Written by Gary R. Burger, Stevenson Memorial Hospital
Sunday, 01 April 2007 00:00

Most end users think OEMs take particular pains to design things that last. That's true in most cases, but not all.

OEMs are in business to sell products. Consequently, countless OEM drive belts are designed with neither the concept of long-term cost savings in mind, nor with the idea that a Maintenance person might need to replace components because of wear. In fact, many belt products simply are selected (and sold) on the basis of lowest price-*and the hope that they will facilitate the widest possible range of field adjustments and operate long enough to make it through the warranty period* . After that, it's up to the end user to make changes. Or, to be more precise, it's generally left to the Maintenance organization to devise a way to make a belt last longer and cost less to operate.

Rules of thumb

The application with the greatest potential cost savings is a drive that operates 24/7-365 and is larger than 1 HP. It will provide the quickest payback and net the Maintenance department the greatest credibility for the changes.

Expected life with a 24/7-365 operation is commonly three years minimum. In the view of this long-time Maintenance professional, any "100% duty cycle" drive that does not last that long desperately needs revision.

The key is to operate a drive near the "Maximum Belt Velocity" or "Rim Speed." That maximum is 6500 FPM for cast sheaves, 8000 FPM for Ductile iron and 10,000 FPM for steel. At FPMs higher than these, centrifugal forces will exceed the tensile strength of the material, risking the sheaves flying apart. The safest approach, however, is to use only the 6500 FPM limit. That's because sometime in the future, someone in your company may try to save money by using the same pitch diameter (P.D.) and install cheaper cast sheaves-*without considering the dangerous consequences associated with that decision…*

Why start by looking at Belt or Rim Velocity? Think of it this way: Would you use a really short tack hammer to pry out a 4" spike? No! You would grab the longest wrecking bar you could find because it would let you apply the greatest force with the least effort. The same is true with sheaves-*the larger the sheave diameter, the greater the length of the lever is and the easier it is to transmit power* . The only limits are the centrifugal force that is generated and the

ability of the sheave material to handle it.

With the foregoing in mind, we should expect an 1800 RPM motor to spin a maximum of a 13.7" P.D. sheave and a 3600 RPM motor to spin a maximum of a 6.9" P.D. When you look at a drive, if the motor sheave isn't close to 12" in diameter (or 6" for higher-speed motors), it is not designed for long service life. This represents a potential drive-improvement opportunity.

For other shaft RPMs, use the formula for Maximum Sheave P.D.

P.D. max = 6500/(0.2618 x RPM)

The ideal pitch will safely operate just under this maximum.

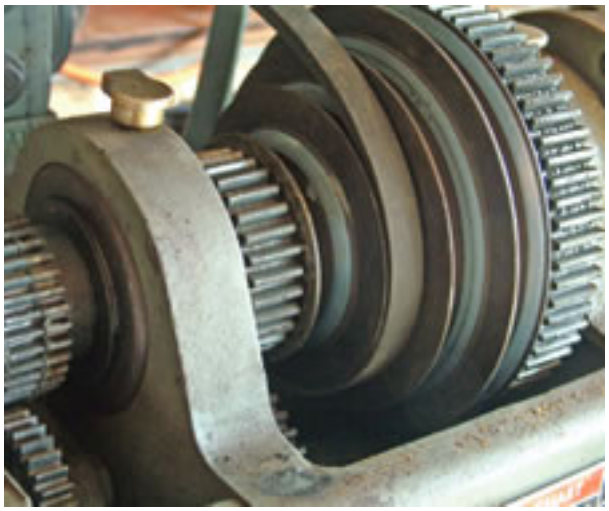
For example…
An existing drive (roughly 6"/ 8"sheaves) has a drive ratio of 1.33:1. From the sheave catalogue we find the following:

Driver Sheave on Motor P.D.	Driver Sheave on Equipment P.D.	Driven RPM	HP/Belt	Belt Velocity
3.2	4.2	1318	2.12	1466
3.8	5.0	1317	3.21	1740
4.4	5.8	1316	4.27	2016
5.0	6.6	1316	5.30	2290
6.2	8.2	1315	7.26	2840

Belt Velocity can be calculated separately using the formula (in Fpm)=P.D. x 0.21618 x Rpm

Note that a 6.2 / 8.2 P.D. sheave set is the largest listed for a 1.33 ratio, yet the belt velocity is calculated and found to be 2840. That is not very close to the previously discussed 6500 FPM maximum. Thus, there is lots of room to move closer to a 6500 FPM belt velocity.

Notice in the HP/Belt column, the same "A" profile belt can handle much more HP as the sheave sets become larger in pitch. Just think what even larger sheaves would do to the values in the HP/RPM and Belt columns?



Next, refer to the Stock Sheave listing for the full pitch range that is available. Stock sheaves in the catalogue are listed from 1.9 P.D. to 37.5 P.D.

