sprockets without removing them, the additional center distance allowance for installation shown in the second table below must be added to the first table data.

Table 4

Center Distance Allowance For Installation and Tensioning

Length Belt (mm) (in)	Standard Installation Allowance (Flanged Sprockets (mm) Removed For Installation) (in)	Tensioning Allowance (All Drives) (mm) (in)
Up to 125 5	0.5 0.02	0.5 0.02
Over 125 to 250 5 10	0.8 0.03	0.8 0.03
Over 250 to 500 10 20	1.0 0.04	0.8 0.03
Over 500 to 1000 20 40	1.8 0.07	0.8 0.03
Over 1000 to 1780 40 70	2.8 0.10	0.8 0.04
Over 1780 to 2540 70 100	3.3 0.13	1.0 0.04
Over 2540 to 3300 100 130	4.1 0.16	1.3 0.05
Over 3300 to 4600 130 180	4.8 0.19	1.3 0.05
Over 4600 to 6900 180 270	5.6 0.22	1.3 0.05

Additional Center Distance Allowance For Installation Over Flanged Sprockets*

(Add to Installation Allowance In Table No. 4)

Pitch	One Sprocket (mm) Flanged (in)	Both Sprockets (mm) Flanged (in)
0.080" (MXL)	8.4 0.33	12.4 0.49
0.200" (XL)	11.7 0.46	18.0 0.71
0.375" (L)	16.3 0.64	21.6 0.85
0.500" (H)	16.3 0.64	24.4 0.85
5mm	13.5 0.53	19.1 0.75
8mm	21.8 0.86	33.3 1.31
14mm	31.2 1.23	50.0 1.97
20mm	47.0 1.85	77.5 3.05

* For drives that require installation of the belt over one sprocket at a time, use the value for "Both Sprockets Flanged"
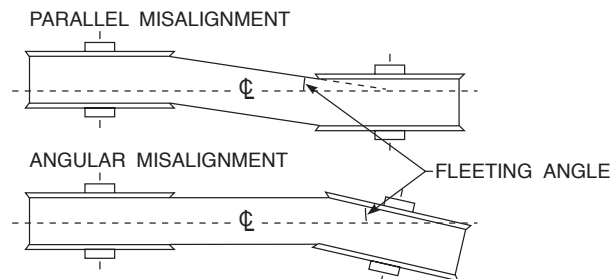
10. Drive Alignment

Provision should be made for center distance adjustment, according to the two tables on this page, or to change the idler position so the belt can be slipped easily onto the drive. When installing a belt, never force it over the flange. This will cause internal damage to the belt tensile member.

Synchronous belts typically are made with high modulus tensile members which provide length stability over the belt life. Consequently, misalignment does not allow equal load distribution across the entire belt top width. In a misaligned drive, the load is being carried by only a small portion of the belt top width, resulting in uneven belt wear and premature tensile failure.

There are two types of misalignment: parallel and angular (See Fig. 7). Parallel misalignment is where the driveR and driveN shafts are parallel, but the two sprockets lie in different planes. When the two shafts are not parallel, the drive is angularly misaligned.

A fleeting angle is the angle at which the belt enters and exits the sprocket, and equals the sum of the parallel and angular misalignments.

Any degree of sprocket misalignment will result in some reduction of belt life, which is not accounted for in the normal drive design procedure. Misalignment of all synchronous belt drives should not exceed 1/4° or 1/16" per foot of linear distance. Misalignment should be checked with a good straight edge or by using a laser alignment tool. The straight edge tool should be applied from driveR to driveN, and then from driveN to driveR so that the total effect of parallel and angular misalignment is made visible.

PARALLEL MISALIGNMENT

ANGULAR MISALIGNMENT

FLEETING ANGLE

Figure 7

Drive misalignment can also cause belt tracking problems. However, light flange contact by the belt is normal and won't affect performance.

For those drives in which the center distance is greater than eight times the small sprocket diameter, belt tracking can be a problem. In these cases, the parallel position of the two sprockets may need to be adjusted until only one flange guides the belt in the system and the belt tracks fully on all sprockets. Regardless of the drive center distance, the optimum drive performance will occur with the belt lightly contacting one flange in the system. The worst case is for the belt to contact flanges on opposite sides of the system. This traps the belt between opposite flanges and can force the belt into undesirable parallel misalignment. Improper installation of the bushing can result in the bush-

ing/ sprocket assembly being "cocked" on the shaft. This leads to angular misalignment and sprocket wobble. Be sure to follow the instructions provided with the bushings.

11. Belt Installation

During the belt installation process, it is very important the belt be fully seated in the sprocket grooves before applying final tension. Serpentine drives with multiple sprockets and drives with large sprockets are particularly vulnerable to belt tensioning problems resulting from the belt teeth being only partially engaged in the sprockets during installation. In order to prevent these problems, the belt installation tension should be evenly distributed to all belt spans by rotating the system by hand. After confirming that belt teeth are fully engaged in the sprocket grooves, belt tension should be rechecked and verified. Failure to do this may result in an undertensioned condition with the potential for belt ratcheting.

12. Belt Pull Calculations

When the machine designer requests shaft load calculations from the drive designer, the following procedure can be applied:

A. Calculate Belt Span Tensions

Belt pull is the vector sum of T_T and T_S, the tightside and slackside tensions. T_T and T_S may be calculated using the following formulas:

Formula 8

$$T_T = \frac{144,067\ HP}{(PD)(RPM)}$$

Formula 9

$$T_S = \frac{18,008\ HP}{(PD)(RPM)}$$

Where: HP = Horsepower
PD = Sprocket Pitch Diameter (in)
RPM = Sprocket Speed (rev/min)

B. Solution For Both Magnitude and Direction

The vector sum of T_T and T_S can be found so that the direction of belt pull, as well as magnitude, is known. This is necessary if belt pull is to be vectorially added to sprocket weight, shaft weight, etc., to find true bearing loads. In this case, the easiest method of finding the belt pull vector is by graphical addition of T_T and T_S. If only the magnitude of belt pull is needed, numerical methods for vector additions are faster to use.

