CHAPTER XX.
BELTING.
§ 276.
SELF-GUIDING BELTING.

Belt pulleys are indirect acting friction wheels (§ 191) and the belt itself is a tension organ combining the functions of driving and guiding (§ 261). Those belts which act without requiring the use of special guiding devices may be called self-guiding belts. This action is obtained by the use of cylindrical pulleys when the edge of the prismatic belt runs in a plane at right angles to the axis of the pulley; or in other words, when the middle line of the advancing side of the belt lies in the plane of the middle of its pulleys.

When a belt runs upon a conical pulley in a direction normal to its axis, its tendency will be to describe a conical spiral path upon the pulley, as will readily be seen upon the examination of the development of the surface of the cone, Fig. 840.

If the pulley is made with a double cone face or a rounded face, Fig. 841, the tendency will be for the belt to run at the middle of the face even when the direction of the belt is not exactly correct.

For leather belting, with a height of the crowning or curvature of the face $s = \frac{1}{4}$ of the width of face, the belt may deviate from the plane of the pulley by $2\frac{1}{4}^\circ$ (tan $= 0.025$), while for cotton belting, on account of the lesser elasticity of the material, the crowning should not exceed $\frac{1}{4}$ of the face, thus reducing very materially the permissible deviation. In ordinary circumstances at least one of a pair of pulleys should be made with rounded faces.

The simplest arrangement of self-guiding belting is that for parallel axes, Fig. 842, where $a$ and $b$, $a$ being for open belt and $b$ for crossed belt, either arrangement being suitable to run in either direction.

For inclined and intersecting axes self-guiding belts are not suitable, except in the case of inclined axes in which the trace $S$, Fig. 843, of the intersection of the planes of the two pulleys passes through the points at which the belt leaves the pulleys. The leading line then falls in the middle plane of each pulley, but the following side of the belt does not, hence such systems can only be run in one direction. The leading points in the figures are at $a$ and $b$. The arrangement gives an open belt when the angle $\beta$ between the planes of the pulleys is $\alpha^\circ$, and a crossed belt when $\beta = 180^\circ$. If $\alpha = 90^\circ$, the belt is half crossed (or as commonly called, quarter crossed); if $\beta = 45^\circ$, it is quarter crossed.

*The above geometrical construction is only approximate; for an exact solution see a paper by Prof. J. E. Webb, Trans. Am. Soc. Mech. Engs., Vol. 11, 1883, p. 246.

The leading off angle may be made as much as $25^\circ$, which occurs when the distance between the axes is equal to twice the

Fig. 842.

GUIDE PULLEYS FOR BELTING.

When a belt transmission is arranged with guide pulleys, the proper guiding action is obtained when each guide pulley is placed at the point of departure of its plane with $\theta$ of the next following pulley.†

Fig. 844.

In Fig. 844 examples are given of guide pulleys for parallel axes, all three pulleys lying in the same plane.

At $a$ is shown a belt transmission with tightening pulley $b$, a device for transmitting motion when great difference of speed is desired. In this case the guide pulley $C$ is as large as the driver $A$, and if desired may also be arranged to act as a tightening pulley at $C$ is Weaver's device for similar use. In this case two belts are used, and the device has been used for driving circular saws. The pulleys should be fitted to run very smoothly in such devices.

The cases in Fig. 845-846 have parallel axes with two guide pulleys. In the first case the guide pulleys are placed in planes tangent to both operating pulleys, and hence driving may occur in either direction. Usually, however, it is required to provide

† See also the paper of Prof. Webb referred to in the preceding note.
‡ Robert's patent (German) for driving the drum of a threshing machine.
§ See Cooper's use of Belting. Phila., 1876, p. 137.
for motion in but one direction, in which case the second form is used as being simpler of installation. The pulley B may be used as one of the guide pulleys, in which case it may be placed loose upon the same shaft as A, and C or D be made drivers or driven.

By placing the guide pulleys between the axes of A and B, instead of beyond them, they will revolve in the same direction, and may be made fast upon one shaft, as in Fig. 847; this arrangement admitting of motion in only one direction.

In Fig. 848 is an arrangement for inclined axes, which is a modification of Fig. 846, as will be seen by the dotted lines. The guide pulleys run in opposite directions, but may conveniently be placed upon the same shaft.