Again using a 1800 RPM motor, the closest sheave sizes under the 13.7 P.D. maximum are 12.0, 13.0 and 13.2. These will become the new candidates for the driver sheave on the 1800 RPM motor.

The Drive Ratio we want to keep is 1.33, so the machinery performs the same as it did before we make changes. Thus, we multiply the 12.0, 13.0 and 13.2 pitch diameters by the ratio 1.33.

This gives the following results:

12.0 x 1.33 = 15.96 P.D.
13.0 x 1.33 = 17.29 P.D.
13.2 x 1.33 = 17.556 P.D.

Now, which one is close to a standard stock sheave?

15.4, 16.0 and 18.4 are listed as standard stock pitch diameters. The 15.96 calculated P.D. and 16.0 listed stock P.D. are the closest match.

Our drive is now shaping up: We have a 12.0 P.D. driver with a 16.0 P.D. driven sheave

According to the nameplate, our nominal 1800 RPM motor runs at 1725 at full load. So, let's recalculate the new drive parameters.

Recalculating the driver…
12.0 P.D. @ 1725 RPM driver speed Belt velocity = P.D. x .6218 x RPM = 12.0 x 0.2618 x 1725 = 5419.26 FPM (Safely under the 6500 FPM limit)

Recalculating the overall drive…
16.0 P.D. driven/12.0 P.D. driver = 1.3333 drive ratio (Excellent)

Recalculating the driven…
16.0 P.D. @ 5419.26 FPM Driven RPM = (driver P.D. /driven P.D.) x driver RPM = (12.0/16.0) x 1725 = 1293.75 RPM (Very close to the original measured RPM)

Dynamic pull comparison…
Now that we have the drive specifics, let's calculate dynamic belt pull. That's the actual pull that transfers power from the motor to the machine.

Using the formula:

Dynamic Pull = ((HP x 126,000)/(RPM x in. Sheave P.D.)) x 1.5 standard service factor. A 10 HP motor with the 6.2/8.2 sheave set will pull 176.72 lbs. on the tight side of the belt.

A 10 HP motor with the 12.0/16.0 sheave set will pull 91.3 lbs. on the tight side. (That is a drop of 85.42 lbs. to transmit exactly the same 10 HP to the machine-almost half the effort.)

Static pull vs. dynamic pull…

Static pull is the amount of tension a mechanic puts on the belt when the drive is installed. This tension is equal on both halves of the belt.

Let's say the mechanic puts 175 lbs. of static pull on a drive. With the 6.2/8.2 sheave set, we see a 176.72 lb. pull. How, though, do these conditions affect the static pull? We can calculate it as follows: There is a pull of 175 lbs. of static pull plus 176.72 lbs. of dynamic pull on the tight side. *That adds up to 351.42 lbs. of pull-quite a hefty force.*

On the slack side, the static pull is reduced by 176.72 lbs. Thus, 176.72 lbs. of dynamic pull is subtracted from the 175 lb. static pull. That leaves us with a a deficit of almost 2 lbs.. The belt's slack side flops around loosely, the belt slips and squeals. What, then, does the mechanic do? Tighten the belt, of course. This brings the static pull much higher than the dynamic pull to keep the drive running quietly.

With the larger 12.0/16.0 sheave set, the dynamic pull is 91.3 lbs. plus the static pull of 175, which only leaves a tight side pull of (91.3 + 175) or 266.3 pounds and, conversely, a slack side pull of 83.7 lbs. There is, accordingly, no need to readjust the drive.

Keep in mind that all of these examples relate to "running condition" and ignore the starting pull on the drive. That short-duration force can be at least three times the dynamic pull.

The finer points Every rubber drive belt is essentially an elastic drive medium that, because of Dynamic Belt Pull, will stretch longer on the tight side than on the slack side. By making the sheaves larger than the original drive, the belt pull is reduced and so is the amount of belt stretch.

When a belt stretches every revolution under load, driven RPM is reduced. This difference-*or "allowed belt slippage"*
-must be kept below 2%. Above 2%, the belt returns to its slack-side length part-way around the driver sheave, and stretches to its tight-side length partway around the driven sheave. This movement, when in contact with the sheaves, causes destructive wear and heat. Under 2%, the rubber distorts, yet maintains its grip on the sheave, absorbing the movement and releasing heat to the air between sheaves. This results in a long-lasting, cooler running drive.

To see how this works, consider this: A driven sheave's 1293.75 RPM, calculated in the foregoing manner with a 2% slippage, is actually 98% of the calculated RPM (slower than designed), or:

1293.75 x .98 = 1267.875 RPM

If the driver is actually 1725 RPM, then the measured driven RPM should be between 1268 and 1293 RPM to minimize any destructive effects of slippage.