If both direction and magnitude of belt pull are required, the vector sum of T_T and T_S can be found by graphical vector addition as shown in Fig. 8. T_T and T_S vectors are drawn to a convenient scale and parallel to the tightside and slackside, respectively. Fig. 8 shows vector addition for belt pull on the motor shaft. The same procedures can be used for finding belt pull on the driveN shaft. This method may be used for drives using three or more sprockets or idlers. For two-sprocket drives, belt pull on the driveR and driveN

shafts is equal but opposite in direction . For drives using idlers, both magnitude and direction may be different.

C. Solution For Magnitude Only

If only the magnitude of belt pull is needed, follow the steps below. Use this method for drives with two sprockets. Use the graphical method shown if the drive uses idlers.

1. Add T_T and T_S

2. Using the value of $\frac{D - d}{C}$ for the drive, find the vector sum correction factor using Fig. 9, where:

D = large diameter
d = small diameter
C = center distance

Or, use the arc of contact on the small sprocket if known.

3. Multiply the sum of T_T plus T_S by the vector sum correction factor to find the vector sum of T_T plus T_S.
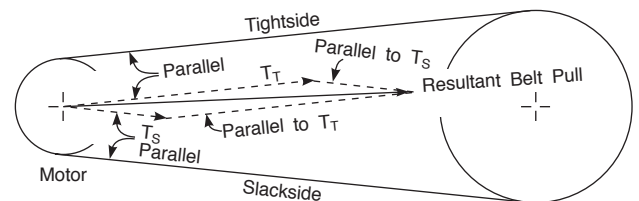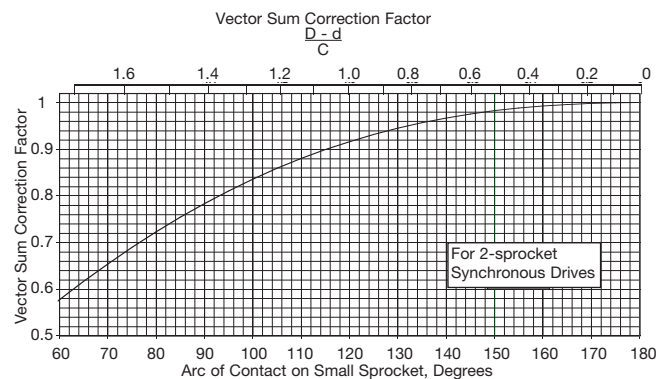

Figure 8

Figure 9

13. Bearing / Shaft Load Calculations

A. Shaft Load Calculations

If true side load on the shaft, including sprocket weight, is desired, the sprocket weight can be added to the belt pull using the same graphical method shown in Fig. 8. The sprocket weight vector is vertical toward the ground. Weights for standard sprockets are shown in the sprocket specification tables.

B. Bearing Load Calculations

In order to find actual bearing loads, it is necessary to know weights of machine components and the value of all other forces contributing to the load. However, it is sometimes desirable to know the bearing load contributed by the synchronous drive alone. Bearing loads resulting from a synchronous belt drive can be calculated knowing bearing placement with respect to the sprocket center and the shaft load as previously calculated. For rough estimates, machine designers sometimes use belt pull alone, ignoring sprocket weight. If accuracy is desired, or if the sprocket is unusually heavy, actual shaft load values including sprocket weight should be used.

A. Overhung Sprocket
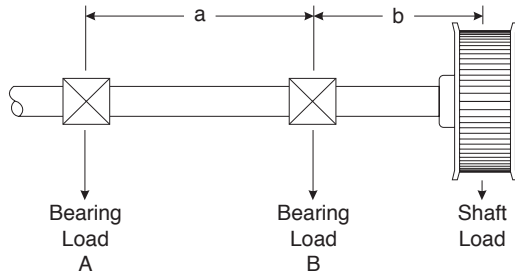
Figure 10

Formula 10

$$\text{Load at B, (lb)} = \frac{\text{Shaft Load} \times (a + b)}{a}$$

Formula 11

$$\text{Load at A, (lb)} = \text{Shaft Load} \times \frac{b}{a}$$

Where: a and b = spacing, (in), per Fig. 10

B. Sprocket Between Bearings
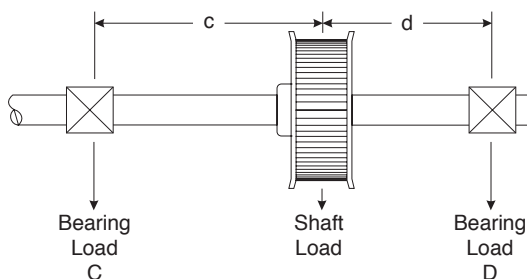
Figure 11

Formula 12

$$\text{Load at D (lb)} = \frac{\text{Shaft Load} \times c}{(c + d)}$$

Formula 13

$$\text{Load at C (lb)} = \frac{\text{Shaft Load} \times d}{(c + d)}$$

Where: c and d = spacing (in), per Fig. 11

14. Self-Generated Tension

All synchronous belt drives exhibit a self-generating or self-tightening characteristic when transmitting a load. Laboratory testing has shown this characteristic to be similar with all tooth profiles. The designer/user should be aware that self-tensioning can result in increased bearing and shaft loads and reduced drive performance; i.e., short belt life. This can be avoided by following proper tensioning procedures.

While belt overtensioning can impose higher bearing and shaft loads and lead to reduced belt life, undertensioning can result in self-tensioning. Properly designed and tensioned drives will not be significantly affected by self-generated tension.