In Fig. 849 is shown an arrangement of quarter-twist belts with guide pulleys. One side of the belt is placed in the intersection S S of the planes of the two pulleys. From any point e

drawn, and in the planes of these tangents the guide pulleys C and C' are placed. Under these conditions the rotation may be in either direction. The arrangement shown in Fig. 850 occurs when the line S S passes through the middle of one of the pulleys.

* An example is Jacob's grinding mill with 40 sets of stones; see Uhland's Praks. Mocchl. Können, 1868, p. 85, 1869, p. 240.

A simplification of the general case occurs when, as in Fig. 853, the guide pulleys fall upon one and the same geometrical axis which is parallel to the axes of both transmitting pulleys. In this case the only inclination of the belt is that given it by the guide pulleys. The rotation can be in but one direction, viz.: that shown by the arrows; if the reverse is desired, the guide pulleys must be placed as shown in the dotted lines. If the inclination of the shafts is too great the belt will be liable to drop off when the pulleys come to rest. The use of guide pulleys involves special hanger, a practical form for which is shown in Fig. 854.
The vertical axis is provided with an oil hole and is fitted by a ball and socket bearing to the bracket $D$. The flange on the lower edge of the pulley keeps the belt from falling off the parallel shafts, one of which intersects its axis at right angles, the other passing beneath.

Another arrangement, devised by the author, is given in Fig. 860. In this case the following side of the belt is passed over an idler pulley, $C_2$ or $C_3$, and a second time around the driver (see also Fig. 752) by which the angle of contact is doubled, and the modulus of friction increased. This may be called a double-action transmission. The cross section of belt may be made half of a single-acting transmission, so that in spite of the increase of length an economy of belting is obtained. One of the guide pulleys may also be used for a tighter. These devices will also be considered in connection with rope transmission (Chapter XXI) to which they are especially applicable.

\[ \text{§ 278.} \]

## FAST AND LOOSE PULLEYS.

Fast and loose, or tight and loose pulleys, as they are sometimes called, are generally used in connection with another belt transmission in order to throw the latter in and out of action, the belt being guided by a belt shifter, which by the means of forks or finger-bars, enables the moving belt to be shifted. These shifting devices may properly be regarded as guide pulleys, and are sometimes fitted with rollers, as shown dotted in Fig. 861, at $c$ and $c_1$.

It is preferable to have the loose pulley upon the driven shaft, since the belt then can be shifted with a gradual spiral action by the shifter $F$, Fig. 861. It is best for the driving pulley to be made straight face, or if two flat pulleys are used side by side on the driving shaft, these should have very slightly rounded faces, if the belt is to be shifted promptly and readily, and for the same object the shifter should be placed as close to the driven pulleys as possible. The loose pulley should be kept thoroughly lubricated, and for this purpose numerous oiling devices have been made. The friction between the hub and shaft acts as a driving force upon the loose pulley, and this has been a source of numerous accidents. This action is avoided in the arrangement in Fig. 85a, in which the loose pulley is carried on a consecutive and stationary sleeve $f$.

A variety of mechanical belt shifting devices have been made, the desire being to prevent the action of the belt from moving the shifter. A useful form is Zimmermann’s Shifter, Fig. 89a.

\[ \text{† Such rollers as especially necessary for shifting cotton belts, which are liable to catch on the shifter fingers, and even larger rollers are best in such cases.} \]

\[ \text{‡ See Berliner Verhandlungen, 1869, p. 125. This has been used by the Society for Prevention of Accidents in Milwaukee.} \]

\[ \text{§ See Berliner Verhandlungen, 1883, p. 174. Ritterhaus, Belt shifters.} \]
The shifter bar \( F \), to which the fork \( G \) can be clamped at any desired point, is operated by the lever \( H \), which turns upon an axis at \( I \), forming a "dead" ratchet mechanism. The similarity to the ratchet devices of Figs. 754 and 755 will be observed. The movement of the bar is effected by connection at \( K \) or \( K' \).

![Fig. 864.