Belt wrap on the smaller sheave also is a concern. For example, if the small sheave wrap is about 170 degrees on the original, with the 12.0 P.D. sheave and the same shaft center distance and drive ratio, there will be the same angle of contact or belt wrap.

The small sheave circumference, times the amount of wrap will indicate the length of belt gripping the sheave.

The formula for the length of belt wrap is:

= (P.D. x 3.1415) x (degrees of wrap/360)

The improved design example of a 12.0/16.0 drive is:

= (12.0 x 3.1415) x (170/360) = 37.698 x .4722 = 17.8" of belt contacting the sheave Using the original example of the 6.2/8.2 drive for comparison:

= (6.2 x 3.1415) x (170/360) = 19.48 x .4722 = 9.197" of belt in contact

An increase from 9.197" to 17.8"of belt in contact will be almost double-a definite improvement in grip.

Installing the new, larger drive is a bit more demanding with regard to parallel and angular alignment in both the vertical and horizontal planes. For instance, with a motor and a fan (both of which have horizontal shafts), an alignment is usually set closely across the hub area and parallel and angular misalignment are removed. When checking vertical alignment between the motor and fan sheave, one sees that the top of the motor sheave tilts toward the motor and the bottom away from the motor. This drive twist- *usually ignorable on smaller drives*-becomes a real issue on larger drives, and requires adjusting out to avoid premature wear. When attempting to extend drive life two or three times the current life, remember that small details like this can adversely affect the desired long-term deliverable life.

If you had a chance to watch both the original and new drives under load, you would have noticed that the smaller unit usually exhibited a noticeable slack-side flop, while the tight side would remain stable. The new, larger drive, however, runs with the tight-side straight and stable, and the slack-side stable with a slight outward bow, when it is eyed down the length.

If the original small drive were started up, you would hear a definite thud or pound, as if the unit were hit by a huge rubber hammer. On the other hand, the new, larger drive's start-up thud will be almost silent-as if a small rubber hammer is being used. Generally, the new drive will run

more smoothly, all the way around.

It is important to note that heat is the greatest single destructive element to which rubber can be subjected. Heat makes rubber harder and less flexible-and more susceptible to fracture and breakage. According to the Rubber Manufacturer's Association, V-belts will operate acceptably at temperature from -30 to 140 F. An internal temperature rise of 18 degrees F in this type of belt will decrease its service life by 50%. With our previously discussed drive modification, the heat generated in repeated stretch-and-relax cycles of the slack and tight sides has been reduced. The slippage heat generated in gripping the sheaves has been cut. The heat generated in bending or flexing around the smaller-diameter sheaves drops significantly.

- A higher horsepower per belt capability, which, in some cases, reduces the number of belts, or drops the belt profile to a smaller one
- Increased belt-wrap length for more gripping surface
- Less belt-bending occurs around a larger sheave set, equating to less flex-generated heat
- Less slippage or stretch-generated heat
- Higher belt velocity and sheave rim speed that enhances drive air-cooling
- New belts for the drive on the same center distance that are longer, resulting in any generated heat being dissipated over a much longer length of belt
- Much cooler operating belts and sheaves, equating to belts that last exponentially longer
- Much lower belt pull, which lowers drive-end bearing loads and start-up shock
- A more efficient belt drive that leads to definite energy savings, given continuous 24/7 power demands
- With regard to PMs, the elimination or revision of monthly, quarterly or semi-annual inspections-down to only one annual inspection. (Even then, you may not have to readjust belt tension in the first or second year, if the initial installation includes a tension check after the first two to 48 hours of operation, then again, after one-month's running, to take up the initial stretch and set the new belts.)

Doing it right

What can you really expect when you increase the velocity of your drive belts?

After five years of 24/7-365 operation, such belts will be riding lower in the sheave and the sheave edges will be visible above the crown of the belt, but not running on the bottom of the groove. When the belts are removed, you will find them to be surprisingly supple and flexible, with no segmented cracks anywhere. When the sheaves are inspected, there will be very little indication of the Vgroove wearing to a U-shape. Sheave grooves will appear highly polished, almost like chrome plating-ready for another five years of continuous operation with a new set of

belts.

According to industry sources, well designed and carefully installed drives typically will generate very little heat. Furthermore, the cooling effect of the belt through the air will tend to cool the whole drive to a temperature very close to ambient. Such drives are expected to run 24/7-365 for three to five years.

Calculate, specify and install drive belts correctly and they will require very little maintenance for years.

Gary Burger worked himself up through the ranks of Canadian Occidental Petroleum, Durez Plastics Division, to become maintenance supervisor and chief engineer. He then joined the Stevenson Memorial Hospital maintenance team in Alliston, ON, Canada, as chief engineer. Over the past 10 years, he has helped lower this facility's energy consumption by over 64%, while keeping it all within budget. E-mail: burgergary@hotmail.com

Joomla SEO powered by JoomSEF