When a belt is too loose for the design load, the self-tensioning characteristic results in the belt teeth climbing out of the sprocket grooves, leading to increased stresses on the belt teeth, accelerated tooth wear and reduced belt life. When a belt is severely undertensioned, this self-tensioning characteristic can result in the belt ratcheting (jumping teeth). When this occurs, significant shaft separation forces are instantaneously developed in the drive, resulting in damage to bearings, shafts, and other drive components including the belt.
NOTE: This is true for all synchronous belts.

Maximum drive performance and belt life are achieved when the belt is properly tensioned for the design load and maintained.

Made-to-order (MTO) Metals & PowerGrip® Belts

In addition to the stock industrial PowerGrip® belts listed in this catalog, Gates offers many special construction, made-to-order belts for use with stock sprockets. The table below lists some of them. Contact Gates for more information.

MTO BELT TYPES	APPLICATION
Alternate tensile member	Special applications: i.e., low rpm, shock loads and precise registration.
Nonstock widths and/or lengths in stock pitches	When exact width or length is required.
High temperature	Dry operation from –40°F to 230°F (–40°C to 110°C)
Oil resistance	For excessively oily conditions, including immersion in commercial motor oil. Temperature range: in oil, –20°F to 240°F (–29°C to 116°C); dry, –20°F to 210°F (–29°C to 99°C)
Static dissipating	Resistance of 6 megohms or less.
Low temperature	Dry temperature operation from –65°F to 180°F (–54°C to 82°C)
Nonmarking backing	For conveyors, food handling, etc., with taste and toxicity subject to customer approval.
PowerPainT™	Paint & Varnish Applications
Precision ground backing	Special applications involving a critical overall belt thickness dimension.
Special thickness rubber backing	For functional and other applications where belt back may require special thickness, durometer or material.
Special tracking	When belt must track in a specific direction.

In addition to the stock industrial PowerGrip® hardware listed in this catalog, Gates offers many additional Made-to-order options. The table below lists some of the available options. For more information, contact the Gates Made-to-order Metals Team at 1-800-709-6001 or via fax at 303-744-4080, or via email at makemymetal@gates.com.

Made-to-order (MTO) Metals	
Pulleys, Sheaves and Sprockets	All Gates Synchronous Profiles and Pitches, Micro-V® and V-belt, Plain or Profiled Idlers
Bores	Plain, Straight, Tapered, Splined or any special bore. Manufactured to accept Taper-Lock®*, Ringfeder®*, QD®*, Torque Tamer, Trantorque®* or other special bushings
Styles	Bar Stock, Idlers, Ringfeder Connections, Torque Tamers, Custom Configurations, Special Hubs and more
Materials	Aluminum, Steel, Ductile Iron, Cast Iron, Phenolic, Stainless Steel or Plastics
Finishes	Hard Coat, Food Grade, Zinc, Black Anodize, Painted, Custom Plating or any special coatings
Processes	Hob Cutting, Shaper Cutting, Die Casting and Molding
Other Services	Sub-Assemblies, Press Bearings, Sprocket/Bushing Balancing and Index Marking

* Taper-Lock is a trademark of Reliance Electric.
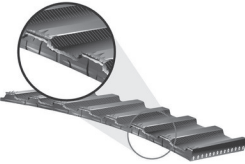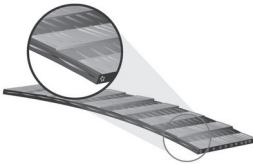
* Ringfeder is a trademark of Ringfeder Corporation.

* Trantorque is a trademark of BTL, a subsidiary of Fenner PLC.

* QD is a trademark of Emerson Electric.

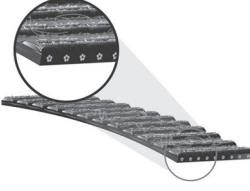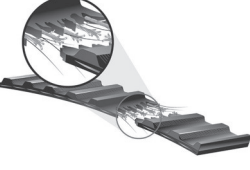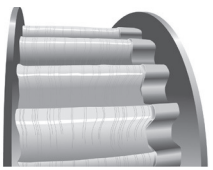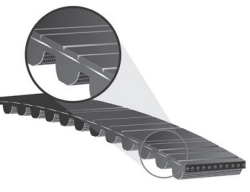* Torque Tamer is a trademark of Reliance Electric.

Troubleshooting

Symptom	Diagnosis	Possible Remedy
Unusual noise	Misaligned drive Too low or high belt tension Backside idler Worn sprocket Bent guide flange Belt speed too high Incorrect belt profile for the sprocket (i.e., GT® etc.) Subminimal diameter Excess load	Correct alignment Adjust tension to recommended value Use inside idler Replace sprocket Replace sprocket/flange Redesign drive Use proper Gates PowerGrip® GT®3 belt/sprocket Redesign drive using larger diameters Redesign drive for increased capacity
Tension loss	Weak support structure Excessive sprocket wear Fixed (nonadjustable) centers Excessive debris Excessive load Subminimal diameter Belt, sprockets or shafts running too hot Unusual belt degradation, such as softening or melting	Reinforce the structure Use alternate sprocket material Use inside idler for belt adjustment Protect drive Redesign drive for increased capacity Redesign drive using larger diameters Check for conductive heat transfer from prime mover Reduce ambient drive temperature to 180°F maximum
Belt tracking	Belt running partly off unflanged sprocket Centers exceed 8 times small sprocket diameter and the large sprocket is not flanged. Excessive belt edge wear	Correct alignment Correct parallel alignment to set belt to track on both sprockets Correct alignment
Flange failure 	Belt forcing flanges off	Correct alignment or properly secure flange to sprocket
Excessive belt edge wear 	Damage due to handling Flange damage Belt too wide Belt tension too low Rough flange surface finish Improper tracking Belt hitting drive guard or bracketry	Follow proper handling instructions Repair flange or replace sprocket Use proper width sprocket Adjust tension to recommended value Replace or repair flange (to eliminate abrasive surface) Correct alignment Remove obstruction or use inside idler
Tooth shear 	Excessive shock loads Less than 6 teeth-in-mesh Extreme sprocket runout Worn sprocket Backside idler Incorrect belt profile for the sprocket (i.e., GT®, etc.) Misaligned drive Belt undertensioned	Redesign drive for increased capacity Redesign drive Replace sprocket Replace sprocket Use inside idler Use proper Gates PowerGrip® GT®3 belt/sprocket Correct alignment Adjust tension to recommended value