Fig. 864 shows a shifter for quarter-twist belt. In this form, devised by the author, the guide pulley, which is required to support the belt, also serves as a shifter to move the belt to and from the belt pulley \( B \), and loose pulley \( R \). If these pulleys are given greater width than that of the belt, as shown on the right, a vertical adjustment can be given to the upright shaft; a condition sometimes required in grinding mills and similar machines.

\[ \beta \] which can occur. For any value of \( \beta = CA \), draw \( MN \) perpendicular to \( MA \) and make \( MN = \text{the arc} \ MC = a \). Drop the perpendiculars \( M P, NQ, PA, NC \), and draw \( NO \) perpendicular to \( MP \). \( NO \) will then be \( \frac{a}{2} \) sin \( \beta \). Through \( N \) draw \( QK \) parallel to \( AB \), and we have \( AQ = PQ + PA = a \) \( (\beta \sin \beta + \cos \beta) \). By taking successively all the values of \( \beta \) between \( 0 \) and \( 90 \) in this manner, we can determine the path of the point \( N \), which will be the evolute of a circle, \( CN \).

From the above construction, if we now draw \( DE \) parallel to \( BA \), and take its middle point \( F \), we have \( DF = EF = \frac{a}{2} \), and hence the proportion:

\[ DF : DE = \frac{a}{2} : \frac{a}{2} = a : a = x : x, \text{ and by similar triangles:} \]

\[ TK = \frac{a}{x} \times QA = \frac{a}{x} (\beta \sin \beta + \cos \beta) \]

This value is dependent upon \( \frac{a}{x} \). If we prolong \( BF \) until it intersects \( AC \), prolonged, the resulting length \( AC' = BC' \) will be equal to \( AB \) and \( \beta \). By working \( BG = t \), and drawing \( GH \) parallel to \( A'B' \), we have \( GH = \frac{a}{x} \). This length being transferred to \( IJ \) gives \( IT = \frac{a}{x} (\beta \sin \beta + \cos \beta) \). We then have only to use \( \frac{a}{x} \) sin \( \beta \) to solve the problem.

Make \( A'F = \frac{a}{2} \), and we have the perpendicular \( RS = \frac{a}{2} \) sin \( \beta \). By levelling this length off above and below \( TV \), we obtain the points \( U \) and \( V \), and this finally gives \( IT \) for the radius \( K \) of the larger cone pulley and \( IF \) for the radius of the corresponding smaller cone pulley. By solutions for successive values of \( \beta \), we obtain the curve \( U'XV \), which can be used for the determination of the radii of any desired pair of pulleys, each pair of ordinates measured from \( HT \) belonging to corresponding pulley on each cone.

In practice it is usual to find one of the cone pulleys given and the dimensions of the other required. In this case \( VU \) may be taken as the difference \( R - R' \), between the radii, where the steps uniform. By taking this difference \( R - R' \), in the dividers, and finding the equivalent ordinates \( UV \) on the curve, and then adding \( VT = R' \), the axis \( HT \) is found.

In order to use the curve conveniently, it may also be laid off left-handed, as shown in the dotted lines \( D'X'E' \).

The use of the diagram will be rendered still more convenient if we omit the unnecessary value \( l \). This enables us to distort the curve in the direction of the abscissæ to any desired extent.
off toward C, the corresponding radius $X d$ and prolong the axial line $d' e'$ to its intersection $w$ with $C E$. Then lay off the given geometric ratio on $C X$, considering $X d$ as 1 (shown in the diagram by the small circles for the ratios $\frac{1}{2}, \frac{1}{3}, \frac{1}{4}, \frac{1}{4}$), and draw rays from $w$ through the points of division, and these rays will intersect the curve at the corresponding points for the pulley radii $R$. We then have for the radii:

\[
a : 1 \text{ and } a' : 1 \text{ for the ratio } 1 : 4
b : 2 = b' : 2 \text{ for } 2 : 4
c : 3 = c' : 3 \text{ for } 3 : 4
d : 4 = d' : 4 \text{ for } 4 : 4
e : 5 = e' : 5 \text{ for } 5 : 4
c : 6 = c' : 6 \text{ for } 6 : 4
\]

Cone pulleys may also be made continuous, thus becoming conoids upon which the belt can be shifted to any point by an adjustable guide or shifter. Such conoids are used for driving the rollers in steam machinery. Each pair of conoids is shown in Fig. 869, the proportions having been determined by the graphical scale. The angular velocity varies in an arithmetical ratio as shown.

The curve $E Y A$ in the scale shows the extent to which the axial line may approach $A E$; this distance must not be less than $X + R = a$, from which $V Y = \frac{1}{2} (A E - V U)$.

\[\text{§ 260.}\]

**Cross Section and Capacity of Belts.**

A belt of rectangular cross section of width $b$, and thickness $h$, will be subjected to a tension $T$ on the right side (see § 254), which must be proportioned to sustain. $F$ is the permissible stress for the unit of cross section, we have, therefore, $T = b h F$.