Gates.com/pt

Troubleshooting

Symptom	Diagnosis	Possible Remedy
Premature tooth wear	Too low or high belt tension Belt running partly off unflanged sprocket Misaligned drive Incorrect belt profile for the sprocket (i.e., GT®, etc.) Worn sprocket Rough sprocket teeth Damaged sprocket Sprocket not to dimensional specification Belt hitting drive bracketry or other structure Excessive load Insufficient hardness of sprocket material Excessive debris Cocked bushing/sprocket assembly	Adjust tension to recommended value Correct alignment Correct alignment Use proper Gates PowerGrip® GT®3 belt/sprocket Replace sprocket Replace sprocket Replace sprocket Replace sprocket Remove obstruction or use inside idler Redesign drive for increased capacity Use a more wear-resistant material Protect belt Install bushing per instructions
Tensile break	Excessive shock load Subminimal diameter Improper belt handling and storage prior to installation Debris or foreign object in drive Extreme sprocket runout Too low or high belt tension	Redesign drive for increased capacity Redesign drive using larger diameters Follow proper handling and storage procedures Protect drive Replace sprocket Adjust tension to recommended level
Unusual sprocket wear	Sprocket has too little wear resistance (i.e., plastic, aluminum, softer metals) Misaligned drive Excessive debris Excessive load Too high, too low belt tension Incorrect belt profile (i.e. GT, etc.)	Use alternate sprocket material Correct alignment Protect drive Redesign drive for increased capacity Adjust tension to recommended value Use proper Gates PowerGrip GT®3 belt/sprocket
Belt cracking	Subminimal diameter Backside idler Extreme low temperature startup Extended exposure to harsh chemicals Cocked bushing/sprocket assembly	Redesign drive using larger diameters Use inside idler Preheat drive environment Protect drive Install bushing per instructions
Excessive temperature (belt, bearing, housing, shafts, etc.)	Misaligned drive Too low or too high belt tension Incorrect belt profile (i.e. GT, etc.)	Correct alignment Adjust tension to recommended value Use proper Gates PowerGrip GT®3 belt/sprocket
Vibration	Incorrect belt profile for the sprocket (i.e. GT, etc.) Too low or too high belt tension Bushing or key loose	Use proper Gates PowerGrip GT®3 belt/sprocket Adjust tension to recommended value Check and reinstall per instructions

Standard Calculations

Required	Given	Formula
Speed ratio (R)	Shaft speeds (rpm)	$R = \dfrac{\text{rpm (faster shaft speed)}}{\text{rpm (slower shaft speed)}}$
	Pulley diameter (D & d)	$R = \dfrac{\text{D (larger pulley diameter)}}{\text{d (smaller pulley diameter))}}$
	Number of pulley grooves (N & n)	$R = \dfrac{\text{N (larger pulley groove no.)}}{\text{n (smaller pulley groove no.)}}$
Horsepower (hp) (33,000 lb-ft/min)	Torque (T) in lb-in Shaft speed (rpm)	$hp = \dfrac{T \times rpm}{63{,}025}$
	Effective tension (Te) in lb. Belt velocity in fpm	$hp = \dfrac{Te \times V}{33{,}000}$
Design horsepower (Dhp)	Rated horsepower (hp) Service factor (SF)	Dhp = hp x SF
Power (kw)	Horsepower (hp)	kw = .7457 x hp
Torque (T) in lb-in	Shaft horsepower (hp) Shaft speed (rpm)	$T = \dfrac{63{,}025 \times hp}{rpm}$
	Effective tension (Te) in lbs Pulley radius (R) in inches	T = Te x R
Torque (T) in N-mm	Torque (T) in lb-inches	T(N - mm) = 112.98 x T(lb - in)
Belt velocity in ft/min	Pulley pd in inches Pulley speed in rpm	$V = \dfrac{pd \times rpm}{3.82}$
Belt velocity in m/s	Pulley pd in mm Pulley speed in rpm	V = .0000524 x pd x rpm
Belt pitch length (PL) in inches (approximate)	Center distance (C) in inches Pulley diameters (D & d) in inches	$PL = 2C + [\,1.57 \times (D + d)\,] + \dfrac{(D - d)^2}{4C}$
Arc of contact on smaller pulley (A/Cs)	Pulley diameters (D & d) in inches Center distance (C) in inches	$A/Cs = 180 - \left[\dfrac{(D - d) \times 60}{4C}\right]$
Torque (T) due to flywheel effect (WR^2) in lb-inches (accel. and/or decel.)	Final speed (RPM) Initial speed (rpm) Flywheel effect (WR^2) in lbs-ft^2 Time (t) in seconds	$T = \dfrac{.039 \times (RPM - rpm) \times WR^2}{t}$
Flywheel effect (WR^2) in lb-ft^2	Face width of rim (F) in inches Material density (Z) in lbs/in^3 Outside rim diameter (D) in inches Inside rim diameter (d) in inches	$WR_2 = \dfrac{F \times Z \times (D^4 - d^4)}{1467}$

Useful Formulas and Calculations

Power Transmission Conversions

FORCE CONVERSION CONSTANTS

Metric to U.S.
Newtons x 3.5969 = Ounces*f*
Newtons x 0.2248 = Pounds*f*
Kilograms*f* x 2.2046 = Pounds*f*

U.S. to Metric
Ounces*f* x 0.2780 = Newtons
Pounds*f* x 4.4482 = Newtons
Pounds*f* x 0.4536 = Kilograms*f*

Metric to Metric
Kilograms*f* x 9.8067 = Newtons
Newtons x 0.1020 = Kilograms*f*

TORQUE CONVERSION CONSTANTS

Metric to U.S.
Newton Meters x 141.6119 = Ounce*f* Inches
Newton Meters x 8.8508 = Pound*f* Inches
Newton Meters x 0.7376 = Pound*f* Feet

Metric to Metric
Newton Meters x 10.1972 = Kilogram*f* Centimeters
Kilogram*f* Centimeters x 0.0981 = Newton Meters
Newton Meters x 0.1020 = Kilogram*f* Meters
Kilogram*f* Meters x 9.8067 = Newton Meters

U.S. to Metric
Ounce*f* Inches x 0.0071 = Newton Meters
Pound*f* Inches x 0.1130 = Newton Meters
Pound*f* Feet x 1.3558 = Newton Meters

POWER CONVERSION CONSTANTS

Metric to U.S.
Kilowatt x 1.3410 = Horsepower
Watt x 0.0013 = Horsepower

U.S. to Metric
Horsepower x 745.6999 = Watt
Horsepower x 0.7457 = Kilowatt

VELOCITY CONVERSION CONSTANTS

Metric to U.S.
Meters per Second x 196.8504 = Feet per Minute

U.S. to Metric
Feet per Minute x 0.0057 = Meters per Second

Metric to Metric
Meters per Second x 3.6000 = Kilometers per Hour

LINEAR BELT SPEED CONVERSION CONSTANTS

Metric to U.S.
Meters per second x 196.8504 = Feet per Minute

U.S. to Metric
Feet per Minute x 0.005080 = Meters per Second
Square Miles x 2.5900 = Square Kilometers

U.S. to U.S.
Feet per Second x 60.00 = Feet per Minute
Feet per Minute x 0.0167 = Feet per Second

Other Conversions

LENGTH CONVERSION CONSTANTS

Metric to U.S.
Millimeters x 0.0394 = Inches
Meters x 39.3701 = Inches
Meters x 3.2808 = Feet
Meters x 1.0936 = Yards
Kilometers x 3280.84 = Feet
Kilometers x 0.6214 = Statute Miles
Kilometers x 0.5396 = Nautical Miles

U.S. to Metric
Inches x 25.4000 = Millimeters
Inches x 0.0254 = Meters
Feet x 0.3048 = Meters
Yards x 0.9144 = Meters
Feet x 0.0003048 = Kilometers
Statute Miles x 1.6093 = Kilometers
Nautical Miles x 1.8532 = Kilometers

AREA CONVERSION CONSTANTS

Metric to U.S.
Square Millimeters x 0.0016 = Square Inches
Square Centimeters x 0.1550 = Square Inches
Square Meters x 10.7639 = Square Feet
Square Meters x 1.1960 = Square Yards
Hectares x 2.4711 = Acres
Square Kilometers x 247.105 = Acres
Square Kilometers x 0.3861 = Square Miles

U.S. to Metric
Square Inches x 645.160 = Square Millimeters
Square Inches x 6.4516 = Square Centimeters
Square Feet x 0.0929 = Square Meters
Square Yards x 0.8361 = Square Meters
Acres x 0.4047 = Hectares
Acres x 0.004047 = Square Kilometers
Square Miles x 2.5900 = Square Kilometers

Useful Formulas and Calculations

Other Conversions — continued

WEIGHT CONVERSION CONSTANTS

Metric to U.S.
Grams x 15.4324 = Grains
Grams x 0.0353 = Ounces (Avd.)
Grams x 0.0338 = Fluid Ounces (water)
Kilograms x 35.2740 = Ounces (Avd.)
Kilograms x 2.2046 = Pounds (Avd.)
Metric Tons (1000 Kg) x 1.1023 = Net Ton (2000 lbs.)
Metric Tons (1000 Kg) x 0.9842 = Gross Ton (2240 lbs.)

U.S. to Metric
Grains x 0.0648 = Grams
Ounces (Avd.) x 28.3495 = Grams
Fluid Ounces (water) x 29.5735 = Grams
Ounces (Avd.) x 0.0283 = Kilograms
Pounds (Avd.) x 0.4536 = Kilograms
Net Ton (2000 lbs.) x 0.9072 = Metric Tons (1000 Kg)
Gross Ton (2240 lbs.) x 1.0160 = Metric Tons (1000 Kg)