The minimum ratio which $T$ bears to the transmitted force $P$ is dependent upon the stress modulus $r$, since $T = r P(\frac{1}{2} b h)$. But $r = \frac{D}{\rho - 1}$, in which $\rho$ represents the modulus of friction $c \theta$. Hence, if $N$ is the horse power transmitted for a belt speed of $v$ feet per minute, we have:

\[N = \frac{P v}{2300} = \frac{b h F v}{2300} \tag{264}\]

This enables us to determine the cross section of the belt, but in practice the width of the belt is the variable factor, the thickness usually being determined by commercial considerations, and limited to few definite sizes.

If we let $q$ represent the cross section of the belt in square inches, we have:

\[N = q \times S \times \frac{v}{2300} \tag{265}\]

This formula is very useful, since it may be used to determine the capacity of a belt from its cross section and velocity. If we put $N = \frac{N}{\rho}$, we have:

\[\frac{N}{v} = \frac{1}{3300} \times \frac{S}{\rho} \tag{265}\]

The value depends upon the material and stress modulus, the latter including the elastic and upon $f$, which itself depends upon the material of both belt and pulley; it may also be considered as dependent upon $e$, independent of the material, in the same manner as was the subject of specific weight. The author has called this value $N_e$, the specific capacity of a belt.

It will be seen that when this specific capacity is determined for any kind of belt, the proper cross section for the transmission of a given horse power $N$ can readily be found, since the velocity $v$ can be chosen, and we have at once:

\[N = \frac{N}{v}\]

For the determination of the specific capacity of any kind of belt it is necessary to find the constants $S$ and $\rho$.

The materials used for belting are:

- Tanned leather.
- Cotton, woven and treated with oil.
- Rubber, interlaid with linen or cotton webbing.

In practice the value of $S$ to be used must depend much upon...
judgment, the value being governed to a great extent by the
cal quality of the material. Customary values are for:

- Leather: \(S = 4000\) to \(6000\) lbs.
- Cotton: \(S = 2000\) to \(4000\) lbs.
- Rubber: \(S = 3000\) to \(5000\) lbs.

The thickness \(\delta\) for single leather belts varies from \(\frac{3}{16}\) to \(\frac{1}{2}\) double, triple, and so on; and even quinquepartite thicknesses being
sometimes used, the thicknesses being secured by cement, and
screwed or riveted together. Cotton belts are usually from \(\frac{3}{16}\) to \(\frac{1}{4}\) thick, while rubber belts are made of any desired thickness.
A web of canvas being interlaid between the successive
thicknesses of rubber.

The stress modulus \(T\) depends upon \(\alpha\) and \(\beta\) and the latter coefficient varies with the age of the belt, being greater with those which have been used some time than with quite new belts. It
is advisable, however, to make all calculations as for new belts,
in which case we have for smooth iron pulleys:

- Leather and cotton: \(T = \frac{0.16}{10} \times \frac{\text{to}}{2.5} \times \frac{\text{to}}{2.5} \times 1.5 \times \text{to} \times 2.1\)
- Rubber: \(T = 0.20 \times 0.25\), \(T = 1.5 \times 2.1\)

These give as approximate values for:

- Leather and cotton: \(\alpha = \frac{2.5}{10} \times \text{to} \times 1.5 \times 2.1\)
- Rubber: \(\alpha = \frac{2.5}{10} \times \text{to} \times 1.5 \times 2.1\)

By using these values together with those given for \(S\) in (262)
we get for the specific pressure for bending

- Leather: \(N_y = 0.0002\) to \(0.0008\)
- Cotton: \(N_y = 0.0005\) to \(0.0008\)
- Rubber: \(N_y = 0.0005\) to \(0.0008\)

These are based upon low and moderate speeds; say up to
9000 feet per minute; and the variations between the limits given
are due to the differences in strength of various kinds of
leather and canvas used.

The resistance to bending or stiffness of a belt must be taken
into account, and the ratio of the thickness \(\alpha\) to pulley radius \(R\)
must not be too great. Practical experience has shown that

\(\frac{\alpha}{R} = \frac{1}{10} \text{ should not exceed to obtain best results.}\)

From the known stress and the thickness of the belt the
the superficial pressure \(p\) between belt and pulley may be
calculated. We have only to substitute in (241) for the width \(B\)
the length of the surface of contact, the width \(B\) of the belt itself, and
sin \(\alpha = \frac{\beta}{\alpha}\) we get the simple relation:

\[ p = \frac{\alpha}{S} \]

**Example 1.** - Required a leather belt to transmit \(100\) H. P. and the speeds of its revolutions. (Taking the speed capacity of \(1000\) and the desired velocity of belt at 5000 feet per minute, we have \(p = 0.00002 \times 0.00002 = 4.5\) sq. in. cross section.

If we use a double belt \(\frac{1}{4}\) thick, the width should be \(\frac{1}{4} \times 12 = 12\) inches.

For the driving pulley we have \(x = \frac{k}{2} = 7.5\), and \(y = \frac{y}{x} = \frac{7.5}{2} = 3.75\).

For the superposition \(p\) we have \(p = \frac{2000 \times \frac{1}{4}}{16} = 1100\) lbs. Also

\(B = 9.5 P = 29700\), hence \(S = \frac{29700}{29700} = 97\). We have also \(D = 150, F = 150\),
which gives \(S = 97\), or a mean of 95 lbs. which in (264) gives a mean value
\[ p = \frac{B}{N_y} = \frac{12}{0.00002} = 600, \text{ on the large pulley, and } p = \frac{B}{N_y} = \frac{12}{0.00002} = 600, \text{ or nearly } 50\text{ pounds.}\]

This is verified above, if \(\alpha = \frac{1}{10} \times 250\),

\[ \frac{75}{2} \times \frac{1}{12} \times \frac{2}{3} \times \frac{0.00002}{150} = 1000\]

which is what \(P_{\delta} \) above.

**Example 2.** - What horse power can be transmitted by a cotton belt \(\frac{1}{4}\) thick at \(2000\) feet per minute, taking the specific capacity at \(2000\), which has been found satisfactory in practice, we have from (262) \(N_y = 0.0005\), \(N_y = 0.0008\), \(N_y = 0.0002\), \(N_y = 0.0008\). The belt speed at 2000 feet per minute, and the belt speed on foot. Taking the specific capacity at \(2000\), we have \(N_y = 0.0005\), \(N_y = 0.0008\), \(N_y = 0.0008\), \(N_y = 0.0008\). If we make \(B = 5.5\) we have a

\[ p = \frac{B}{N_y} = \frac{5.5}{0.00002} = 275\text{, say}\]

\[ p = \frac{B}{N_y} = \frac{5.5}{0.00002} = 275\text{, say}\]

* For cotton the thinner belts from \(\frac{1}{4}\) to \(\frac{1}{2}\) are preferable.

For extraordinary cases the fundamental formula should always be applied. For double-acting belts, as in Fig. 803, at which \(x = 2 x\) instead of \(x\), the value \(f = 1\), and the modulus of stress is only \(\alpha\) of the preceding value, hence \(g = \alpha\) is reduced in the same proportion. If the belt velocity \(v\) is very high, it is no longer permissible to neglect the influence of centrifugal force. For a speed \(v = 5000\) feet, and a stress \(S = 500\) pounds (see § 261) the exponent in the friction modulus becomes \(0.5 + f\) instead of \(f\), which for \(f = 0.16\) and \(\alpha = x\), gives \(f = 0.16 \times 0.10 \times 0.08 = 0.012.\) This gives \(\alpha = 2.51\) or about \(f\) of the normal value, which requires one eighth greater cross section \(g\) for the belt. The highest limit of belt speed in ordinary practice

\[ \text{appears to be } \frac{1000}{20} = 500\text{ feet per minute.}\]

### Chapter 11: Examples of Belt Transmission

The table of existing examples of belt transmission on next page will serve to furnish data for comparison with calculated results.

The great variations in the values of \(S\) and \(N_y\), in the following table are not surprising when the differences in the quality of material, and the various conditions are considered. Many leather belts are working under high stresses which are only practicable because of the excellence of the material. Some such belting can be operated under stresses as high as 2000 pounds, which enables much lighter sections to be used. Many belts which appear to be excessively heavy have simply been calculated to work at a moderate stress.