DECIMAL AND MILLIMETER EQUIVALENTS OF FRACTIONS

| Inches		Millimeters	Inches		Millimeters
Fractions	Decimals		Fractions	Decimals	
1/64	.015625	.397	33/64	.515625	13.097
1/32	.03125	.794	17/32	.53125	13.494
3/64	.046875	1.191	35/64	.546875	13.891
1/16	.0625	1.588	9/16	.5625	14.288
5/64	.078125	1.984	37/64	.578125	14.684
3/32	.09375	2.381	19/32	.59375	15.081
7/64	.109375	2.778	39/64	.609375	15.478
1/8	.125	3.175	5/8	.625	15.875
9/64	.140625	3.572	41/64	.640625	16.272
5/32	.15625	3.969	21/32	.65625	16.669
11/64	.171875	4.366	43/64	.671875	17.066
3/16	.1875	4.763	11/16	.6875	17.463
13/64	.203125	5.159	45/64	.703125	17.859
7/32	.21875	5.556	23/32	.71875	18.256
15/64	.234375	5.953	47/64	.734375	18.653
1/4	.250	6.350	3/4	.750	19.050
17/64	.265625	6.747	49/64	.765625	19.447
9/32	.28125	7.144	25/32	.78125	19.844
19/64	.296875	7.541	51/64	.796875	20.241
5/16	.3125	7.938	13/16	.8125	20.638
21/64	.328125	8.334	53/64	.828125	21.034
11/32	.34375	8.731	27/32	.84375	21.431
23/64	.359375	9.128	55/64	.859375	21.828
3/8	.375	9.525	7/8	.875	22.225
25/64	.390625	9.922	57/64	.890625	22.622
13/32	.40625	10.319	29/32	.90625	23.019
27/64	.421875	10.716	59/64	.921875	23.416
7/16	.4375	11.113	15/16	.9375	23.813
29/64	.453125	11.509	61/64	.953125	24.209
15/32	.46875	11.906	31/32	.96875	24.606
31/64	.484375	12.303	63/64	.984375	25.003
1/2	.500	12.700	1	1.000	25.400