The plausible but erroneous idea that the pressure of the atmosphere influences belt action cannot be admitted. It is
controlled not only by the fact that the same coefficient of friction exists for ropes as for belts, but also by the recent and careful experiment made in a vacuum by Delcourt which confirmed the theory of the modulus of friction.

### Chapter 12: Belt Connections

The various methods of connecting the ends of belts generally give a greater stress at the point of connection than in the body of the belt. The attempt to reduce this weakness and also provide for the greatest facility in the making of the joint, has caused a great variety of methods to be proposed; some of the best of these are here given:

**n. b. c. d.**

In Fig. 870, a is a lap joint sewed with hamper thread; \(\delta\) a lap joint secured with screw threads; \(e\) is a plate coupling; \(f\) a plate and prongs being made in one malleable casting and the prongs bent over and clinched after insertion in the belt, several clamps being used for belts more than \(4\) inches in width. As is shown belt lacing for use with single or double belts. The upper one has the defect of giving intersections which make the lacing itself thick, and the knot at the edge of the belt reduces the strength of the joint. These defects are both avoided in the lower form, which is an American belt lacing.

* In the construction of the Airberg tunnel a hoisting machine was used in which the belts had a velocity of \(1200\) feet per minute, which worked well for fourteen months.

Delcourt has used the upper form of lacing for a belt of \(36\) wide, \(2\) thick, with excellent performance and durability.

* See Cooper, Use of Belting, p. 128.
**EXAMPLES OF BELT TRANSMISSION.**

<table>
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<th>No.</th>
<th>Horse Power P</th>
<th>( n )</th>
<th>( R )</th>
<th>( v )</th>
<th>( P )</th>
<th>( b )</th>
<th>( d )</th>
<th>( s )</th>
<th>( W )</th>
<th>Remarks</th>
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<td>74.4</td>
<td>27</td>
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<td>497</td>
<td>( \frac{70}{144.4} )</td>
<td>99.4</td>
<td>48</td>
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<td>445</td>
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<td>96</td>
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<td>( \frac{48}{120} )</td>
<td>120</td>
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<td>60</td>
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<td>412</td>
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<td>334</td>
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<td>3130</td>
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<td>0.72</td>
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<tr>
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<td>29.5</td>
<td>2706</td>
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<td>21</td>
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two or three separate pulleys combined in one, except that the proportions of the arms should be $0.8$ or $0.7$ times that of single arm pulleys, or in the proportion of $\sqrt[3]{2}$ and $\sqrt[3]{3}$.

The thickness of the rim may be made: $h = 0.3$ to $0.4$ for $h$; this being frequently turned much thinner. The width of face should be from $\frac{1}{8}$ to $\frac{1}{4}$ of the width of the belt.

The thickness of metal in the hub may be made $w = h$, to $\frac{1}{2}h$. The length of hub may be $b$, for single arm pulleys, and $2b$ for double arm pulleys. Light pulleys are usually secured to the shaft by means of set screws, as in Fig. 873 and 877; heavier ones are keyed as in Fig. 194, either with or without set screws.

For many purposes pulleys are made in two parts, each being commonly called "split pulleys." The forms of split pulleys are shown in Figs. 873 to 875.

The arrangement of the two halves is clearly shown, that of Fig. 873 with hollow clamping section, being especially good.

The form in Fig. 875 is the design of the Walker Mfg. Co. of Cleveland, Ohio, the clamps being made of malleable iron or steel. In all three cases there is no especial method of fastening the shaft. In England and America pulleys are frequently made with wrought iron rims and cast iron hubs. This construction greatly simplifies the casting of the arms, and at the same time gives pulleys $25$ to $50$ per cent lighter than those of cast iron, which in large transmissions greatly reduces the friction at the bearings of the shafting.

Fig. 878 shows the sliding pulley. The rim is curved in bending rolls, and also given a rounding face, and is countersunk for the rivets at the attachment of the arms. The pads on the arms are truly finished, as is also the rim after it is riveted on, thus giving an accurate and useful pulley.

A metal pulley by the Harford Engineering Company $60^\circ$ diameter and $16^\circ$ face weighed 370 pounds. A cast iron pulley of the same dimensions made by the Berlin-Asphalt Works, weighed 700 pounds, and one by Briegleb, Hansen & Co., a little narrower face weighed 528 pounds.

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* Leclaire has used a dynamometric belt-stretcher for tensions of $\sqrt[3]{3}$, $(T + I) = 380$ pounds.