The following illustrations show a few of the many ways that PowerGrip® belt drives can be used to transmit both power and motion. Synchronous belt drive systems are amazingly versatile, yet reliable and efficient. The examples that follow utilize conventional endless, Long-Length and Twin Power® belting, all of which is readily available.
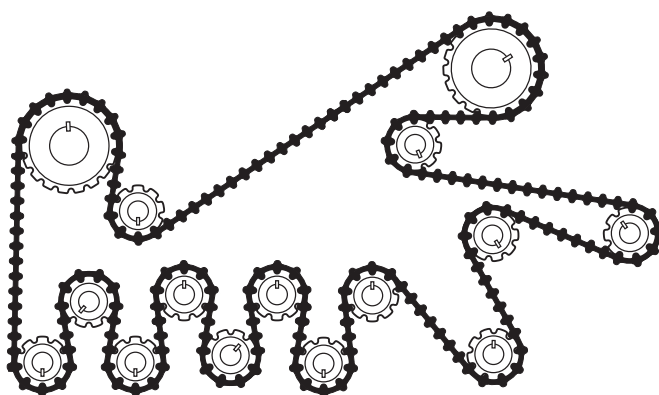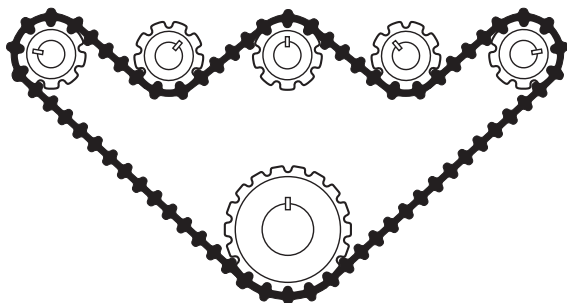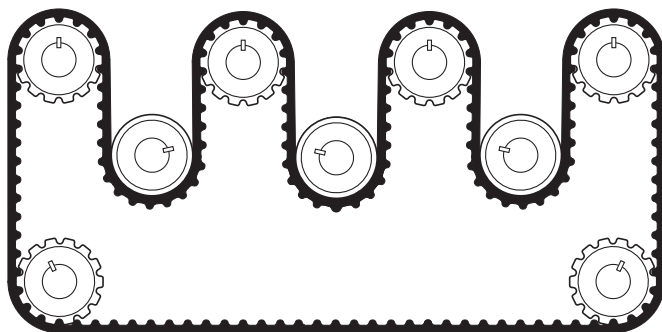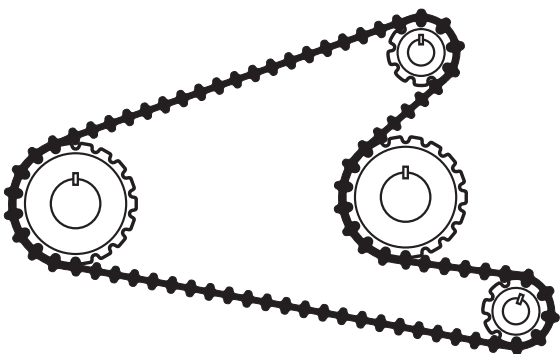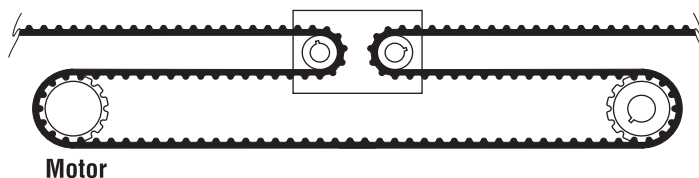
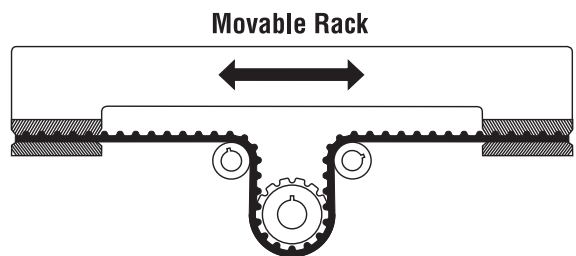## Common Drive Configurations
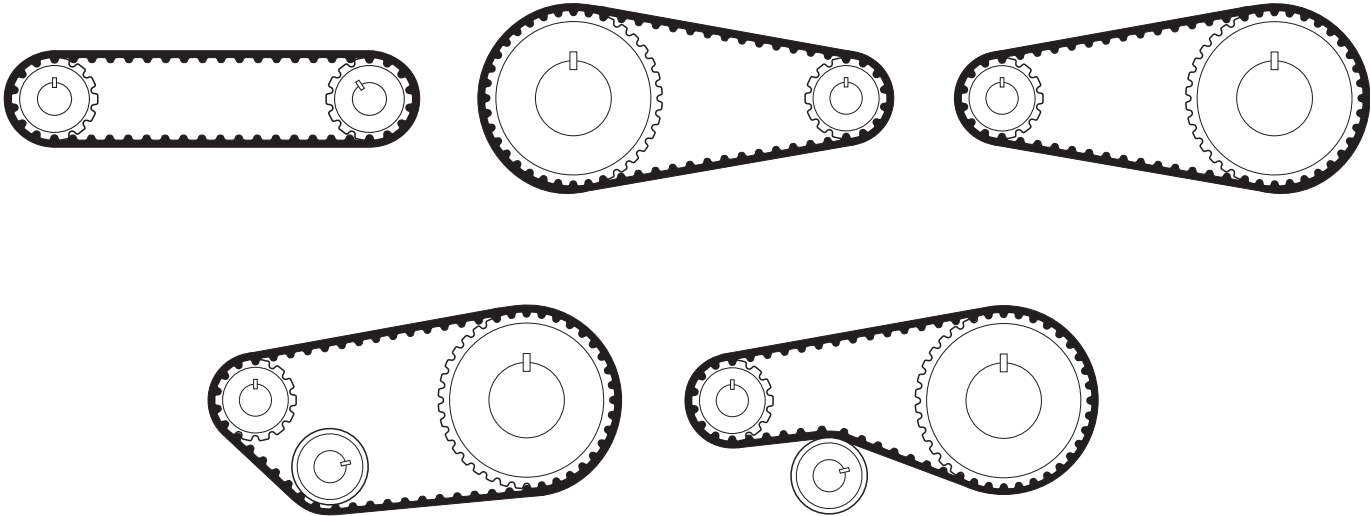
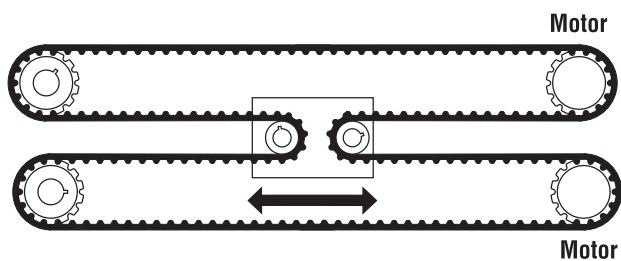## Multiple Shaft Drive Configurations
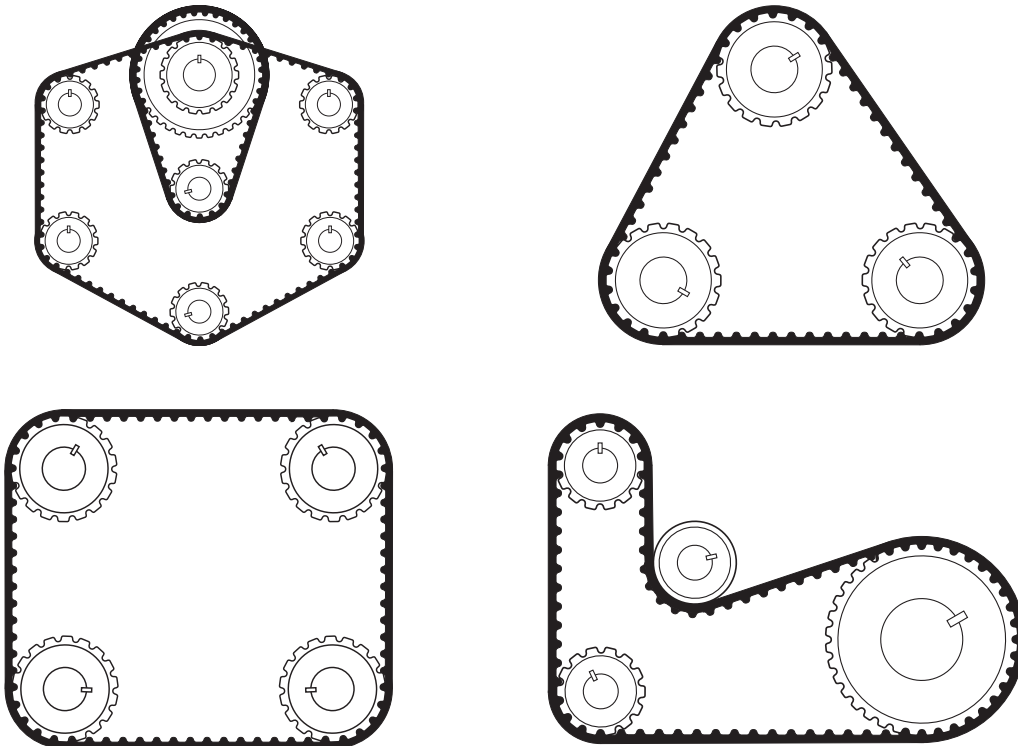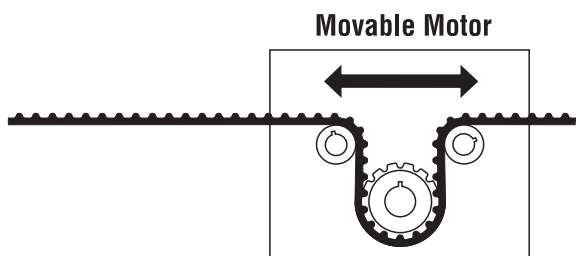
Serpentine Drive Configurations

Conveying and Material Transport Applications
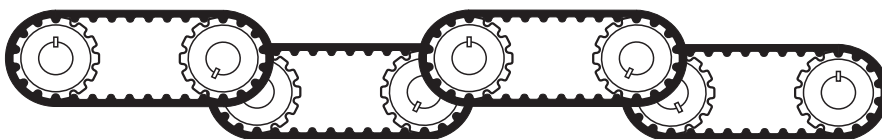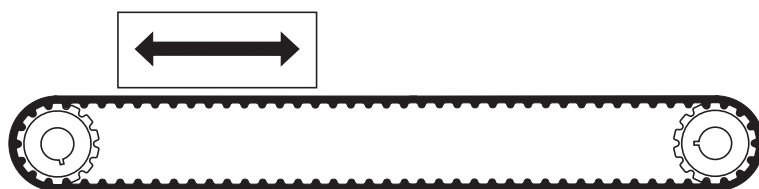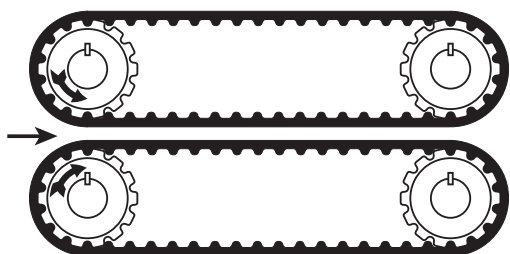
Rack and Carriage Drive Configurations

Movable Motor

Motor

Motor

Movable Rack

Motor

Long Length Drive Applications

Gates.com/pt
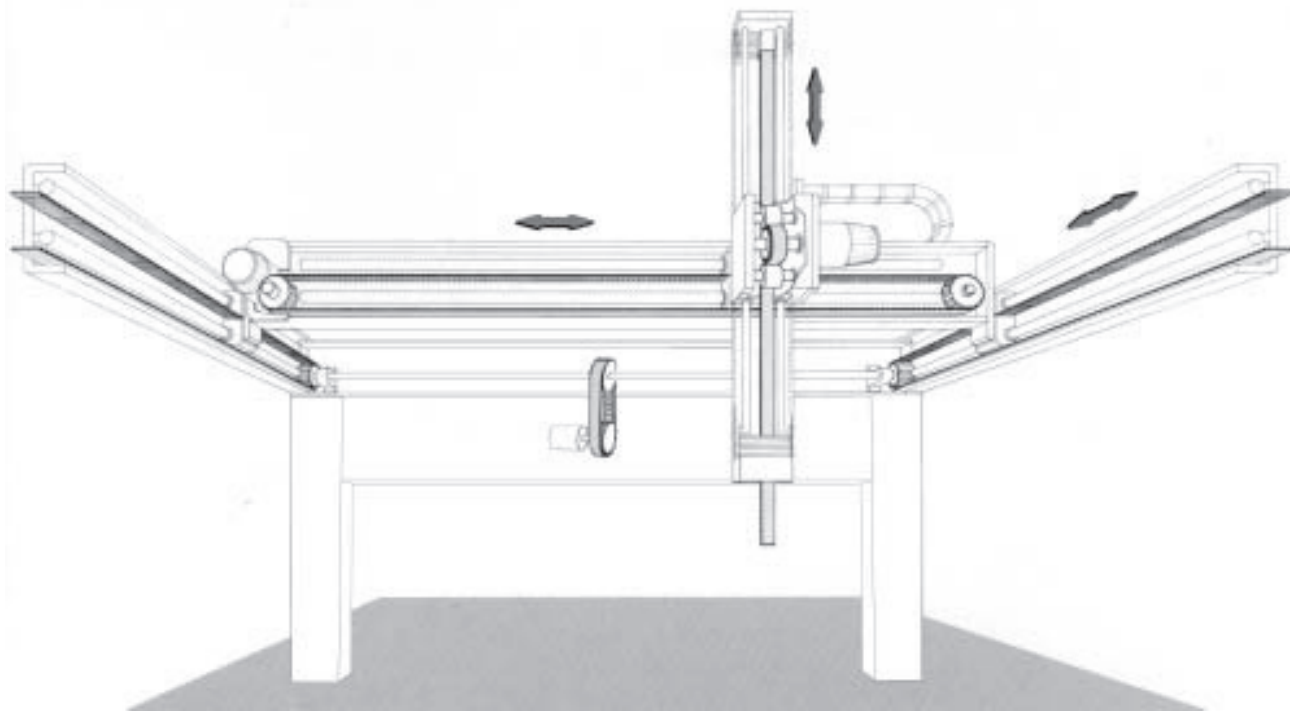
Complex Carriage Drive Configuration

Motor 1   Motor 2

Traverse
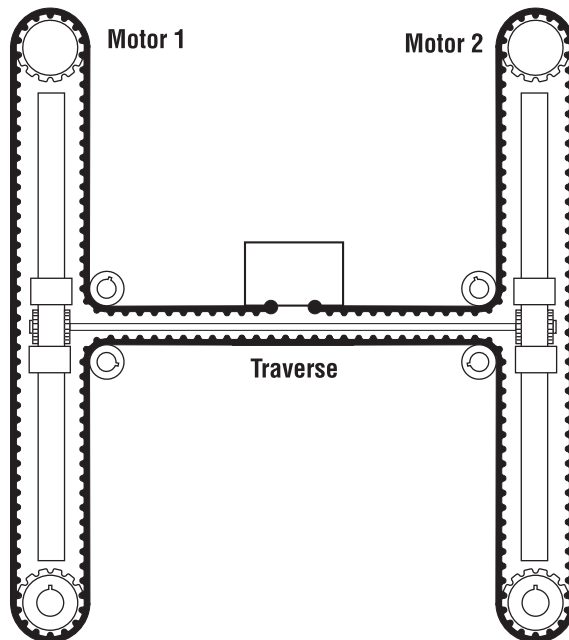
Lead Screw Drive Applications